Doctoral Thesis

The decentralized approach to achieve comfortable indoor environments in tropical climates using low-exergy techniques of integrated design

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THE DECENTRALIZED APPROACH TO ACHIEVE COMFORTABLE INDOOR ENVIRONMENTS IN TROPICAL CLIMATES USING LOW-EXERGY TECHNIQUES OF INTEGRATED DESIGN

A thesis submitted to attain the degree of
DOCTOR OF SCIENCES of ETH ZURICH
(Dr. sc. ETH Zurich)

presented by
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2014
Dedicated to Appa and Amma who have always been the constant source of my inspiration and encouragement
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The burgeoning growth of cities around the world is becoming increasingly responsible for the increase in energy demand. With more than 50% of all industrial products produced and material that is transported being for the building industry, buildings today consume a large share of a city’s energy. Of this, air conditioning systems are known to gourmandize maximum energy. That means more than 50% of CO₂ emissions are related to buildings.

Investing therefore in serious time and money to strategize ways to improve building air conditioning systems and make them more efficient is the call of the day. This would then lead to a sustainable use of energy and resources.

Traditional systems use large centralized Air Handling Units (AHUs) to move cool air to building interiors. But the movement of water, which is more effective than air in removing heat owing to its higher heat capacity, could be a better solution. The transition from convection cooling based all air centralized air conditioning systems to ceiling attachable radiant cooling based all water decentralized air conditioning systems could be considered as a welcome change to air condition buildings. This type of a system could also do away with the false ceiling and decrease floor to floor height by about 20%. The advantage is that radiant cooling is a process employing high temperature cooling techniques, with chilled water at 15°C to 17°C being used to cool room interiors. Traditional systems on the other hand use chilled water at 6°C to 8°C, which are supplied to air handling units. This decrease in chilled water supply temperature directly translates into increase in energy demand. Centralized systems also have both processes, of cooling and dehumidification being performed in the AHU. Decentralized systems on the other hand, bifurcates this combined process into two separate activities. The high temperature radiant panels take care of temperature control and an independent ventilation system take care of dehumidification. This could prove very beneficial in tropical climates, where only the highly humid outside air that is needed for ventilation needs to be cooled and dehumidified by low temperature chilled waters, while the room air can be cooled at much higher temperatures using radiant panels. An issue that has plagued the adoption of such systems though, in the tropics has always been of fears of condensation on radiant panels.

This study scrutinizes some of the problems that could be associated with using such systems in the tropics and gives recommendations on solutions to such problems. Some of the issues that are investigated in this research are: challenges posed by addition of internal humidity from building materials, which could be about 31.6% greater at the ceiling level when compared to the floor level as seen in this study, characteristics of air leakage through floor joints and infiltration, the importance of cooling capacity of radiant panels, risks of condensation on radiant panels, stratification of air in the indoor space and how operation of radiant panels effect stratification, how perimeter solar heat gain affects stratification etc. The BubbleZERO laboratory, which houses the decentralized system in Singapore, was used for
conducting most of the experimentation. The study found that an increase of 50% of chilled water loops on radiant panels can increase cooling capacity by 24.7%. A thermal comfort and Indoor Air Quality (IAQ) study was carried out to reveal the working of the decentralized system under different operating conditions. The energy required to run the decentralized system and its comparison with conventional systems installed in the tropics was also investigated to get a sense of energy efficiencies. The study found that reducing ventilation rates by 33.5% could decrease chilled water energy demand by 13.3%. Finally an alternative ventilation mechanism involving CO₂ adsorption techniques was explored for its feasibility in maintaining indoor air quality in the tropics.
e. Abstrakt


Deshalb sollte Zeit und Geld darauf verwendet werden, Klimaanlagen zu verbessern und ihre Effizienz zu erhöhen, so dass ein nachhaltiger und bewusster Umgang mit Ressourcen und Energie erreicht werden kann.


Das BubbleZERO-Labor, das die dezentralen Systeme in Singapur beherbergt, wurde für die Durchführung der meisten Experimente verwendet. Ein Ergebnis dieser Arbeit ist, dass eine Verdoppelung der Anzahl an Wasserschleifen eines Deckenpanels die Kühlleistung um 24.7% erhöhen kann. Eine Studie zum thermischen Komfort und zur Raumluftqualität wurde durchgeführt, um die Arbeitsweise des dezentralen Systems unter verschiedenen Betriebsbedingungen zu untersuchen. Der Energiebedarf, um das System zu betreiben, wurde ebenfalls untersucht und mit einem konventionellen System verglichen, um ein Gefühl für die Energieeffizienz zu bekommen. Es zeigte sich, dass eine Reduktion der Zulufrate um 33.5% zu einer Abnahme des Energiebedarfs zur Kaltwassererzeugung von 13.3% führte. Ausserdem wurde ein alternatives Lüftungskonzept untersucht, das auf der Adsorption von CO2 beruht.
This chapter highlights the need to look into energy demands of air conditioning systems in tropical buildings. Traditional cooling concepts used in centralized systems are discussed to understand the problems associated with such systems. An introduction to low exergy based decentralized concept of conditioning buildings is explored as an alternative air conditioning mechanism. The construction of a prototype laboratory in Singapore called the BubbleZERO, housing decentralized systems is discussed along with its variety of operational capabilities. The research objectives, hypothesis, motivation and background of the research are also presented in this chapter.

1.1. Motivation & Background

The movement of people from agriculture driven rural lands into service, technology and industry driven urban cities is occurring at a rapid pace today. This geographical shift of population is causing urbanization which, maybe argued as a welcome move but it also brings with itself many new challenges that need to be solved. The Global Health Observatory (GHO) of the World Health Organization (WHO) provides certain numbers to ascertain this pattern of human movement [1]. The report claims that one hundred years ago, 2 out of every 10 persons lived in an urban area. By 1990, less than 40% of the global population started to live in cities, but as of 2010, more than half of all people live in urban areas. It is also predicted that by 2030, 6 out of every 10 persons will live in a city and this number is slated to increase to 7 out of every 10 persons by 2050. This surge in urban population will soon demand for high density living and better comfort living conditions. The important thing is to look into the trade-offs that accompany this demand and how best it can be addressed. Energy demand is one such trade-off that can be associated with urbanization. High density living is causing buildings to go vertical and development of cities is increasing causing buildings to become air conditioned; this could only bring an increase in energy demand in pumping energy and the energy needed to transport air for air-conditioning purposes. Air conditioning of buildings irrespective of its typology is slowly being perceived as a necessity rather than luxury. The per capita energy demand of cities will therefore only increase with time. Some already performed research claims that one third of the world’s primary energy is consumed during heating, cooling and lighting applications [2] in the buildings we work, live and play. A very careful introspection of the energy consumed by air conditioned buildings, leads to questions such as why such a large quantum of the building’s energy demand goes to run its air conditioning equipment. What is the goal of air conditioning? Is air conditioning being performed in the most efficient way? Is there a requirement to change the focus from quantity of energy used to run these air conditioning systems to the quality of energy used to run them? Most people consider air conditioning to be cooling alone, but the best way to provide this cooling may not depend solely on the conditioning of air. The reason being that air has a low density and heat capacity, which makes it a poor medium for heat transfer. The use of water as a heat transfer medium could be more efficient as the latter has much greater heat capacity than the former. This could directly then translate into a correlation with energy demands, which could be reduced with the use of water instead of air.
Energy demand of buildings is also shaped by the external environment of buildings and building regulations of a city. The extremity of the micro environment surrounding buildings can determine to some extent the effective working of the system and its efficiency. Some countries like Singapore have stringent laws that limit the amount of external heat gain transmitted into a building. Certification platforms like Green-Mark and LEED have also encouraged the use of technically advanced materials to increase the efficiency of buildings. The use of high quality facades can effectively keep out solar heat and to some extent infiltration if the building is properly sealed; and this could mean a great deal in hot and humid tropical climates, where infiltration can cause a moisture build-up in building interiors.

But what is tropical climate? It is the climate that occurs in Cebu, Philippines; Singapore; Colombo, Sri Lanka; Cairns, Australia; Chittagong, Bangladesh; Mumbai, India; Macapa, Brazil; Jakarta, Indonesia; Rio de Janeiro, Brazil; Naples, Florida; Lagos, Nigeria; Hawaii, USA; Hong Kong, China; Kuala Lumpur, Malaysia; Wuhan, China etc. This shows that a large geographical extent of the world lives in the tropical region. Or in other worlds it can be claimed that a large part of developing Asia lives and experiences tropical climates. That is the reason it is so important to study air conditioning of buildings in this climate. These equatorial climates usually keep day lengths and mean temperatures fairly constant throughout the year. The sun rises daily to a near-vertical position at noon, ensuring a high level of incoming radiant energy in all seasons. The urban geometry and profile of buildings – shape, height, size, orientation, nature of surfaces etc. are in a major way influenced by the urban climate around. This gives rise to complexity of interactions and feedback, which exists between buildings and their outdoor environment.

The rate at which air conditioning is consuming buildings in developing cities is alarming. The energy demand from air conditioning buildings in the tropics comes not only from sensible cooling of air but from latent dehumidification of air, the latter being relatively more energy demanding. It is therefore important to identify a better way to air condition buildings in the tropics, trying to decrease the energy demand for dehumidification or at least trying to decrease the overall energy demand for air conditioning. Taking the example of Singapore that offers a hot and humid climate with annual temperature variation between 23°C to 32°C and relative humidity variation between 82% to 87% [3], it does not make logical sense to use 6°C to 8°C chilled water temperatures to reduce the temperature of air from outdoor conditions to 23°C to 25°C indoor conditions [4] in order to achieve thermal comfort. Therefore the need to evolve from the large conventional Air Handling Unit (AHU) type of an air conditioning concept that is in exacerbate use today to a concept that can be a combination of air-water system or an all water system to perform air conditioning is most urgent. A new scheme of low exergy based air conditioning is discussed in this thesis to achieve the same comfort conditions and air quality but with a focus on the quality of energy used to run the air conditioning process. The system proposed to be used in this study is a combination of radiant panels and decentralized ventilation units.

Singapore, a typical; developed tropical city with increasing energy demands for building air conditioning is chosen as an example during this study. Singapore can be said to have three
types of climates round the year. Geographically located almost at the equator, naturally the climate is hot and humid all year round with everyday rains. The irony is that buildings in Singapore are known to exhibit a winter indoor environment to its occupants. The reason being that air conditioning systems, which employ the psychometric process of overcooling to remove humidity, do not couple the thermodynamic process of reheating after dehumidification due to building regulations in Singapore, which ultimately is responsible for air being blown through indoor room diffusers at 18°C to 19°C. Traditional over-head air distribution systems use a mixing ventilation strategy involving the concept of dilution of air to achieve indoor space thermal comfort and air quality. This type of a system, which usually has ducting, installed above false ceilings demand a higher floor to ceiling height. An under floor air distribution system employing the concept of displacement ventilation, does the same job more efficiently, cooling the indoor space through thermal stratification. The process also requires reduced ceiling to floor height with the elimination of overhead duct work. A low exergy based concept in building cooling design could possibly provide a solution to achieve higher performances. Exergy, which basically focusses on the quality of energy identifies the potential increase in energy efficiency in the energy utilization flow process e.g. in the operation of a boiler. The increase in energy efficiency cannot be quantified in the thermodynamic process but the energy efficiency is close to 100%. In the same process the exergy efficiency is only about 8% which is quite low. This is because the quantity of high quality energy consumed to produce such a low quantity of low quality energy is very high. According to the laws of thermodynamics, the concept of exergy can be more clearly understood as being the maximum work output that can be extracted from an energy flow process caused due to a change in the state of systems as enunciated later in detail through section 1.3.

1.2. Design of Buildings in a Tropical Context

Most of a person’s time is spent inside buildings, shifting from one typology of building to another – offices; homes; shopping arcades; public transport stations; airports; hospitals; hotels; etc., which are all mostly enclosed air conditioned spaces nowadays. Previously the design of buildings had the conventional operable windows for satisfying thermal comfort and indoor air quality requirements, but the transition to a sealed building concept, has brought up the need for providing thermal comfort and indoor air quality through air conditioning and mechanical ventilation systems. This type of a transition can be debated to have brought about the outdoor environment to be irrelevant with respect to our everyday activities of work and play. As an example if we look at the breakup of energy pie in Singapore from Fig 1.1, that shows that energy demand of buildings in Singapore and what are the areas using this energy demand; it can be seen that the building sector is responsible for consuming 31% of the total energy generated in Singapore [5]. Of this, 60% of the energy is used for running air conditioning and ventilation equipment [6]. This data highlights the importance to focus on reducing air conditioning energy demand by giving a sense of the quantum of energy being used to air-condition buildings in a tropical country like Singapore.
The work of an air conditioning system is to provide thermal comfort and acceptable indoor air quality to the indoor occupied space. There are 3 main functions of an HVAC system - Heating, Ventilating and Air-Conditioning, of which only ventilation and air conditioning are used in Singapore or usually in the tropics. Ventilation is the process of changing or replacing air in any space to control temperature or remove any combination of moisture, odours, smoke, heat, dust, airborne bacteria, or carbon dioxide, and to replenish oxygen. Ventilation includes both the exchange of air with the outside as well as circulation of air within the building. Air conditioning on the other hand involves cooling of the air to reduce temperature and dehumidification of air to remove moisture. The conditioning of air in a conventional system takes place in the evaporator or the Air Handling Unit (AHU). Since the water supplied to the cooling coils of the evaporator is at a temperature below the dew point temperature of air passing over the cooling coil, the moisture in air condenses on the evaporator tubes and flows out hence dehumidifying and cooling the air. This water is then collected in an evaporator pan, which is drained out. Such conventional air conditioning systems usually use a centralized air conditioning unit (or package systems) with a combined outdoor condenser evaporator unit and these are often installed in modern residences, offices, and public buildings. An alternative to central systems, which is popular today, is the use of a separate indoor and outdoor coil in split systems. These systems, although most often seen in residential applications, are gaining popularity in smaller commercial buildings owing to their lower capital cost and ease of installation. The evaporator coil is connected to a remote condenser unit using refrigerant piping between an indoor and outdoor unit instead of ducting air directly from the outdoor unit. Indoor units are suspended from ceilings, or fit into the ceiling. A drawback with such systems is that they do not separately provide for ventilation of the indoor space. All these systems are primarily based on or derived from the classical Carrier model of air conditioning that was introduced way back in 1900s. Some questions that rise with the use of such systems are why does it require water to be chilled to 8°C to 10°C (typical AHU supply chilled water temperatures) in order to maintain the room air at 23°C to 25°C? Fig 1.2 shows a representation of the AHU based centralized system.
The AHU primarily acts like an air management unit, trying to usually manage the outdoor air and recirculated air from building indoors. This air mixture is passed over cooling coils having chilled water flowing in them. The chilled water to the AHU is received by separate chilled water supply and return piping from the chiller plant. The air is blown across the cooling coil to undergo cooling and dehumidification. The air that is supplied can also be only outdoor air as in the case of a Dedicated Outdoor Air Supply (DOAS) system. The conditioned air from the AHU is then supplied to different rooms usually through ceiling supply grills or through under floor diffuser. The psychrometric process that takes place in the AHU is sensible cooling and latent dehumidification of air. The air, which is then usually at 15°C to 16°C, undergoes the psychometric process of sensible reheating to attain a temperature of 20°C to 21°C before being delivered to the room as shown in Fig 1.3. In Singapore however, building regulations established by the Building and Construction Authority (BCA) prohibit this thermodynamic process of reheating on the pretext of additional energy demand used for this process. This therefore causes the air to be supplied at relatively lower temperatures, without reheating at 17°C to 18°C as shown in Fig 1.3 causing the overcooling problem in Singapore.
Fig 1.3: The conventional and Singapore process of air conditioning leading to overcooling problems

From Fig 1.3 it can be seen that the outside air supplied by the blower is passed through a filter to remove impurities, dust and pollen. The air is then passed onto a cooling coil where the process of cooling and dehumidification takes place. The air is then passed through a heating coil in a conventional design, where sensible reheating of air takes place before the air is delivered to the room. So the outside air having an air temperature of 30°C and moisture content of 21g/kg is converted to air with air temperature of 20°C and moisture content 12g/kg at the exit of the AHU as shown in Fig 1.3. The conditioning of air takes place using chilled water of 8°C supplied by the chiller. This process ensures the dehumidification process takes and the moisture content in air reduces to 12g/kg, but the absence of reheating coils has air exiting the AHU at 16°C to 18°C. To avoid this problem, the study here proposes a decentralized design, where both processes of cooling and dehumidification that is handled by the AHU in a conventional system are split into 2 systems, each handling one process. The dehumidification or humidity control is achieved by a decentralized ventilation system, which also takes care of ventilation requirements. The cooling or temperature control part is achieved separately by a radiant panel setup. Fig 1.4 shows the decentralized concept used in this study for achieving thermal comfort and IAQ of the indoor space. A high temperature cooling strategy is therefore achieved, where the radiant panels can be supplied with chilled waters at 16°C to 18°C temperatures and the ventilation units can be supplied with chilled water at 8°C to 12°C temperatures. Such a process could also reduce energy demands as they operate with high temperature chilled waters.
Singapore has three regulatory standards or code of practices that are employable on buildings. The three standards are SS 553 (Code of practice for air conditioned and mechanical ventilation in buildings), SS 554 (Indoor air quality for air conditioned buildings) and SS 530 (Energy efficiency standard for building services and equipment). SS553 is the code of practice that provides a general guidance to the design and construction, installation, testing and commissioning, operation and maintenance of air conditioned buildings. The standard also specifies minimum ventilation rates that need to be provided for particular space typologies. SS 530 provides for a code of practice for energy efficiency of building services and equipment. SS554 on the other hand highlights different threshold levels for indoor pollutants in buildings. Tab 1.1 specifies the acceptable limits of thermal comfort and Indoor Air Quality (IAQ) as specified by the standards [7] [8]. These standards along with ASHRAE Std. 55 [4] are used for achieving conformance criteria during this study.
Tab 1.1: Thermal comfort and IAQ threshold limits for Singapore as specified by SS 553[7] and SS 554 [8]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Acceptable Limit (8 hours)</th>
<th>Unit</th>
<th>Measurement method / Analytical method</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Thermal Comfort Parameters</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operative temperature</td>
<td>24 to 26</td>
<td>°C</td>
<td>Air temperature – by hot wire, thermistor, thermometer sling or equivalent method. Globe temperature – by globe thermometer.</td>
</tr>
<tr>
<td>Relative humidity</td>
<td>&lt;65 (for new buildings)</td>
<td>%</td>
<td>By thin film capacitor, hygrometer, thermometer sling or equivalent method</td>
</tr>
<tr>
<td></td>
<td>&lt;70 (for existing buildings) (under peak and common part load conditions)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Air movement</td>
<td>0.10 to 0.30</td>
<td>m/s</td>
<td>By hot wire method for linear air velocity or Kata thermometer for Omni-directional air velocity method or equivalent.</td>
</tr>
<tr>
<td><strong>Chemical Parameters</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>CO&lt;sub&gt;2&lt;/sub&gt;</td>
<td>700 above outdoor</td>
<td>ppm</td>
<td>By real time non dispersive infra-red sensor or equivalent method</td>
</tr>
<tr>
<td>CO</td>
<td>9</td>
<td>ppm</td>
<td>By real time electro chemical sensor or equivalent method (NIOSH manual of analytical methods 6604)</td>
</tr>
<tr>
<td>HCHO</td>
<td>0.1</td>
<td>ppm</td>
<td>By detection tubes, real time electrochemical sensor or equivalent method for screening (ISO 16000-2)</td>
</tr>
<tr>
<td>TVOC that are photo ionisable</td>
<td>3000</td>
<td>ppb</td>
<td>By real time photoionization detector or equivalent method</td>
</tr>
<tr>
<td>Respirable suspended particles</td>
<td>50</td>
<td>μg/m&lt;sup&gt;3&lt;/sup&gt;</td>
<td>By real time optical scattering or piezoelectric monitors or equivalent method</td>
</tr>
<tr>
<td>(aerodynamic diameter less than 10µm sampled with a particle size selective device having a median cut point of 4µm)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Biological Parameters</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total viable bacterial count</td>
<td>500</td>
<td>cfu/m&lt;sup&gt;3&lt;/sup&gt;</td>
<td>By Andersen single stage impactor (N6) or equipment designed for air borne microbial sampling, flow rate at 26.3 L/min for 4 minutes or equal volume of air. Bacteria is cultured by Tryptone Soya Agar (TSA) media and incubated for 48 hours at 35°C.</td>
</tr>
<tr>
<td>Total viable mound count</td>
<td>Up to 500 is acceptable if the species present are</td>
<td>cfu/m&lt;sup&gt;3&lt;/sup&gt;</td>
<td>By Andersen single stage impactor (N6) or equipment designed for air borne microbial sampling, flow rate at 26.3 L/min for 4 minutes</td>
</tr>
</tbody>
</table>
1.3. Exergy Focussed System Design

The exergy concept had its origin with Rant coining the word in the 1950s. It was then used as a tool to optimize thermal power plants [9], but not long ago it has interested engineers to apply the same in the field of sustainable design and building technology [10][11]. Exergy is better defined as a balance between entropy and energy combining the two laws of thermodynamics. It is a better way of defining the potential of the system to produce useful work in a specific environmental condition. Exergy quantifies the net potential of a system as influenced by both quantity of energy available and the temperature (quality) available relative to the system’s surroundings. [11][12][13].

From the second law of thermodynamics; No work can be produced when a system is at thermodynamic equilibrium with its surrounding environment. This is called the state of zero exergy. But as the system changes its state thermodynamically, it can be exploited to extract work with a part of the energy being dissipated or dispersed as waste energy or anergy. This dispersion generates an unsteadiness called entropy or it is called the generation of anergy. To make the working of a chiller more efficient, the exergy needs to be maximized and the anergy needs to be minimized. But this maximization is limited by the Carnot efficiency or the cooling COP operating between the source and sink temperatures given by the equation:

\[
\text{COP}_c = \frac{Q_c}{Q_{h} - Q_c} = \frac{T_c}{T_h - T_c} \quad \text{eq. 1}
\]

Where COP\(_c\) is the cooling COP, \(Q_c\) and \(T_c\) are the energy at the source and source temperature; \(Q_h\) and \(T_h\) are the energy at the sink and sink temperature. The concept of cool exergy is more elaborately described by a research conducted by Shukuya and Hammache [14]. In another research a comparison of decrease in energy and exergy in a thermodynamic process using high quality energy sources is shown; where during the energy generation process the destruction of energy is by 20% but the destruction of exergy is by 80% [15]. This shows that the efficiency of utilizing usable energy is lost to the order of four during the energy generation process itself.

A corollary to this can be seen when it is required to achieve a temperature of 20\(^\circ\)C to 21\(^\circ\)C in temperate climates during winters. Heat needs to be supplied only at a temperature slightly above this temperature and the need to burn fossil fuels at relatively very high temperatures in the order of thousands of degrees to generate a small temperature lift is unfounded. Translating the same to cooling cases in the tropics where a temperature of 24\(^\circ\)C to 25\(^\circ\)C can be more easily achieved by supplying air at a temperature slightly less than 24\(^\circ\)C, does not need for supplying chilled water at 8\(^\circ\)C as the usual practice may be. The practice of
supplying chilled water at temperatures as low as 4°C in also a reality in some district cooling operations. The exergy fraction required for this low quality consumption process of heating and cooling is only 7%. So this may not be the most efficient way for supplying coolness for air conditioning purposes. Producing domestic hot water at 55°C on the other hand requires an exergy fraction of 15% and lighting requires a 100% exergy fraction [15]. So it is seen that a building having these three types of requirements have a fairly large variation in the requirement of the quality of energy supplied. The exergy approach talks about finding ways to supply these requirements with matching energy quality sources. This quality based energy supply could therefore be unstructured and related to use and the need to supply high quality energy to all processes can be avoided. This quality factor is equal to the Carnot efficiency. An extension of this concept was realised in the B35 project where the estimated heat demand was reduced to only 22% of the predicted heat demand far less than the 15 kWh/m2-year benchmark specified by stipulated standards for Switzerland [16]. The B35 project was a house created to be as a laboratory wherein Low-Ex technologies were incorporated in the building for testing [16]. The aim of the low exergy approach is to satisfy energy demand in buildings with increased efficiency of energy processes. This ultimately leads to a potential decrease in the total amount of exergy needed in the demand chain and thus a more channelized and customised distribution of exergy can be implemented based on a consumer specific exergy requirement.

![Thermodynamic energy flow process from the 2nd law of thermodynamics](image)

**Fig 1.5:** Thermodynamic energy flow process from the 2nd law of thermodynamics

A heat engine has a hot reservoir and a cold sink and the flow of energy is from the hot reservoir to the cold sink, thereby producing work. A heat pump or chiller on the other hand
has a reverse flow of energy, from the cold sink to the hot source and this is possible when an external effort is applied as work as shown in Fig 1.5.

Energy (E) as such is a combination of two elements; i.e. exergy (Ex) and anergy (A). In mathematical expression \( E = Ex + A \). A Carnot refrigeration cycle which operates between 2 reservoirs of hot and cold are shown in Fig 1.5. By the 1\(^{st}\) law of thermodynamics we have the energy balance equation \( W = Q_h + Q_c \) where \( Q_h \) is the heat removed from the cold reservoir and \( Q_c \), which takes a negative value is the heat supplied to the hot reservoir. The thermodynamic temperature relationship in the thermodynamic cycle is defined as

\[
\frac{-Q_c}{Q_h} = \frac{T_c}{T_h} \quad \text{eq. 2}
\]

and the Carnot efficiency is defined as

\[
\text{Carnot efficiency} = \frac{W}{Q_c} = \left(1 - \frac{T_h}{T_c}\right) \quad \text{eq. 3}
\]

This expression takes a negative sign expressing that work is done on the system. The maximum work that can be extracted therefore is

\[
W = Q_h + Q_c = Q_c - \left(Q_c \ast \frac{T_h}{T_c}\right) = Q_c \ast \left(1 - \frac{T_h}{T_c}\right) \quad \text{eq. 4}
\]

So in a general sense the exergy of cold can be calculated using the equation

\[
Ex = Q_c \ast \left(1 - \frac{T_h}{T}\right) \quad \text{eq. 5}
\]

Now the heat at very high temperatures can theoretically be converted totally into work i.e. as temperature approaches infinity, the exergy factor becomes unity [17]. But when the temperature \( T_c < T_h \), the quality factor in the brackets becomes negative, and when \( Q_c \) has a negative value due to its directionality in the thermodynamic process the exergy becomes positive due to the mathematical expression explained above. So the exergy factor can be placed as a modulus value represented as

\[
\frac{Ex}{Q_c} = \text{mod} \left(1 - \frac{T_h}{T_c}\right) \quad \text{eq. 6}
\]

1.4. Aims & Objectives of the Research
This section describes the research objectives and aims that this thesis will achieve. The research hypothesis is also detailed through this section.
1.4.1. Research Objectives

The objective of this research is to evaluate and characterize the working of a tropical decentralized ventilation system coupled with a radiant panel cooling system installed at the BubbleZERO laboratory in Singapore. The possibility of achieving air conditioning of buildings by having the radiant panel system take care of a part of the sensible load and the ventilation system take care of the total latent load and remaining of the sensible load is investigated through this research. Problems affecting the operation of such a system like the effect of stored humidity released into room air to influence the thermal characteristics of indoors, occurrence of condensation on radiant panels, proportion of cooling performed through radiation and through convection with varying operational conditions, leakage of air in Under Floor Air Distribution (UFAD) systems, stratification achieved and the effect of working radiant panels on changing stratification etc. are also realised in this study. The positives and negatives of using a ducted network UFAD system in comparison to a plenum supply UFAD air distribution network is researched through collaborative experimentations performed in The Field Environmental Chamber (FEC) at the National University of Singapore (NUS), which houses a plenum supply UFAD system.

Radiant panels and ventilation units installed at the BubbleZERO laboratory are studied for their capacity and effectiveness in removing sensible and latent loads respectively. The amount of sensible cooling performed by the radiant panels through radiation and through convection is quantified for the existing setup at the BubbleZERO and a change in this ratio for an enhanced design setup is also discussed. The importance of radiant panel surface temperature and how it influences cooling capacities is investigated in detail through chapter 4. A heat load of the BubbleZERO laboratory is generated to understand the different heat sources and moisture sources influencing the thermal characteristics of the laboratory. The rise in indoor dry bulb temperature and indoor dew point temperature is studied when the system installed at the BubbleZERO is switched off on weekends or overnight, which could give an idea if condensation occurs on radiant panels during nights due to increase in moisture content of the room. A study on the amount of time required to decrease indoor dew point temperature and dry bulb temperature, when the system is turned on can help determine the air purge out period in the mornings before occupancy commences. An operational matrix for the feasible and successful operation of the decentralized system has been established and corresponding thermal comfort, IAQ and energy demand criteria have been realised. The matrix would help determine the best case operating scenarios for a set point dry bulb temperature without having to fear condensation on radiant panels. Biological contamination issues that could arise in such setups and maintenance procedures for preventing such problems have also been investigated.

An energy analysis of the system in the laboratory is performed to determine the COP and kW/ton during different operating scenarios. The energy demand results of the BubbleZERO laboratory are compared with a business as usual building in Singapore. The increase or decrease in efficiency by using the decentralized system over a conventional system is therefore registered. The feasible use of a CO₂ extraction system, which adsorbs CO₂ from
return air of the room and supplies it back into the room, is scrutinized for its working. The potential of using this type of a ventilation system to clean air and to reduce outdoor ventilation rates are studied. Any trade-offs in energy demand that may be associated with this approach is also presented. The feasibility of maintaining the pollutant levels of other gaseous indoor pollutants like CO; HCHO; TVOCs and particulate matter below their thresholds levels specified by building regulatory standards are perceived when used with a working CO$_2$ adsorption system.

1.4.2. Hypothesis
The hypothesis of this research is that acceptable indoor thermal comfort and indoor air quality can be achieved in the tropics when a radiant panel system performing sensible cooling is operated with a coupled standalone ventilation system performing latent dehumidification. Radiant panels can be operated in the tropics without the fear of condensation when the indoor space is maintained at a positive pressure to keep out humid outside air. This high temperature cooling operation, working on the concept of exergy is able to lower energy demands in comparison to conventional systems in use today. Uncontrolled humidity addition through infiltration or diffusion or internal humidity released from building materials can act as the biggest challenge such systems could face in their successful operation in the tropics. Ventilation rates in buildings could be reduced without compromising indoor air quality requirements specified by popular standards with the use of a novel carbon-di-oxide extraction system.

1.5. Construction of the BubbleZERO Laboratory
The BubbleZERO laboratory represents a work space of 6 m x 5 m x 2.8m. The project was started through collaboration between ETH, Zurich and National Research Foundation (NRF), Singapore in The Future Cities Laboratory (FCL). The project was aimed to look at adaptation techniques and the use of Low-Ex building systems already incorporated in Zurich to achieve thermal comfort and indoor air quality in a tropical climate like Singapore, a process being realized through this doctoral research. Collaboration with the Dept. of Building, NUS was done to use their laboratories, specifically the Field Environmental Chamber (FEC), which has an existing setup of plenum supply under floor air distribution systems. Construction of the BubbleZERO laboratory started in April 2011 in Zurich. Two 20 foot containers were delivered in Zurich; concrete was poured into the ceiling of one container and into the floor of the other container. The half concrete flooring side had embedded ducting in them to represent a concrete embedded duct system. The other half floor of the container was designed to be a raised floor. With one of the ceiling designed to be a concrete ceiling representing concrete ceiling rooms, the other half ceiling was designed to be a false ceiling. So this way one container had a concrete floor and false ceiling and the other container had a concrete ceiling and false flooring. The containers were shipped to Singapore during September 2011. Work on removing one of the longer facades of each of the containers took place and the two containers were joined together to represent the 30m$^2$ laboratory. The joints between the 2 containers was sealed using façade sealants. 2 sets of container doors on each side were removed and were replaced with wood facades
manufactured in Zurich. Each of the facades had one set of M-glass (2 on each side). 2” thick XPS foams were then attached to the outer side of the longer walls of the laboratory and an insulation white cover membrane was fixed to the laboratory. The insulation membrane was a 2 layer material. The space between the 2 layers was supplied with continuously moving air that was pumped using an air pump. The exhaust ports at the top of the membrane exhausted the air being supplied to the membrane. The idea was that moving air would remove any solar heat incident on the laboratory. The glazing used in the laboratory was a special glass called the M glass, which had a U-value of 0.9 W/m²K. This high performance glazing material was developed by a research group in the University of Basel. The high performance glazing kept out incident long wave and short wave solar radiation. Fig 1.6 shows the BubbleZERO laboratory that was setup on the National University of Singapore (NUS) campus with the membrane, wood facades and M glass.

The interior of the laboratory had cooling and dehumidification systems installed in them. One chiller and 2 chilled water tanks to store chilled water were also shipped from Zurich to Singapore. The chiller that was installed in the laboratory had a capacity of about 2kW, much lower than the assumed 5kW capacity that it was designed for in Zurich. A hydraulic system network was then designed in Singapore to connect the chiller, cold water tanks to the 2 radiating panels and 4 ventilation units that formed the total system setup. Four ventilation units were designed in Zurich to be a part of this laboratory to dehumidify outside air and provide for ventilation. These four Dedicated Outdoor Air Supply (DOAS) units were installed in the 4 corners of the laboratory floor as shown in Fig 1.7. Two of the units were installed in the concrete flooring and two of the units were installed in the false flooring. The ventilation units were embedded in a ventilation box pit that was created as a part of the
construction setup. Both ceilings were installed with radiant panel systems. The location of ventilation units, chilled water tanks and radiant panels in the BubbleZERO are shown in Fig 1.7.

The hydraulic circuit was designed in such a way that both tanks were connected in series and the cold water generated by the chiller filled both tanks. A manifold setup was used to connect the four ventilation units and two radiant panels separately to the chilled water tanks. The chilled water from the left tank as shown in Fig 1.7 was supplied to the radiant panel system using small DC pumps and the chilled water from the right tank was supplied to the ventilation units. The radiant panels had a dimension of 4.8m x 1.5m and they had 4 hydraulic loops of water running through them as enunciated in Chapter 4. The ventilation units had 3 cooling coil heat exchangers connected in parallel as shown in Fig 1.8c. The ventilation units supplied outdoor air to the heat exchangers with the help of small DC fans as shown in Fig 1.8c. The air was then supplied into the interior of the laboratory through a network of underfloor and concrete embedded ducts terminating at underfloor diffusers. The interior of the laboratory was made of wood and a lining of plywood was provided on all walls to fix the different components and hydraulic circuits to the façade as shown in Fig 1.8b. The hydraulic circuit that was designed for the BubbleZERO are shown through Fig 1.8a and Fig 1.8b.
Fig 1.8a: Single line diagram of the hydraulic circuit in the BubbleZERO laboratory

Fig 1.8b: Hydraulic circuit connected to the chiller and chilled water tanks

Fig 1.8c: Ventilation units’ setup schematic [19]
Each of the hydraulic loops in the hydraulic circuit had temperature sensors to measure the supply and return chilled water temperature and flow sensors to measure the flow rate of water. The indoor space had SHT wireless sensors to measure the dry bulb temperature and relative humidity in the space. The location of the different sensors is shown in Fig 1.9. The use of radiant panel integrated LED lighting eliminated lighting loads as the panel’s design has a central shaft which exhausted the heat generated by LEDs directly to the outside.

![Sensor locations in the BubbleZERO laboratory](image)

Before the containers, chiller and technologies were shipped to Singapore, some initial tests and experiments were conducted in Zurich to have an idea of the climatic variation the ventilation system was going to experience. The ventilation system for temperate climates was made of a single coil module of heat exchanger as humidity was not the main issue in Zurich. For Singapore the air would have 20g/kg of moisture, so the heat exchanger had to be redesigned. A small tent was setup with a heater and humidifier to achieve tropical conditions in Zurich. The air from the tent was supplied to the modular setup with the single coil module which in turn had a chilled water source supply. Humidity sensors and thermocouples were used to monitor the relative humidity and temperature in the inlet and outlet air streams and the temperature of supply and return water temperatures. This data helped determine the heat exchanger capacity and the design of the first prototype with three heat exchangers unit. In a mixing ventilation system the absolute humidity changes were calculated primarily on mass balance principles which follow the following equation.

\[ V \frac{dg}{dt} = (Q \ast g_s) - (Q \ast g) \quad \text{eq. 7} \]
Where $V \frac{dg}{dt}$ represents the rate of change of absolute humidity, $Q \cdot g_s$ represents the absolute humidity in the supply air for the four decentralised supply units and $Q \cdot g$ represents the absolute humidity in the exhausted air. Getting to know how three heat exchanges were realised, if we neglect infiltration, set supply conditions to 8g/kg absolute humidity with initial conditions assumed to be $28^\circ$C and absolute humidity of 20g/kg, taking each ventilation unit supplying 80 m$^3$/h of air which adds to 320 m$^3$/h of air in total for four units, we see that theoretically we need three heat exchanges with series supply to satisfy this supply air condition.

Initial experiments performed in Singapore had a problem, all ventilation units and radiant panels were not able to operate at the same time due to insufficient capacity of the chiller. So experiments were carried out with only one of the systems, the ventilation units or the radiant panels deemed to be working at one time. The hydraulic circuit shown in Fig 1.8a was also designed by non-professionals and therefore a lot of water leaks in the hydraulic network were experienced from time to time. The water leaks caused the laboratory facade material and the floor insulation foam to become wet and clogged with water for long periods of time. The effect of this problem is seen in the results discussed in chapter 2 where the influence of stored humidity released by building materials adds to humidity control problems in the laboratory. A decision to install a new chiller to enhance the insufficient cooling capacity was taken in January 2013. The design was to allow the old chiller to satisfy the cooling capacity of radiant panels and the new chiller would take care of the cooling capacity of ventilation units. This way the radiant panels and ventilation units could operate at the same time. The chilled water tanks were also separated, to contain chilled water at different temperatures for use in radiant panels and ventilation units respectively. The left side tank from Fig 1.7 stored chilled water for the radiant panels at a higher temperature of 15°C to 19°C. The right side tank from Fig 1.7 stored chilled water for the ventilation units at 8°C to 12°C. A couple of engineers were flown from Switzerland to redesign the new hydraulic circuit for this modified system as shown in Fig 1.10.
Another problem that caused a delay in the setup of this laboratory is the compatibility of hydraulic fittings from Switzerland to the ones available in Singapore. The Swiss fittings that were used in the laboratory followed the European standards in thread pattern and sizes and the fittings available in Singapore followed the British standard or the American standard. So procuring apt hydraulic fittings was not possible and was very difficult in Singapore and a good amount of time was lost in sourcing fittings from Switzerland. The BubbleZERO laboratory was also shifted to a new site on the UWC campus in September 2013. All facades and glazing were uninstalled; the containers were split apart before the move. After the laboratory was installed at the new location, the facades, glazing etc. were installed back. A new chiller was installed to handle the ventilation units’ capacity by December 2013. The facades were also replaced with new ones simultaneously as shown in Fig 1.11.
The BubbleZERO also had radiant panels, that were designed to have a cooling capacity of 50W/m$^2$, but the calculated cooling capacity as shown from Chapter 4, was only 30W/m$^2$. The laboratory was also shifted once and has had 2 major changes in 2.5 years, which caused changes in experimental conditions from time to time. The maintenance and replacement of components in the laboratory was a challenge as it took time to find the particular product in Singapore, and the non-availability of the product forced orders from Switzerland. The data acquisition system was not very stable and regular loss of data was experienced. The sensors were not calibrated regularly and the ventilation units’ chiller was experiencing a persisting pressure loss problem due to a fault in the air side of the system. The BubbleZERO laboratory is a pilot setup that could give us a sense of how these systems and concepts work in the tropical context, but addressing the many challenges learnt in the setup of this laboratory and realizing a more robust and industrial setup can probably give a better sense of the acceptance of this concept in the tropics. The large façade to floor area in the BubbleZERO is also something very unique to this laboratory. The adoption of such a system in a real life building in the tropics and making subjective measurement analysis could bring better answers and validate the acceptance of this system to the people in the tropics.
Chapter 2
Moisture Addition Indoors
This chapter highlights the various sources of indoor moisture pollution and discusses the importance of maintaining indoor humidity balance especially in those spaces using radiant cooling systems. There could be three types of processes through which uncontrolled moisture addition takes place into building interiors; Infiltration, vapour diffusion through building materials and stored humidity released by building materials. While quite some work has been done studying infiltration and vapour diffusion; their impact on influencing indoor humidity balance etc., there hasn’t been significant work done to realise the impact of stored humidity released from building materials on influencing indoor moisture equilibrium. The chapter would enunciate infiltration and vapour diffusion briefly and focus on how stored humidity released by building materials can alter indoor humidity balance. The latter has been published in the HVAC & R research journal.

2.1. Infiltration

The uncontrolled or unintentional introduction of outside air into the interiors of an air conditioned building can be termed as infiltration. This air usually enters through cracks in the building envelope or through gaps in construction joints, door joints or window joints. As shown in Fig 2.1, if air enters into the building it is called infiltration and if air exists the building it is called exfiltration. The main reason for infiltration or exfiltration is the existence of a pressure difference between two adjoining spaces. With respect to Fig 2.1, if \( P_1 > P_2 \) there will be infiltration and if \( P_1 < P_2 \) there will be exfiltration. Infiltration or exfiltration can be a boon or a bane depending on the usage of the interior space. For example in air conditioned hotels, guest rooms are usually maintained at a positive pressure with respect to outside corridors. This helps the room air to exfiltrate into the common corridors to then return through corridor return diffusers. Exfiltration however, can cause harm to adjoining spaces in building typologies such as chemical laboratories or biomedical facilities etc., where it would be beneficial to contain the space air inside the room by negatively pressurizing the room. This would prevent air contamination through leakage. Air tightness of buildings can thus play a crucial role in determining infiltration or exfiltration. Facades need to be air tight and proper sealants need to be used to prevent infiltration. Doors could also be sealed effectively. A dual door concept is also used in the tropics, to make a pressure barrier between the outside and inside. A research done to study the impact on public health originating by the insulation and increase of air-tightness of residential buildings and fine particles in three European countries (Switzerland, Czech Republic and Greece) showed the same duality of the boon and bane of infiltration [1]. The research established that insulation and an increase in air-tightness led to two separate effects: reduction of health effects due to fewer emissions of particle precursor substances to the outdoor air, and increase of health effects due to an accumulation of particles indoors if high indoor particle sources are present [1].
Infiltration, which is more commonly referred to as air leakage in the Heating, Ventilating and Air-conditioning (HVAC) industry is mainly caused due to negative pressurization of buildings. Infiltration can play a major role in jeopardizing the design air change rate of buildings. It can not only cause an imbalance in the thermal characteristics of the space by introducing unaccounted moisture in high humid climates, but can also affect the indoor air quality of a space by introducing dust, particulate matter and other outdoor air pollutants. Infiltration is also known to contribute to a total heat loss of about 10% in cold climates [2]. A popular method used to check infiltration rate is the tracer gas concentration decay method. The concentration decay of a trace gas such as SF$_6$ can be used to measure the infiltration rate due to air leakage by injecting SF$_6$ into the building up to a certain concentration. Then the gradient of concentration decay is a result of air exchange due to leakage when mechanical ventilation is turned off. Studies have shown that the driving force for infiltration is wind pressure and temperature (density) difference between the inside and outside of the building environment [3] and [4]. From the energy point of view, infiltration has known to cause an increase of about 15%–30% of energy use in space heating, cooling and ventilation requirements [5]. But it could be cheaper to install PV panels on the buildings to cover the electricity consumption of chillers or fresh air ventilators and have a positive pressurized indoor space than build up an expensive moisture barrier or moisture recovery system.

The role played by infiltration becomes increasingly important to study in air conditioning buildings of hot and humid climates. Ventilation of buildings in such climates is already an energy intensive process, owing to the cooling and dehumidification of outside air. Addition of humidity and heat into the conditioned space by infiltration could thus only magnify the problem. For buildings using radiant cooling systems to perform sensible cooling of the indoor space, the focus on infiltration becomes paramount as infiltration not only impacts the building cooling load; but changes to indoor humidity levels could inhibit the ability to operate radiant cooling panels without facing condensation problems. Studies done on radiant panels operating in tropical climates also highlight the same view [6]. Feasible operation of radiant panels is only possible when the dew point temperature of the air surrounding the radiant panels is lower than the surface temperature of the panel. So any humidity addition indoors through infiltration could prove disastrous for space cooling. It is advantageous to
have a slight positive pressure in the room or a pre-clause for the feasible operation of radiant cooling systems in tropical environment could be to have a sealed or air tight building, the latter being more difficult to achieve. There have also been some studies done to cash-in on infiltration. Infiltration traditionally was calculated independent of the building envelope performance. However, it has been established that a thermal coupling exists between infiltration and conduction heat transfer of the building envelope. This effect is known as infiltration heat recovery (IHR). Experiments have shown that infiltration heat recovery can typically reduce the infiltration thermal load by 10%–20% [7]. The heat exchange by infiltration air does not have any influence on the temperature profile of the building envelope. However, in reality the infiltrated air, exchanges heat with the solid matrix in the envelope and the temperature distribution in the envelope is different from the case of pure conduction [8]. So infiltration could also influence the surface temperature of building envelopes.

2.2. Vapour Diffusion

Infiltration as a problem is known to most designers today and much has been discussed on ways to prevent infiltration. But moisture diffusion through building materials is increasingly capturing the attention of building operators as it is seen to be the silent killer of buildings. About 90% of damage to buildings, excluding the damage caused by mechanical stress, result from inappropriate levels of temperature and moisture [9]. This is particularly important in buildings using wood or wooden materials, because these materials can become more susceptible to degradation due to fungal attacks during certain conditions [9]. The humidity limit for mould growth on surfaces connected with the ambient air is more than 88–90% for concrete and more than 78–80% for organic materials. Clean EPS and mineral wool can also be infected due to a long-term exposure to humidity exceeding 97–98% [10]. Tropical climates have often seen humidity levels in these limits, so it becomes increasingly important to address the issue of vapour diffusion.

High humidity environments and frequent rains in tropical climates cause moisture to penetrate into porous building materials as shown in Fig 2.2. The building materials absorb water vapour present in the surrounding air according to its equilibrium-sorption-moisture content, which depends on the porosity of the material, the specific surface of the solid matrix, and the pore size distribution of the material [11]. Heat transfer in the micro-capillaries of the porous building materials occur mainly by molecular conduction through the framework of the body and the material bound within the pores (vapour, gas, and liquid) [12]. The moisture accumulated in the pores of the building material is then transferred to the surrounding air by, a convective diffusive transfer of vapour from the surface of the solid to the surroundings [13]. When water vapour permeability of different materials is compared, the permeability per 2.54 cm of material is 0.4-1.6 for PUF insulation, 1.2 for XPS insulation and 118 for glass wool. So insulation material used in construction could also play an important role in catalysing vapour diffusion. The vapour diffusion is characterised by the non-linear diffusion equation:
Where $\theta$ [m$^3$m$^{-3}$] is the volumetric moisture content and $D_0$ is the moisture diffusivity.

![Diagram of building materials](image)

Fig 2.2: Vapour diffusion of moisture through building materials

A great deal of focus has been placed upon energy use and its conservation in the last fifty years. Governments are imposing stricter energy codes on building operators. Efforts to decrease infiltration and ventilation rates have only resulted in more air tight buildings. The consequence of an ill controlled air tight building is the possibility of moisture build-up in building components due to increased humidity levels. This moisture build up can lead to the growth of fungi and mould, which can in turn affect the structural integrity of building components. The effect of changing outside environmental conditions can slowly dampen building materials, thus giving rise to moisture movement from outside to inside. With modern building constructions being subjected to diverse constraints and objectives; having boundaries of achieving a cost effective design; keeping energy efficiency and environmental concerns has become a laborious task and moisture transfer prediction has become very complex. One of the reasons is attributed to the slow movement of moisture through porous materials, which in turn causes the experimental period to be extensively long. Moisture diffusion in building materials (wood being among the most common) was believed to be the least expected source of problems in the building industry. However, over the last few decades, it has been determined that extensive failures in building components are often a result of both thermal and moisture loads [14]. Failures in many building structures were attributed to the lack of knowledge of the effects of both temperature and humidity. A solution that is being used is moisture barriers. Installing stapled sheets of polyethylene plastic on interior walls are being done to stop the flow of moisture. With moisture stopped in its tracks, condensation and resulting water damage could be prevented. Moisture diffusion is usually measured in g/m$^2$-day.

While the dependence of specific heat capacity of building materials on temperature is very low, its dependence on moisture content is always significant because the specific heat capacity of water is four to five times higher, as compared with most building materials in a
dry state [15], [16] and [17]. Several researchers focused on the moisture-buffering properties of different building materials and, in general, on the moisture-buffering effect [18] and [19]. Another form of moisture diffusion into building interiors is from absorbed moisture being released into room air. Due to human activities in the house, such as cooking, washing and showering, the internal relative humidity can rise very quickly in a short time. Some materials that are exposed to these humidity variations are able to absorb moisture when the relative humidity increases and release moisture when the relative humidity decreases. Hygroscopic materials usually have this ability to moderate peaks in the relative humidity of the indoor air. The ability to moderate humidity oscillations depends on the material’s composition and the structure that influences the active thickness, the vapour permeability and the moisture-storage capacity [20], while the water release depends on the temperature, air velocity, water capacity, permeability to water vapour and the time-scale of the processes [21]. The hygroscopic nature of dry wood actually reduces the availability of water to moulds that would normally form on surfaces with a high relative humidity, but only until the wood reaches the fibre saturation point [22]. Several researchers proved that the effectiveness of the buffering capacity of hygro-thermal materials reduces at increased ventilation rates, because the moisture is exhausted by ventilation [23], [24] and [25]. A study shows that humidity present in ambient air and room air affects room temperature and in a hygroscopic region it can alter the room temperature by 2°C to 3°C depending upon the amount and direction of temperature and moisture gradients [26]. Preventing moisture diffusion could therefore increase the life span of the building.

The next section describes the role played by indoor humidity released by building materials in influencing indoor moisture balance. This issue can significantly affect the feasibility of operating radiant panel systems in a high humidity tropical environment.

2.3. Stored Humidity Released from Building Materials

The feasibility of performing high-temperature radiant cooling in tropical buildings when coupled with a decentralized ventilation system

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Abstract
In Singapore’s hot and humid environment air conditioning has moved from being perceived as comfort to now being a necessity. Buildings here usually employ large energy consuming air systems to condition interior spaces. The use of water based high temperature cooling concepts is hard to find due to fears of condensation. The objective of this research is to enunciate the decentralized approach adopted in performing space cooling and dehumidification. The experiment aims at understanding the practicability of using radiant panels with high temperature chilled water for accomplishing sensible cooling coupled with an independent ventilation system for achieving latent dehumidification. The dew point temperature of air surrounding the radiant panels is monitored to check for concerns of condensation. The decentralized ventilation system tested here was effective in reducing absolute humidity in air from 22.3 g/kg in outside air to 12.8 g/kg at the under floor air distribution system’s diffuser outlet. The associated dew point temperature and air temperature measured at the diffuser outlet was 13.9 °C (57 °F) and 19.9 °C (67.8 °F) respectively. The measured dew point temperature of air surrounding the radiant panels was 18.3 °C (64.9 °F), which demonstrates the viability of operating radiant panels using high temperature chilled water to achieve sensible cooling. The Air Exchange Effectiveness (AEE) of the system running at low supply air velocity was determined to be 1.12 which demonstrates a displacement ventilation system. The air change rates measured per hour using tracer gas techniques ranged between 0.9–2.2 when the supply fan speed was varied between 4000–8000 rpm. The amount of outdoor air thus supplied to the designed space ranged between 0.9–2.1 L/s-m² (assuming the system ventilation efficiency as 1). These values are above 0.7 L/s-m², which is the minimum requirement for ventilating office spaces as per Singapore standard.

1. Introduction
Air conditioning systems account for almost one third of energy consumptions in tropical buildings (Ameen, 2005). The general practice of using over-head air distribution systems in the industry to cool building spaces applying the concept of mixing ventilation and convection, usually cools more than the required volume. Air in general has a low density and a low heat capacity when compared to water, and cooling spaces using large volumes of air may not be the most efficient way to condition air. In turn the use of water, which has a heat transfer coefficient around fifty times more than that of air and specific heat capacity four times that of air, could yield better results. In this research we try to explore the feasibility of using water-based systems to condition room air. The challenge of preventing condensation though is paramount to the success of such a concept.

Generic systems couple latent cooling and sensible cooling processes in a single system. Energy can be utilized more efficiently if the latent demand is separated from the sensible demand by the use of a decentralized system. Studies have shown that energy consumptions could be reduced by 42% when a radiant panel system combined with a dedicated outdoor air supply system (DOAS) is used instead of a conventional variable air volume (VAV) system combined with an air side economizer system (Jeong, Mumma & Bahnfleth, 2003). An air
conditioning system applying a similar concept is being studied in Singapore as a part of the research at the Future Cities Laboratory. This system employs a radiant panel arrangement to perform sensible cooling and an independent ventilation unit to perform latent cooling and dehumidification.

2. Background
Singapore can be said to experience two kinds of climates, the naturally occurring hot and humid climate outside buildings and air conditioned winter like environments inside buildings. The reason for such a situation is the psychometric process employed to condition air in Singapore buildings. Over-cooling of air is performed to remove humidity from outside air. This process usually requires a coupled thermodynamic sensible reheating process, which is disallowed by building regulations in Singapore, leading to air being blown into the room at a temperature below 19 °C (66.2 °F).

2.1. Radiant Cooling
Radiant heat transfer processes have been used in the past mainly for heating purposes. Although used effectively in low humidity environments for cooling purposes, this process attracts concerns of condensation when used in humid climates. Mumma (2001) in his study has realized that radiant cooling in humid climates cannot be considered unless a parallel system is in place to decouple the space latent loads. He also discerned that an increase in occupancy by a factor of 2 or 3 to the design conditions would cause condensation, but it would take 90 minutes to 14 hours for the condensation thickness to equal the diameter of a human hair.

The advantages of using such a system are that it can reduce cooling energy demand by 15%–20% (Dieckmann, Roth & Brodrick, 2004). When radiant cooling was combined with desiccant cooling systems, 40% savings in primary energy consumption was achieved in comparison to a conventional constant volume all air system (Zhang & Niu, 2003). Another study by Mumma (2002) showed that the variation of energy balance on the human body is different with and without radiant cooling. He studied that the heat rejection from the human body by radiation is increased from about 35% without radiant cooling to 50% with radiant cooling. Likewise, the heat loss due to convection decreases from about 40% without chilled ceilings to about 30% with radiant cooling. So the net effect being that less heat is rejected by perspiration in the presence of radiant cooling (Mumma, 2002). It was also concluded that the human head, which emits much of the body's heat, can more effectively emit energy with a cooled ceiling above. The result of the cooled ceiling is a cool face and warm feet which in turn increases comfort and alertness (Jones, Hong & McCullough, 1998).

2.2. Decentralized Ventilation
Ventilation models that usually exist are developed around the concept of a centralized air supply system using large air handling units and blowers to eventually distribute conditioned air through bulky ducts running across the building. A decentralized system on the other hand could be an independent system working locally to satisfy latent demands. The type of
system used here was developed at the Building Systems Group at ETH-Zurich (Baldini, 2009). It is a demand controlled ventilation setup that extracts outside air using computer fans at the building’s façade, conditions the air locally and transports the conditioned air directly through short air ducts into the room. The system provides for a variable air-flow rate per unit of up to 100 m$^3$/h (3531 ft$^3$/h). Such a setup would also lead to achieving low ventilation rates of 1-2 air changes per hour (Meggers, Baldini & Pantelic, 2011), which is only possible when the cooling is decoupled from the ventilation.

2.3. Exergy

Exergy quantifies the net potential of a system as influenced by both quantity of energy available and the temperature (quality) available relative to the system’s surroundings (Borel & Favrat, 2010). Buildings normally use large amounts of energy but can potentially consume small amounts of exergy as most demands like cooling and domestic hot water requirements can be satisfied by low temperature energy sources. The exergy fraction required for the low quality energy used for room air heating and cooling is only about 7% (Schmidt, 2009). Also Shukuya (2009) realised in his research that exergy of a volume of air at 25 °C and 50% relative humidity is 400 J/m$^3$, while the exergy of air at 28 °C and relative humidity of 60% is 100 J/m$^3$ concluding that conventional dehumidifying process coupled with convective cooling is intensive in terms of the exergy fraction and that low exergy radiant cooling systems together with natural ventilation may be of importance to pursue. In another study it was realised that annual energy consumptions could reduce by 47.8% if exergy is the benchmark used to design air conditioning systems, they achieved an annual exergetic efficiency increase of 85.2% for their PTGSD system (Alpuche, Heard, Best & Rojas, 2005). In another study using this concept, the B35 project in Switzerland reduced the estimated heat demand by 22% of the predicted heat demand, far less than the 15 kWh/m².yr benchmark specified by stipulated standards for Switzerland (Meggers, Ritter, Goffin, Baetschmann & Leibundgut, 2012). Also another study showed in their exergy demand calculations that for buildings with environmental temperatures around 30 °C and room temperatures around 25 °C, the exergy demand was 58.5 times lesser than the energy demand (Jansen & Woudstra, 2010).

3. Methodology

3.1. BubbleZERO

The BubbleZERO shown in Figure 1 is an experimental laboratory setup in Singapore using two 20-foot high containers to represent a work-space of 6m x 5m x 2.8m. Radiant panel systems and decentralized ventilation units developed in Zurich were installed to provide for the sensible cooling and latent dehumidification requirements respectively.
Facades using M glass having a selectivity function greater than 2 and a U-value of 0.9 W/m²K (Mack, Kamecke, Steiner & Oelhafen, 2009) were used. Chilled water generated from an installed 2.5 kW chiller was stored in two 300L tanks. The tanks in turn were connected to the radiant panel system and the ventilation units through a hydronic network. The chiller was designed primarily to only serve the sensible load requirements of the radiant panel system at a temperature of 18 °C. The decentralized ventilation system, which was added later caused a capacity increase of around 4kW and also demanded 8 °C chilled water for its operation. The absence of enough chiller capacity to support this change prohibited us from running both systems simultaneously. However we could retain experimental conditions for 4 hours. The installation of a new and independent chiller to operate the ventilation units is currently in progress. Simultaneous operation of both systems would then be feasible.

3.2. Dew Point Analysis
The prerequisite for prevention of condensation is to have the air surrounding the radiant panels at a lower dew point temperature when compared to the temperature of chilled water circulating in the panel. In our experimentations the dew point temperature of air surrounding the radiant panel was monitored. The radiant panel though, was not operational during the experiments. The decentralised ventilation system was supplied with 8.5 °C (47.3 °F) water to dehumidify outside air. Figure 2 shows a schematic arrangement of the heat exchangers in the ventilation units for concrete and non-concrete bedding in the laboratory. The ventilation systems in the concrete side were supplied with chilled water flowing from heat exchanger HE3, through HE2 and exiting through heat exchanger HE1 as shown in Figure 2. Reheating of the air stream was performed by the thermal mass of the concrete, which formed the bedding. The ventilation systems in the non-concrete side however, had a different setup to facilitate reheating inside the last heat exchanger HE3. The chilled water circuit is as shown in Figure 2.
Figure 2: Schematic of heat exchangers setup in concrete and non-concrete bedding

The dew point temperature and air temperature were measured using wireless Sensirion SHT type sensors. The sampling locations are shown in Figure 3.

Figure 3: Sensors used and its deployment at the radiant panels

### 3.3. Air Quality Analysis

Tracer gas is the most common technique used to determine the ventilation characteristic in a building. Here we used the basic tracer gas strategy of concentration decay to measure air change rates and evaluate the effectiveness of the designed ventilation system. This method involves releasing of a small amount of tracer gas in the middle of the experimental space to achieve complete mixing and then monitoring the change in concentration with time. A multi–point sampler (Innova 1303) and a photo-acoustic multi gas analyser (Innova 1321) were used to measure the concentration at different locations. A small amount of sulphur hexafluoride ($\text{SF}_6$) used as the tracer gas was released at the centre of the laboratory 1m above the ground. The gas concentration was monitored at the four corners of the laboratory at 1.5m above the ground and also at two points in the exhaust air stream. The sampling points were distributed uniformly inside the space to fully reveal the characteristics of the
ventilation system. Local Mean Age of Air (LMAA), Air Exchange Rate (AER), and Air Exchange Effectiveness (AEE) at each point were determined as indicators of the ventilation characteristics compared to standard mixing ventilation strategy (ASHRAE Std., 1997). LMAA at a point was defined by the average age of all the air molecules arriving at that point. AER is a measure of how many times the air in a space is replaced with outside air in unit time (usually hour). AEE reflects the combined effects of indoor airflow pattern and mechanical recirculation on the age of air at the breathing level.

AER in units of ACH is calculated based on the slope of the tracer gas concentration curve in logarithmic scale as described in eq. 1.

\[
ACH = \frac{[\ln C(0) - \ln C(\tau)]}{\tau} \text{eq. 1}
\]

C(0) is the initial concentration, C(\tau) is the concentration at time = \tau and \tau is the total measurement period in hours.

AEE is calculated based on the test method described by ASHRAE standard 129 by determining the age of air from tracer gas decay tests (LMAA) and nominal time constant (\(\tau_n\)). These three parameters are defined as described by eq. 2 – 4 (ASHRAE Std., 1997).

\[
A_i(LMAA) = \frac{(t_{stop} - t_{start})C_{i,avg}}{c_i(t_{start})} \text{eq. 2}
\]

\[
\tau_n = \frac{\sum_m(Q_{ex,m}A_{ex,m})}{\sum_m Q_{ex,m}} \text{eq. 3}
\]

\[
AEE = \frac{\tau_n}{A_{avg}} \text{eq. 4}
\]

\(A_i\) is the age of air at location i, \(t_{start}\) is the time at the beginning of decay graph measured in hours, \(t_{stop}\) is the time at the end of the decay graph measured in hours, \(C_{i,avg}\) is the average concentration of tracer gas, \(C_i\) is the concentration of tracer gas at location i, \(A_{avg}\) is the average age of air measured at breathing level, \(Q_{ex,m}\) is the rate of airflow in the exhaust air stream measured in cubic meter per hour, \(A_{ex,m}\) is the age of air in the exhaust air stream.

After the tracer gas measurements were completed, a smoke test using a fog machine was performed inside the laboratory to check for infiltration or exfiltration near the floor plenums, and the door and façade junctions.

4. Results & Discussion

4.1. Initial Experimentation
The first experiments that were carried out determined the time taken for condensation to occur on the radiant panels when the panels were supplied with chilled water at different temperatures. Ambient air was supplied through the ventilation units. This process helped
realise the importance of the problem of condensation in Singapore. The results are summarised in Table 1.

Table 1: Time taken for condensation to occur in Singapore (Iyengar, 2012)

<table>
<thead>
<tr>
<th>Chilled water supply temperature in radiant panels °C (°F)</th>
<th>Time taken for condensation to occur (min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>16 (60.8)</td>
<td>15</td>
</tr>
<tr>
<td>18 (64.4)</td>
<td>30</td>
</tr>
<tr>
<td>20 (68)</td>
<td>&gt; 60</td>
</tr>
</tbody>
</table>

A calculation was performed to estimate the cooling energy demand of the whole system for different cooling coil temperatures when the radiant panel was supplied with chilled water at 12.6 °C (54.7 °F). Internal gains of 80 W/person for four occupants occupying 30m² area were considered. The total equipment load was taken as 20 W/m² and outdoor air intake was quantified at 30 m³/hr (1059 ft³/h). The results are summarized in Table 2.

Table 2: Cooling energy demand (Meggers, Baldini, Bruelisauer & Leibundgut, 2012)

<table>
<thead>
<tr>
<th>Cooling coil temp °C (°F) / Dew point temp °C (°F)</th>
<th>Latent load (kW)</th>
<th>Sensible load (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 (46.4) / 10 (50)</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>12 (53.6) / 14 (57.2)</td>
<td>1.6</td>
<td>3.4</td>
</tr>
<tr>
<td>16 (60.8) / 18 (64.4)</td>
<td>1.2</td>
<td>3.8</td>
</tr>
</tbody>
</table>

An estimate of the chiller COP based on 4 operational hours of the ventilation system with 8.5 °C chilled water was calculated. The compressor power consumption extrapolated from the manufacturer’s catalogue data was found to be 0.74 kW at 8.5 °C evaporator temperature. The calculated energy consumed for 4 operating hours is thus 10,656 kJ. The cooling demand on the chiller to reduce the water temperature from 25 °C to 8.5 °C for supplying to the ventilation units for 4 hours is 41,580 kJ. Thus the COP of the ventilation system can be calculated to be approximately 4. The COP of the radiant panel system which would eventually operate at 18 °C would therefore be approximately 8.

4.2. Dew Point Analysis

The ventilation units were supplied with chilled water at 8.5 °C (47.3 °F). The air temperature and dew point temperature of the space were measured at different heights (figure 5). The rate of humidity addition and its causes were realized. The possibility of safe operation of high temperature cooling using radiant panels was inferred by measuring the dew point temperature of air surrounding the radiant panels. The measurements were performed for 4 hrs where a steady state period of 2 hrs was achieved. Insufficient chiller
capacity to supply the required temperature chilled water was responsible for the short measurement period.

Figure 4: Air temperature and dew point temperature at ventilation unit’s inlet and outlet

The ventilation system’s effectiveness to condition outside air is shown in Figure 4. The initial exponential dip in the graph is caused due to system start up. The average reduction in air temperature between the inlet and outlet of the ventilation units is measured to be 8.3 °C (46.9 °F). An average dew point temperature of 14.5 °C (58.1 °F) is achieved at the diffuser outlet during the steady state period. The rise of air temperature and dew point temperature at the diffuser outlet towards the end is due to exhaustion of 8.5 °C (47.3 °F) chilled water. The lowest dew point temperature achieved by the system is 13.9 °C (57 °F) at 19.9 °C (67.8 °F) air temperature. These results show that the ventilation system is effectively conditioning the air. With an average dew point temperature of 14.5 °C (58.1 °F) achieved at the diffuser outlet, the possibility of running radiant panels without fearing condensation is very high if similar conditions are achieved at the panel level. Therefore realization of humidity addition through infiltration or stored humidity release by laboratory materials becomes very important.
Figure 5 shows the variation in air temperature and dew point temperature with varying heights in the laboratory space. Since no human sources added humidity to the space and infiltration being negligible as conformed by the smoke test in section 4.3, the dew point temperature increase could be from stored humidity release from the laboratory materials. This shows that stored humidity release could be a major factor adding to internal humidity of a space in addition to any human sources which would be present in normal practice. Also it can be realised that dew point temperatures do not vary too much from occupant height to the panel height. Inoperability of radiant panels is responsible for ineffective removal of the sensible load and hence the increase in air temperature is justified. Operational radiant panels would have provided for the major part of sensible cooling thus lowering dry bulb temperatures of air.
Figure 6: Air temperature and dew point temperature at radiant panel height

We observe from Figure 6 that the dew point temperature of air surrounding the radiant panel is 18.3 °C (64.9 °F) during the steady state period. Once the system turns on and air is added at 30 m³/h (1059 ft³) at low fan speeds, the existing air in the room, which is at ambient conditions is exhausted and replaced slowly with conditioned air from the ventilation units. Hence the graph shows an exponential dip in dew point temperature for the first 2 hours. The rate of drop in dew point temperature is 2.8 °C/h (37 °F/h). This drop is stalled with the exhaustion of chilled water and the dew point temperature starts to increase towards the end. The absence of a working radiant panel system to remove sensible load is responsible for higher air temperatures in the space. Although the steady state condition achieved here would be sufficient to run the radiant panels, the presence of occupants in real situations could cause a further increase in dew point temperatures making high temperature cooling unfeasible. However, the availability of sufficient chiller capacity, for longer periods of time would possibly exhaust all stored humidity releases causing a further decrease in dew point temperatures in the present situation. This could then compensate the increase in dew point temperatures caused by the presence of occupants.
Variation in absolute humidity across various heights in the experimental space is shown in Figure 7. The absolute humidity drops from 22.3 g/kg in outside air to 12.8 g/kg at the diffuser outlet. The absolute humidity at the panel level is 16.2 g/kg averaged in the steady state period. The addition of humidity during the steady state period is averaged at 2.2 g/m³h (0.00013 lb/ft³h). The wood used for the façade was a type used in concrete formwork and thus has very low vapour diffusivity, and the metal façade of containers would not allow vapour diffusion. For this reason we can assume the diffusion of humidity should be relatively low, and nowhere near the magnitude of 2.2 g/m³h (0.00013 lb/ft³h). With infiltration being very low as seen in section 4.3, the source of humidity addition could be through the stored humidity in the materials and equipment which make up the laboratory. This shows that stored humidity release could play a role in increasing dew point temperatures of a space. This factor becomes more important to consider when designing buildings using radiant cooling processes in high humidity environments.

### 4.3. Air Quality Analysis

Tracer gas decay tests have been conducted at different operating conditions of the decentralized air supply units. Since the volume of the laboratory is small (around 60 m³/2472 ft³), uniform mixing of the tracer gas was achieved after several minutes. The air change rates measured by tracer gas techniques for different fan speeds (4000-8000 rpm) of decentralized units ranged between 0.9 – 2.2 ACH. By assuming the system ventilation efficiency as 1, the amount of outdoor air supplied in the designed space ranged between 0.9 – 2.1 L/s-m², which is above the limit for an office building space of 0.7 L/s-m² in Singapore (Singapore Std., 2009). The decimal accuracy of AEE is justified based on the accuracy of
the measuring device and uncertainty caused in this parameter. The calculated age of air and air exchange effectiveness in different points at breathing level is summarized in Table 3:

Table 3: Age of air and Air Exchange Effectiveness at breathing level at a fan speed of 4000 rpm and 8000 rpm (Saber, Meggers & Iyengar, 2013)

<table>
<thead>
<tr>
<th>Sampling Point</th>
<th>$A_i$ (s)</th>
<th>$A_{ex}$ (s)</th>
<th>$A_{avg}$ (s)</th>
<th>$\tau_n$ (s)</th>
<th>AEE</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>4000</td>
<td>8000</td>
<td>4000</td>
<td>8000</td>
<td></td>
</tr>
<tr>
<td>Occupant 1</td>
<td>3970</td>
<td>1690</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Occupant 2</td>
<td>3569</td>
<td>1604</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Occupant 3</td>
<td>3914</td>
<td>1630</td>
<td>-</td>
<td>-</td>
<td>3818</td>
</tr>
<tr>
<td>Occupant 4</td>
<td>×</td>
<td>1817</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust 1</td>
<td>×</td>
<td>-</td>
<td>1528</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust 2</td>
<td>-</td>
<td>-</td>
<td>4289</td>
<td>1834</td>
<td>-</td>
</tr>
</tbody>
</table>

× Missing data

It can be seen from Table 3 that units running at lower fan speeds (lower supply air velocity) have the effectiveness of the system fall under the displacement ventilation category (AEE = 1.12). However, the units working at full capacity (8000 rpm) would have the system effectiveness to be around 1 which is equal to an overhead system with mixing ventilation. The ventilation effectiveness of low velocity (0.8 m/s supply jet at 1.4 m height) floor supplies having ceiling return nomenclatures provide a thermal stratification of 1.2 (ASHRAE Std., 2007). The supply air velocity from floor diffusers, even at maximum capacity of the decentralized units, is lower than ASHRAE thresholds and would be categorized in the low velocity floor supply group. Low efficiencies of the system at 8000 rpm fan capacity show signs of mixing flow patterns and also exfiltration due to positive air pressure indoors. The exfiltration takes place through door gaps or other joints in the laboratory facades as seen by the smoke test in Figure 8. The smoke test has also shown that there are some leakages near the door and two wall connections which have a negative effective on the effectiveness of the ventilation system. So it can be concluded that very negligible infiltration takes place and most of the humidity addition as examined in Figure 7 is through emission of stored humidity from the materials and equipment of the laboratory.

Figure 8: Smoke test experimentation
5. Conclusions
From our results we realize that water based high temperature radiant cooling methods could be a feasible option for conditioning air in tropical buildings when the dew point temperature of air surrounding the panels is controlled to be lower than the temperature of chilled water flowing through the panels. This control in our case was achieved using decentralized ventilation units which dehumidify outdoor air. The ventilation system tested here was effective in reducing absolute humidity of the incoming air by 42.6% from outside conditions to diffuser outlet conditions. Although air at the diffuser outlet had a dew point temperature of 13.9 °C, which was enough for performing radiant cooling, the dew point temperature of the air surrounding the radiant panels was 18.3 °C, an increase of 31.6% between the floor and panel surface caused primarily by stored humidity releases. With infiltration being negligible, humidity addition through stored humidity release by building materials becomes a crucial factor to be considered while designing spaces using radiant cooling systems, especially during start up phases. The presence of human subjects and its effect on the humidity balance in the experimental space will be tested in the next phase of the experimentation. The availability of sufficient chiller capacity to run the ventilation units for a longer period of time can help exhaust all stored humidity releases. This can cause a further decrease in dew point temperatures which will help to operate radiant panels to provide sensible cooling, thus lowering dry bulb temperatures too.

Nomenclature
AER = Air Exchange Rate, h⁻¹
ACH = Air Changes per Hour, h⁻¹
C(0) = Concentration at time=0, kg/m³
C(τ) = Concentration at time=τ, kg/m³
τ = Total measurement period, h
AEE = Air Exchange Effectiveness
A = Age of air
τn = Nominal time constant
Ai = Age of air at location i
Aex,m = Age of air in exhaust airstream m
Aavg = Average age of air measured at breathing level
tstart = Beginning of decay graph, h
tstop = Final tracer gas measurements, h
Qex,m = Rate of airflow in exhaust airstream, m³/h

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References


### 2.4. Takeaway

Infiltration can affect thermal characteristics of building interiors. Air tightness of buildings is paramount to prevent infiltration. Addition of moisture from building exteriors through infiltration can increase dew point temperature of air surrounding radiant panels, thus giving
a chance to condensation and preventing radiant panel operation. Positive pressurization of the indoor space is a solution for preventing infiltration as making a building air tight is a tough ask. Vapour diffusion also affects the lifespan of a building. Diffusion of moisture through building material can cause the growth of moulds and fungi over a long period of time and can thus affect indoor moisture balances and indoor air quality. Materials with low permeability and moisture diffusivity coefficient should therefore be used to inhibit vapour diffusion. Alternatively moisture barriers can be used to prevent vapour diffusion through building materials. Experiments show that a radiant panel system’s operation is feasible in the tropics only when it is coupled with another system that dehumidifies outdoor air. If the dew point temperature of air surrounding the radiant panels is maintained below the surface temperature of the radiant panels, there will be no condensation. Experiments also show how a constant amount of humidity could be added by building materials to room air, thus giving a vertical humidity gradient in the space. This stored humidity released from building materials could be exhausted over a period of time; nonetheless it is an important factor to consider during radiant panel operations. Another possibility of mitigating this stored humidity released from buildings could be by employing a combined dehumidification system along with the radiant panel at the ceiling level. This could dehumidify the air at the panel level and avoid condensation. The limits of operating radiant panels in the tropics are discussed in further detail through chapter 4 and chapter 5.
Chapter 3
Under Floor Air Distribution
This chapter discusses the concept of displacement ventilation along with the benefits and problems associated with the practice of this concept. A design approach using this concept, the Under Floor Air Distribution (UFAD) system is discussed in detail through this chapter. Thermal characteristics and stratification of air in ducted UFAD systems are discussed in detail based on experimental findings. A research work published in the International Journal of Ventilation enunciates the leakage characteristics study between plenum supply UFAD systems, ducted supply UFAD systems and Over Head Air Distribution (OHAD) systems, which is presented through this chapter.

3.1. Displacement Ventilation (DV) & UFAD Systems

Conventional Heating, Ventilation and Air-Conditioning (HVAC) systems supply high velocity air through ceiling diffusers into indoor spaces. Thermal comfort of indoors is achieved through mixing of room air. Displacement ventilation however, is a concept wherein air is introduced into the room at the floor level rather than at the ceiling level. The air is supplied at relatively lower velocities and at a temperature that is only a few degrees below the temperature experienced by indoor occupants. The heart of displacement ventilation theory is the simple physics concept that air becomes less dense and rises when it heats up. Cool air supplied at the floor level is warmed by thermal plumes created by occupants, electrical equipment etc. and ascends upwards due to buoyancy driven thermal lift in the room. The vertical temperature gradient that is established gives rise to a stratified volume of air. Stratification of air is one of the most important defining elements in a displacement ventilation process as it not only helps to achieve better localized thermal comfort, but also provides for improved ventilation effectiveness, which is an indicator of indoor air quality [1]. Many research works have reported the advantages of displacement ventilation, theoretically and experimentally and in different applications [2], [3] & [4]. Lee in his research explained how different stratification distributions resulted in distinct flow rates of air and ventilation efficiencies in a space [3]. Higher ventilation effectiveness caused by stratification of air provides for more polluted air being able to occupy upper parts of the room, giving occupants less polluted air at the breathing zone. Displacement ventilation systems can take care of sensible and latent loads of a space when used as a standalone system. When such systems are coupled with radiant cooling systems, which handles the bulk of sensible loads they can take care of latent loads and a part of the sensible load. Typical displacement ventilation strategies can include installing of wall integrated diffusers or floor integrated diffusers. The latter has coined a word for itself called Under Floor Air Distribution (UFAD) systems.

UFAD systems have been used extensively in the past for air-conditioning data centres, where data centre racks are cooled using such processes. Communication equipment, data servers and other information technology equipment consume electrical power and generate heat. The heat produced by this equipment can have large effects on the performance, reliability and useful life of the equipment. In particular, rack-mounted equipment housed within enclosures is particularly vulnerable to heat build-up and hot spots. The amount of heat generated by a rack is dependent on the amount of electrical power drawn by the
equipment in the rack. So the amount of heat a given rack or enclosure can generate, can vary considerably from a few tens of watts up to about 10,000 watts or beyond [5]. The high heat output produced from data centre units generates a large thermal plume that can be exhausted easily with upward buoyant flows of air from UFAD systems. This concept has off late interested building designers to use the same for achieving occupant comfort.

The popularity of using UFAD systems over wall integrated units for achieving displacement ventilation in occupant space is due to the flexibility it offers in allowing to change diffuser locations with changing furniture layouts in buildings. Diffusers supplying conditioned air to occupant zones are usually integrated with raised floors and the floor tiles can be easily rearranged based on localized needs. The raised floor, which creates a plenum, below itself, is supplied with conditioned air either through pressurization of the plenum space to create a plenum supply UFAD system or through ducts to create a ducted supply UFAD system. However, plenum supply UFAD systems can face some problems. Convective heat gains in the plenum tend to rise air temperatures in the plenum as it passes from the plenum supply location to the diffuser outlet and this has been reported to be one of the problems experienced by plenum supply UFAD systems, altering the thermal characteristics of air [6]. Plenum air temperature rise is caused by cool supply air coming in contact with relatively warmer raised floors. Another problem that may occur in plenum supply UFAD systems is that the pressure at different diffuser locations in the plenum may not be constant and may vary with distance from air supply location in the plenum. This provides for a possibility that the farthest diffuser may have lesser pressure and velocity of air compared to the nearest diffuser. Cracks in false flooring and tile joints could cause category 2 leakage of air into the occupied zone. More on the leakage of air has been explained in section 3.3. A classical plenum supply UFAD arrangement has been shown in Fig 3.1. A ducted supply UFAD system however can reduce the problem of leakage as explained in section 3.3. The use of such systems is gaining popularity with the option of integrating the ducted system into concrete floors hence eliminating the need for an underfloor plenum, and thus reducing floor to floor heights.

Fig 3.1: Plenum supply UFAD arrangement
On the energy demand side, claims of UFAD systems reducing energy demand by 30% compared to conventional systems has been reported [7]. A study showed that, when compared to overhead systems, the integrated UFAD cum radiant cooling system showed a 22% to 23% reduction in total energy consumption during peak cooling months and a 31% reduction in peak hourly electricity demand [8]. When compared to the UFAD systems of similar types not using radiant cooling, these reductions were 21% to 22% and 24% respectively [8]. Rijksen in his work presented an optimal HVAC design for a call centre using a UFAD system with passive chilled beams, which helped to reduce electrical energy consumption by 41% [9]. This system also consumed 41% less operational energy cost when compared to a traditional building using an overhead VAV system for air conditioning [9]. These energy savings potential act as a welcome sign to designers, with some industry-watchers predicting that as many as 35% of future office buildings will include UFAD systems [10]. But there are some financial hurdles that could deter the large scale use of such systems. Although the chiller and air system are smaller for the UFAD cum radiant cooling case when compared to only UFAD case (19% and 23% smaller respectively), the added cost of the hydronic tubing and heat exchangers would potentially overwhelm these cost reductions [8]. Also, the added operational (and design) complexity of such a system cannot be overlooked. It also appeared in the research done by Raftery that the energy savings associated with the UFAD cum radiant system may not offset the additional initial cost of such a system in today’s economic environment [8].

The next section looks into different ways in which heat gain from facades or glazing affect the thermal characteristics of indoor perimeter areas and how the use of radiant panels could mitigate this influence of perimeter heat gain.

3.2. Perimeter Heat Gain Affecting Thermal Characteristics of UFAD Systems

The thermal characteristics of UFAD systems have been researched quite well. Achieving a stratified volume of air and a vertical temperature gradient is paramount for the working of this concept. ASHRAE Standard 55-2013 recommends that the maximum temperature gradient between the supply location and return location of air in UFAD systems should not exceed 3°C [11]. McQuiston suggested that an ideal condition would be to have a uniform room temperature from the floor to about 1.8 m above the floor [12]. However, a gradient of 2°C should be acceptable to about 85% of the occupants to remain thermally comfortable [12]. When dissatisfaction as a function of vertical air temperature difference between head and ankles is compared, it has been observed that the percentage dissatisfied varies exponentially, with 2% dissatisfied at 2°C and rising to 60% dissatisfied at 8°C [13]. Another area in UFAD systems that has been debated quite well in the past is the achievement of thermal comfort at low air velocities. Zhang in his research showed that when compared to conventional systems, the proper design, installation, maintenance and operation of displacement ventilation systems can still maintain a thermally comfortable environment, even when such systems achieved air velocities lesser than 0.2 m/s [14]. The research also
claimed that the percentage of dissatisfied people were 10% [14]. Occurrence of cold feet is another problem UFAD designers fear. Good designs suggest that the supply air temperature at diffusers should be above 18°C to avoid occupant cold feet, unless the occupant is designed to be seated 1 m away from the diffuser location [15]. Of late perimeter heat gain from glazing and facades is a factor that has caught the attention of designers as it could influence stratification of indoor air in perimeter areas. The occurrence of these phenomena could affect occupant’s thermal comfort in the perimeter zone of the building. A study is conducted to realise the impact of perimeter heat gain on temperature gradient of ducted UFAD systems through this research.

The BubbleZERO laboratory was used for studying the impact of perimeter heat gain on stratification. The construction and technical details of the laboratory is explained in the Research Methodology section of the journal paper discussed in section 3.3. During the experiments, the ventilation units were supplied with 12.1°C average chilled water temperature. The return chilled water temperature was on average 15.3°C. The ventilation rate was 11.1 L/s. There were no occupants in the room and all lights were switched off to prevent heat from lighting equipment to influence results. The outside conditions were normalised. Two measurement pods were distributed in the chamber for simultaneous measurements. One pod was placed next to the glazing to realise the influence of perimeter heat gain. The second pod was kept in an interior location (Spatial location 2) as shown in Fig 6 of section 3.3. This pod had no influence of perimeter heat gain. Three measurement locations, namely at 0.1m from the ground determining the diffuser level, 1.5m from the ground determining the human level and 2.1m from the ground determining the panel level respectively were identified for the measurements. The locations of all pods were selected in a way to avoid the effects from the supply diffusers’ throw. Dry bulb temperatures were measured at the three locations using SHT Sensirion sensor as shown in Fig 3 of the HVAC & R Journal paper discussed in Chapter 2. A continuous measurement setup from 1000 hours to 1800 hours was established for all experimental setups.

Fig 3.2a: Stratification achieved at perimeter area with panels turned off
Fig 3.2b: Stratification achieved at inside area with panels turned off

Fig 3.2a and Fig 3.2b gives a summary of the stratification achieved by the ducted UFAD system in perimeter area and inside area when panels are turned off. The panels were turned off, to prevent any influence of radiant heat transfer on dry bulb temperature. The results show that the ducted UFAD system achieves an average stratification of 3.4°C between the diffuser level and panel level when there is no influence of perimeter heat gain. With influence of perimeter heat gain, the stratification becomes 3.7°C. The average dry bulb temperature at the human level inside the laboratory was 23°C. The average dry bulb temperature at the human level increased to 23.4°C in perimeter areas due to perimeter heat gain from glazing.

Fig 3.3a: Stratification achieved at human level in the perimeter area and inside area with panels turned off
Fig 3.3b: Stratification achieved at panel level in the perimeter area and inside area with panels turned off

Fig 3.3a and Fig 3.3b shows the stratification achieved at the human level and panel level in perimeter area and inside area when panels are turned off. The results show that there is a 0.4°C increase in average dry bulb temperature at the human level due to the influence of perimeter heat gain. There is also a 0.5°C increase in average dry bulb temperature at the panel level due to perimeter heat gain.

Fig 3.4a: Stratification achieved at human level in the perimeter area with panels turned off and when panels are supplied with 16°C chilled water
Fig 3.4a and Fig 3.4b shows the stratification achieved at the human level in perimeter area and inside area when panels are turned off and when panels are supplied with 16°C chilled water. The average surface temperature of the panels is 19°C and the panel return temperature is 18.6°C. The results show that the average dry bulb temperature at the human level reduces from 23.4°C to 23.1°C in the perimeter area and from 23.1°C to 22.5°C in the inside area when radiant panels are operated with 16°C chilled water.

From the results we can conclude that operating radiant panels with supply water temperature of 16°C will influence the temperature gradient of air in UFAD systems. Operational radiant panels act as a counter to perimeter heat gain by decreasing the dry bulb temperature at the human level, hence aiding thermal comfort. The average dry bulb temperature at spatial location 2, changes from 19.8°C at the diffuser, to 22.5°C at the human level, to 22.5°C at the panel level when the radiant panels are working. The same set of temperatures for non-operational panels are 19.8°C, 23.1°C and 23.2°C. The stratification values for perimeter zone are 19.9°C at the diffuser, 23.1°C at the human level and 23.1°C at the panel level with operating panels and 19.9°C, 23.4°C and 23.6°C with non-operational panels.

Air leakage can be a major issue that affects UFAD systems. Leakage from interactions to the outdoor environment or through undecided indoor locations can alter indoor temperatures and moisture content of the room, thus affecting especially the working of radiant panel systems as discussed in Chapter 2. Leakage characteristics and different types of air leakage are discussed in the next section.
3.3. Leakage Characteristics

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A comparative study of leakage characteristics between an Under Floor Air Distribution system and an Over Head Air Distribution system

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Abstract
This research aims at quantifying the leakage occurring in Under Floor Air Distribution (UFAD) systems and Over Head Air Distribution (OHAD) systems. The study also classifies the leakage occurring in the different systems into Category 1 leakage and Category 2 leakage. The study is performed between a plenum supply UFAD system (having a maximum flow rate of 1272 m³/h), ducted supply UFAD system (having a maximum flow rate of 286 m³/h) and an OHAD system (having a maximum flow rate of 2004 m³/h). Results show that in plenum supply UFAD systems, 5.9% and 14.8% of the volume of air supplied by the Air Handling Unit (AHU) is attributed to leakage when the AHU is running at 50% and 100% fan speeds respectively. In ducted supply UFAD systems however, the leakage drops to 2.1% and 5.6% respectively at 50% and 100% AHU fan speeds. A ducted supply UFAD system in comparison to a plenum supply UFAD system lowers leakage by 62.2% when the AHU is operating at 100% fan speed and by 64.4% when the AHU is operating at 50% fan speed.

Keywords
Leakage Characteristics, Under Floor Air Distribution, Over Head Air Distribution, Plenum Supply, Ducted Supply
1. Introduction
It is a well-established fact that the built environment contributes significantly to the global energy consumption, green-house gas emissions and consequently to the climate change phenomenon. In a typical breakdown of sector based energy utilization in most industrialized countries around the world, buildings may attribute to 30-40% of the total annual energy consumption, of which up to 50% could be used for Heating, Ventilating and Air-Conditioning (HVAC) requirements (Mahmud et al. 2010). In view of the increased demand for cooling, caused by climate change effects, it becomes imperative that energy efficient HVAC technologies and air distribution strategies are explored.

In the context of comfort air-conditioning, buildings are designed for human occupancy. While thermal comfort requirements are to be met, it is also equally important to ensure that adequate ventilation is provided at all times to ensure acceptable Indoor Air Quality (IAQ). Under Floor Air Distribution (UFAD) systems, by virtue of its design has the advantage of moving air in the same direction as the buoyancy driven thermal lift in the room. The upward air flow pattern, vertical temperature gradient and warmer supply air temperature are the most important characteristics that differentiate UFAD systems from conventional Over Head Air Distribution (OHAD) systems. UFAD systems are becoming increasingly popular owing to their expectations of better performance in terms of IAQ and energy consumption in comparison to OHAD systems. Additionally, it can offer better flexibility to changes in office layouts with the ease of relocating diffuser outlets to suit workstation locations (Loudermilk 1999). But UFAD systems could face problems of air leakage if not designed correctly. Leakage of air can be caused either by infiltration/exfiltration to the outside and is called category 1 leakage or from floor/tile joints on the inside, which is called category 2 leakage.

This research reports the outcome of an experimental study that was carried out at the National University of Singapore (NUS) to investigate the leakage characteristics of pressurized plenum UFAD system and an OHAD system, both of which are installed in the same research facility at NUS and a ducted UFAD system installed in another nearby research facility at the Bubble ZERO laboratory.

2. Background
Indoor spaces using UFAD systems usually employ raised floors in their design. The underfloor plenum that is created due to the use of raised floors usually acts as a passage to deliver conditioned air into the occupied zone of the building. Underfloor plenums provide the ease of access to lay and configure cables and power networks, so it gives designers an alternative to overhead spaces (usually created above false ceilings). This plenum space has been exploited by HVAC designers to supply conditioned air into the building interiors. There are two methods for UFAD plenum design, a pressurized plenum (using passive supply diffusers) or a ducted air supply network below the raised floor, which is basically an inverse version of a conventional OHAD system.
2.1 Pressurised Plenums
The Air Handling Unit (AHU) used for pressurizing plenums in UFAD systems maintains a small positive pressure in the under-floor plenum relative to the conditioned space. A pressurized plenum is one of the most common options among all UFAD configurations mainly due to the prospect of using passive diffusers, which helps to reduce energy demand required from fans. Studies have shown that plenums with as little as 0.178 m of height can still deliver air uniformly as far as 24m from the air supply location in the plenum (Bauman et al. 1999).

Whilst one of the benefits of having a UFAD system is its flexibility of having floor panels being able to be removed regularly during space reconfigurations, improper installation or repeated removal and installation of these floor panels can cause gaps in the joints between raised floor tiles. These gaps can lead to air leakage from the plenum to the room interior. Care should also be taken on the positioning of any plenum partitions and diffuser locations and the ease of changing the location of these partitions and diffusers with changes in space configurations. Another issue that could affect plenum designs is that the pressure of air exiting the farthest diffuser from air supply location in the plenum should not be lower than the pressure of air exiting the nearest diffuser in a significant way. This could affect thermal characteristics of the space as the velocity of air could potentially drop.

2.2 Ducted Network in Plenums
Using ductwork to distribute supply air, particularly to the farthest diffuser, could be an answer to the problem of low air pressures at the farthest diffuser outlet. This type of a ducted setup in the plenum would however, resemble a conventional overhead air distribution system concept. Such a ducted system could potentially help in ensuring better temperature and pressure control of air at the diffuser outlet. However, this will potentially lead to an increase in plenum height due to the use of such ducts in underfloor plenums.

Daly suggested an alternative to avoid ductworks in plenums by using multiple vertical shafts serving multiple plenums underneath the floor (Allan Daly 2002). Designing such shafts close to staircases or integrating it with columns or other permanent features of a building could minimize any architectural inflexibility that may be caused by using such a concept. Alternatively, if the ducting can be integrated into hollow concrete slabs (created by using innovative techniques of concrete construction using of formers or synthetic cobiix balls), it can potentially eliminate the use of underfloor plenums, thus decreasing floor to floor heights of buildings. A pilot project using such a concrete integrated ducting has been used in the BubbleZERO laboratory (Bruelisauer et al. 2013).

2.3. Leakage Issues with UFAD Systems
The usual complaint from occupants in air conditioned spaces is the perception of the space being too hot or too cold. Research reveals that thermal comfort is a personal preference, which expresses satisfaction with the thermal environment and it differs from one individual to another individual (Bauman et al. 1996). The mixing ventilation strategy used in a conventional OHAD system design prevents it from providing such a personalised control on
the thermal characteristics. A UFAD system, however, by virtue of individual diffusers provided at individual work spaces, could provide a significant amount of control on air volume, which translates to a control on air temperature and air velocity. But leakage issues associated with UFAD systems, especially the existence of category 2 leakage will prove detrimental to the control of air volume and in turn on the control of local air temperature and air velocity. Air leaking through floor joints of a raised floor in a plenum supply UFAD system could lead to a decrease in the desired volume of air exiting the diffusers, hence reducing diffuser level air pressure and in turn affecting local thermal characteristics. A study done by Emmerich also highlighted that air leakage issues could increase cooling energy costs by 33% (Emmerich et al. 2005).

2.4 Types of Leakage
Air leakage is particularly prevalent in pressurized plenum designs, which could impair the system performance if it is not controlled (Allan Daly 2002). Air leakage is one of the most important factors to be considered while designing UFAD and OHAD systems as excessive leakage into the occupied zone through floor cracks and joints as category 2 leakages can lead to a loss of pressure at the diffusers. Exfiltration or Infiltration through door and window joints can cause category 1 leakage, which can affect the air temperature of the interior space. The pressure loss would require AHU fans to supply more air, which could lead to energy being wasted. Hence, proper sealing of floor plenum joints are particularly important in UFAD systems. Although the leakages may be unavoidable, they should be taken into consideration during the system cooling load design. There has been significant work done previously on leakage issues in buildings. Literature shows that mechanical system leakage can be quite significant in influencing the overall leakage of a building. A study by Bohac in evaluating leakage characteristics of a particular set of buildings in United States built after 2000 showed that 25% of the average leakage in buildings was due to construction imperfections of the mechanical systems (Bohac et al. 2014). Another study by Harrington showed that average duct leakage can be as high as 28% of the volume of air supplied by the AHU, and effective sealing using aerosols can bring down this leakage by almost 90% (Harrington et al. 2014). This shows the importance of sealing joints to avoid leakage. It is therefore important to have the correct and quality workmanship to enhance the air tightness of the space and prevent leakage in a building. A study by Laverge showed that air leakage can vary from around 1800 m³/h to 3800 m³/h @ 50Pa pressure difference between indoors and outdoors due to construction techniques adopted by different work forces (Laverge et al. 2014). So it is important that good workmanship is accompanied with quality materials and sealing options to minimize leakage. But there are few experimental studies done on leakage characteristics of actual UFAD systems or mock-up experimental chambers and this research highlights some of the concerns that leakage can cause in such mechanical systems.

2.4.1 Construction Quality Leakage
Air tight building envelopes are crucial to the performance of any HVAC system. As shown in Figure 1, poor construction of exterior walls, columns, stairwells etc. will result in air passing through these wall cavities and floor penetrations, possibly triggering short-circuits to
the return plenum, to the outside of the building or even to the floor below (Webster et al. 2008).

![Diagram of air leakage through poor construction quality](image)

**Figure 1: Category 1 plenum UFAD system air leakage through poor construction quality (Webster et al. 2008)**

This type of leakage is called as category 1 leakage and it is detrimental to the performance of the system as air is moving outside the designed zone. This could result in an increase in fan power and cooling load. This type of leakage is usually known as infiltration/exfiltration. A popular perception on air tightness of buildings is that older buildings are less air tight due to their age. But in a study performed by Persily, to determine the correlation between air leakage and building age showed that there is no correlation between air leakage and building age and as long as a building achieved at least a maximum of 10% category 1 air leakage, an older building could still be reconfigured with an UFAD system (Persily 1999). So UFAD systems can also be used as a retrofit air conditioning option in older buildings as long as category 1 leakage is a non-issue.

### 2.4.2 Floor Leakage

Air can penetrate into the occupied zone through several openings in the floor plenum other than diffusers. This type of leakage, which is also known as category 2 leakage as shown in Figure 2 could arise from floor and carpet joints, electrical floor boxes, etc. (Webster et al. 2008).
Figure 2: Category 2 plenum UFAD system air leakages from floor joints (Webster et al. 2008)

The air leaking into the occupied zone through floor joints may result in reduced pressure supplied to individual diffusers and a stronger fan maybe needed to compensate the reduced pressure at the diffusers. As such, penetrations are usually very small cracks in between tile joints and are therefore hard to detect, so counter measures should be taken to help improve the air-tightness of the raised flooring by providing better seals in-between tile joints. Bauman has suggested that a layer of carpeting can significantly reduce air leakage between floor panels (Bauman 2003).

3. Research Methodology

Two laboratories namely the Field Environmental Chamber (FEC) and the BubbleZERO were used to conduct the experiments. Experiments relating to leakage characteristics of plenum supply UFAD systems and OHAD systems were performed in the FEC. Experiments relating to leakage characteristics of ducted UFAD systems were performed in the BubbleZERO. A volumetric flow hood was used to measure the volumetric flow rate of air supplied at the supply diffusers and exiting at the return grille. All door joints, window joints, wall to floor joints, light fixture joints, and OHAD supply diffusers were sealed using duct tape to prevent infiltration/exfiltration during plenum flow UFAD measurements. Only operational UFAD supply diffusers and return grilles are left open. During OHAD measurements, only supply diffusers of plenum flow UFAD system were sealed. The supply diffusers of OHAD system and return grilles were open. The leakage in plenum supply UFAD network was calculated as the difference in total volumetric flow rate of air exiting all return grilles and the total volumetric flow rate of air supplied through all diffusers. The leakage of OHAD system was calculated as the difference in total volumetric flow rate of air...
supplied from all OHAD diffusers and total volumetric flow rate of air exiting through all return grilles. The leakage in the BubbleZERO laboratory was calculated by measuring the difference in total volumetric flow rate of air supplied through all ventilation units and total volumetric flow rate of air exiting all supply diffusers of the ducted UFAD system.

3.1 Field Environmental Chamber (FEC)
The FEC is a research facility in the Department of Building at the National University of Singapore (NUS). The chamber is a replica of a typical office environment equipped with air-conditioning and mechanical ventilation systems. The facility is designed to be capable of operating the air distribution system in the Over Head Air Distribution system (OHAD), Displacement Ventilation system (DV) or Under-Floor Air Distribution system (UFAD) mode. A schematic diagram of the FEC is shown in Figure 3.
As shown, the size of the FEC is 11.12 m length, 7.46 m width, 2.6 m height. The FEC has a raised access floor and a suspended acoustic ceiling. The total floor area of the chamber is about 79.54 m². There are sixteen workstations in the chamber positioned at the locations shown in Fig 3. Each workstation consists of one table, one desktop computer, one monitor, and one office chair. There are three plenum sections divided by two plenum dividers. Among all diffusers, only nine supply UFAD diffusers were selected to operate throughout the experiments. The operating supply diffusers are identified as SD # 1, SD #2 … SD #9. Rest of the diffusers were completely blocked and sealed. In total there are six return grilles installed in the ceiling at the positions shown in Fig 3 namely RD #1, RD #2… RD #6. These return grilles were used to extract the air from the FEC during UFAD and OHAD experiments. Six ceiling supply diffusers were used in the experiments concerning OHAD systems, but their locations are not shown in figure 3.

Although the FEC was designed to contain a raised floor for the use of DV and UFAD systems, visual inspections showed that the joints in between floor panels and floor/wall connections were not properly sealed. This resulted in high amounts of uncontrolled air leakage from the raised floor into the chamber. So, effective sealing of the gaps between floor and ceiling panels, electrical outlets, floor-to-wall joints, unused floor diffusers, north wall folding doors, south wall glass panels, doors, light fixtures, ceiling supply diffusers and window joints were done for the UFAD measurements as shown in Figure 4. Additional heavy duty Styrofoam boards were placed on top of sealed diffusers and electrical outlets to prevent leakage.

![Figure 4: Fully taped floor and walls, Styrofoam boards placed on top of floor diffusers](image)

### 3.2 BubbleZERO
The BubbleZERO is an experimental laboratory made by ETH, Zurich & Singapore-ETH Centre for Global Environmental Sustainability (SEC) in Singapore. It consists of two 6.1 m high containers to represent a work-space of 6m length, 5m width, and 2.8m height. The BubbleZERO contains ducted UFAD systems. A representation of the laboratory is shown in Figure 5. The BubbleZERO laboratory was used to explore the leakage characteristics in a
ducted UFAD system. The laboratory includes radiant panel systems in the ceiling along with a building integrated decentralized ventilation system at the floor level.

Figure 5: A representation of the BubbleZERO laboratory (Iyengar et al. 2013)

Outside air is taken into the laboratory through four Dedicated Outdoor Air Supply (DOAS) ventilation units namely VB1, VB2, VB3 and VB4 as shown in Figure 6. The ventilation units are used for cooling and dehumidifying the incoming air. Each of the ventilation units has three sets of heat exchangers in series with each other. Chilled water is passed through the coils of the heat exchanger. The conditioned air is then transported through small circular ducts as shown in Figure 6 to the seven swirl diffusers D1, D2……..D7. Each of these diffusers are inter connected to each other, which is shown in the complex duct network. There are four return grilles in the ceiling that are not shown in figure 6, which are integrated in the radiant panel system. The laboratory is placed in an outdoor environment. The two cross marks are the spatial locations where measurements were performed.
Figure 6: Sampling locations, ducting network, outdoor air intake locations and swirl diffuser locations in the BubbleZERO laboratory ducted UFAD setup

3.3 Instruments Used
The data acquisition instruments were used to obtain measurements to examine the leakage characteristics of the UFAD and OHAD systems. AccuBalance Air Capture Hood 8371 from TSI, which is a compact balometer that is capable of measuring air volume flowing through diffusers and grilles, was used. The air hood is 0.610 by 0.610 m and fits on a typical FEC supply and return diffuser. It has an accuracy of ±5% of reading or ±5 CFM (±2.4 l/s, ±8.5 m³/hr) and is capable of capturing a flow range of 30 to 2000 CFM (15 to 1000 l/s, 50 to 3500 m³/hr).

4. Results and Discussion
The experiments include comparing leakage characteristics of plenum supply UFAD systems, ducted supply UFAD systems and OHAD systems. Each experimental case was allowed 60 minutes to reach steady-state conditions. The experiments in the FEC were carried out at a room temperature set point of 23°C, chilled water supply temperature of 6.1°C with flow rate
of 0.6 L/s and static pressure at 250 Pa in the supply duct of the AHU. The experiments in the Bubble ZERO were carried out with the following conditions: The room set point was 23 °C, the chilled water supply temperature to the ventilation units were 10 °C at 0.03 L/s flow rate. The radiant panels were switched off during the experiments. The ventilation rate was 11.1 L/s.

The volumetric flow rate of air supplied by the AHU to the plenum supply UFAD system is lower than that supplied to the OHAD systems on average by 864.5 m$^3$/h. The volumetric flow rate of air supplied by the AHU to the ducted supply UFAD system is lower than that supplied to the OHAD systems on average by 1767.3 m$^3$/h. The BubbleZERO laboratory, which houses the ducted supply UFAD system, has Dedicated Outdoor Air Systems (DOAS) ventilation units running on smaller capacity fans and can supply a maximum of 286 m$^3$/h of air into the indoor space, so the volumetric flow rate of air supplied to the ducted supply UFAD system is lower than the volumetric flow rate of air supplied to the plenum supply UFAD system on average by 913.3 m$^3$/h.

Figure 8 shows the measured volumetric flow rate of air at supply diffusers and return grilles for different AHU fan speeds of UFAD (plenum and ducted supply) systems and OHAD systems. In an ideal plenum supply UFAD system, which is devoid of leakages, an increase in AHU fan speeds would typically see an increase in air volume exiting the supply diffusers. But as seen from Figure 8, an increase in AHU fan speed from 50% to 100% does not reflect the same trend. The volumetric flow rate of air exiting the diffusers at 50% AHU fan speed is 1146 m$^3$/h and at 100% AHU fan speed is 1108 m$^3$/h. This suggests that there is leakage occurring in the under-floor plenum and air is leaking through cracks in floor joints into the room interiors instead of exiting from diffuser openings. The total volume of air returning through return grilles is an indication of the same effect, which shows that the volumetric flow rate of air exiting through the return grille...
increases as the AHU fan speed increases. The volumetric flow rate of air exiting through the return grille is a summation of volumetric flow rate of air exiting through floor diffusers and the volumetric flow rate of air leaked into the room through floor joints. Sealing of doors and window joints eliminated infiltration or exfiltration in this case. This indicates the amount of Category 2 leakage that is experienced by the FEC during a plenum supply UFAD system operation.

During the operation of OHAD systems in the FEC, the floor joints and floor diffuser openings were sealed. The window and door joints were unsealed. The main aim of doing this was to determine the category 1 leakage that could exist in the FEC. The measurements show that the volumetric flow rate of air exiting through return grilles is lower than the volumetric flow rate of air supplied through diffusers; this indicates a Category 1 leakage experienced through exfiltration in the FEC during OHAD system operations. The quantification of this leakage is shown in Figure 9.

The supply and return volumetric flow rates of air in ducted systems however, showed very little quantitative variation. The measurements showed that the volumetric flow rate of air at the supply diffuser was less than the volumetric flow rate of air being supplied by the ventilation units. The average leakage in ducted UFAD systems was 3.8%. This indicates category 2 leakage occurring in ducted UFAD system, the magnitude however, being relatively small. The air leakage could be through duct joints.

![Figure 9: Total percentage leakage for different supply air volumes from AHU for UFAD (plenum supply and ducted supply) and OHAD systems](image)

Figure 9 shows the percentage leakage occurring in UFAD and OHAD systems at different volumetric flow rate of air supplied by the AHU. The results show that the leakage in plenum supply UFAD system increases from 5.9% to 10.2% to 14.8% when the supply air volume by AHU changed from 50% (636 m³/h) to 75% (954 m³/h) to 100% (1272 m³/h) respectively. In the ducted supply UFAD system, the change was relatively lower from 2.1% to 3.9% to 5.6%
when the supply air volume by AHU changed from 50% (143 m$^3$/h) to 75% (214.5 m$^3$/h)) to 100% (286 m$^3$/h). The leakage in OHAD systems was averaged at 19.7%. The leakage in plenum supply UFAD systems is shown positive as return air quantities were higher than supply air quantities. The leakage in ducted UFAD systems is shown to be negative as the amount of air measured at the diffuser is lower than that supplied by the DOAS system.

5. Conclusion
Leakage characteristics analysis shows that OHAD systems experience Category 1 leakage, mainly through exfiltration. Sealing of window and door joints could prevent this type of a leakage from occurring. Pressurized plenums in plenum supply UFAD systems experience Category 2 leakage through gaps in floor joints of the false flooring. Carpeting or better sealing of floor joints could prevent this type of a leakage. Category 2 leakages also occur in ducted supply UFAD systems through duct joints and better sealing of duct joints could prevent such leakages. The results show that the leakage in UFAD systems increases as AHU fan speed increases. The leakage in a ducted UFAD system in comparison to a plenum supply UFAD system was 64.4% less at 50% AHU fan speed, 61.8% less at 75% AHU fan speed and 62.2% less at 100% AHU fan speed respectively. Plenum supply UFAD systems could be used in large office layouts or open office designs where individual thermal comfort is not a priority. Ducted UFAD systems could be used to achieve localised thermal comfort controls at diffuser level in individual work cabins. UFAD systems could work better to OHAD systems due to the elimination of large false ceilings and the advantage of having to move air in the same direction as the buoyancy driven thermal lift in the room.

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References


### 3.4. Takeaway

The takeaway from this chapter is that air must be stratified with a temperature gradient for the working of the displacement ventilation concept. The ducted UFAD system shown here is a working proof of displacement ventilation. Perimeter heat gain can affect stratification in a significant way and can in turn influence human comfort. The dry bulb temperature of air increases significantly due to perimeter heat gain. This increase can be minimized by incorporating working radiant panels supplied with high temperature chilled water. The issue
of cold feet must be taken into account and preferably the supply air temperature at the diffuser must be more than 18°C. The temperature gradient between the supply and return locations of air must be at most 3°C. The energy saving potential of UFAD systems is immense from literature findings, but the cost of employing such systems is still debatable. UFAD systems mostly experience category 2 leakage and care must be taken to seal floor and tile joints in a plenum supply UFAD system and duct joints in a ducted supply UFAD system. Ducted UFAD systems perform better when it comes to reducing leakage in comparison to plenum supply UFAD systems. Category 1 leakage can lead to an increase in temperature and moisture content indoors as enunciated in detail through chapter 2.
Chapter 4
Radiant Panels & Ventilation Units
The design and cooling capacities of the radiant panel system and ventilation units installed in the BubbleZERO are discussed through this chapter. Any influence panel surfaces temperatures could have in shaping cooling capacities of radiant panels are also highlighted. The percentage of indoor cooling achieved through convective heat transfer and through radiant heat transfer for different supply temperatures of chilled water to radiant panels is calculated. Enhanced radiant panel designs to achieve higher cooling capacities are also studied. This chapter also investigates the cooling capacities of ventilation units and indoor stratification achieved at different supply chilled water temperatures to ventilation units and at different ventilation rates. Finally a heat load of the BubbleZERO laboratory is generated to understand the impact of various heat sources on the laboratory.

4.1. Radiant Panels

4.1.1. Literature

Radiant heating is a method that has been used extensively from the past few decades for heating indoor spaces. The success of the heating concept gave way for cooling system designers to try and incorporate the same concept for cooling purposes. The energy conservation and Indoor Air Quality (IAQ) benefits from using such a system coupled with a ventilation system has begun to attract recent attention in the building industry [1], [2] and [3]. Different types of radiant cooling mechanisms include metal ceiling panels, chilled beams or tube imbedded ceilings or walls or floors. The mechanism of radiant cooling involves two heat transfer processes namely convection and radiation. The capacity of radiant cooling can be influenced by numerous factors like height of the radiant panel system from the occupant zone, the panel surface temperature, temperature of chilled water supplied to the radiant panels, surface temperature of the indoors, supply air temperature, outdoor airflow rate and cooling load [4]. A study by Jeong [5] found that the total cooling capacity of radiant panels can be enhanced in mixed convection situations by 5% to 35% under normal operating panel surface temperatures. The most important concern that needs to be addressed for the feasible operation of this system is to avoid the risk of condensation on the chilled ceiling panel. To prevent condensation on the panel, it is important to properly control the system for transient regimes, such as at the start up and shutdown periods, and to minimize infiltration of humid outdoor air. Loveday in his research [6] showed that a low chilled ceiling temperature, could increases the chilled ceiling cooling capacity and could hence increase air velocities in the occupied zone [6]. This velocity increase is due to the downward airflow motion caused by the negative buoyancy force from the chilled ceiling. But such a convection process could induce air velocities that are still low and usually do not cause draft. The highest air velocities, however would still be in the occupied zone and is usually just above the floor (~5 cm), which is the jet region and approximately the position of the maximum jet velocity from the displacement diffuser [7]. For the convective part of heat transfer, the convective heat transfer coefficient expresses the heat exchange between a specific surface and the air boundary layer; therefore it depends on several changing parameters like air velocity, air temperature, turbulence etc. Several algorithms deduced from experimental tests can be used to define, from time to time, the actual value of this parameter. In the case of radiant surfaces,
high values of the convective heat transfer coefficient are possible only for: cooled ceiling, heated floor and cooled or heated walls, which are able to generate, through buoyancy forces, air movements inside the room [8]. Instead, in the case of heated ceilings and cooled floors, the convective heat transfer coefficient has lower values, because few air movements are generated [9].

4.1.2. Radiant Panels Cooling Capacity

The indoor sensible cooling using radiant panels take places using 2 heat transfer processes; namely radiant cooling and convective cooling. A part of the cooling is done through radiative heat transfer and a part of it is done through convective heat transfer. Cooling capacities of the 2 processes in combination gives us the total cooling capacity of the radiant panels. The amount of heat removed through the radiant cooling process is given by equation 1 [10]:

$$q_r = 5\times10^{-8} \left[ (T_p + 273.15)^4 - (AUST + 273.15)^4 \right] \ldots \ldots \text{eq. 1}$$

Where

$q_r$ is the amount of heat removed through radiant cooling (W)

$T_p$ is the effective panel surface temperature (°C)

AUST is the area weighted temperature of all indoor surfaces (°C)

AUST is calculated using the formula in eq. 2 [10].

$$\sum A_i \varepsilon_i T_i / \sum A_i \varepsilon_i \ldots \ldots \text{eq. 2}$$

Where

$A_i$ is the area of each of the indoor surface being considered (m$^2$)

$\varepsilon_i$ is the emissivity of each of the indoor surface being considered

$T_i$ is the surface temperature of the indoor surface being considered (°C)

Fig 4.1 shows the radiant panel cross section with the chilled water flow circuit as installed in the BubbleZERO. Fig 4.1 shows that the chilled water flows only through certain parts of the radiant panel. There is a distinct middle section of the panel surface on both sides of the CO$_2$ extraction and LED lights section, which do not have direct contact with the chilled water pipes. The total area of chilled water pipe casing in contact with the panel surface is 4.5m x 0.15m x 4 lengths, which totals to 2.7m$^2$. 
Fig 4.1: Cross section of the radiant panels installed in the BubbleZERO showing the chilled water flow loop

Fig 4.2 shows the total radiant panel area, which was divided into 24 sections to measure surface temperature of the panel surface in each section. The temperatures measured were numbered as $T_1$, $T_2$, $T_3$, $T_4$,$\ldots$,$T_{23}$, $T_{24}$. For each of the sections shown in Fig 4.2, effective heat removed through radiant cooling was calculated using eq. 1. The $T_p$ for each of the section is the temperature ($T_i$) that was measured in that panel section.
Fig 4.2: Surface temperature distribution across the radiant panel (T<sub>i</sub>)

The AUST calculations for each of the sections shown in Fig 4.2 had the following assumptions of emissivity and area as shown in Tab 4.1 and Tab 4.2. The values have been calculated by looking at manufacturing data. The panel surface temperature T<sub>i</sub> is the temperature measured for each of the section as shown in Fig 4.2.

Tab 4.1: Indoor surface emissivity values of BubbleZERO

<table>
<thead>
<tr>
<th></th>
<th>0.1</th>
<th>0.92</th>
<th>0.8</th>
<th>0.85</th>
<th>0.85</th>
<th>0.8</th>
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<tbody>
<tr>
<td>ε&lt;sub&gt;m glass&lt;/sub&gt;</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ε&lt;sub&gt;non m glass&lt;/sub&gt;</td>
<td></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ε&lt;sub&gt;facade&lt;/sub&gt;</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>ε&lt;sub&gt;gyypsum board&lt;/sub&gt;</td>
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<td></td>
</tr>
<tr>
<td>ε&lt;sub&gt;concrete&lt;/sub&gt;</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>ε&lt;sub&gt;raised floor&lt;/sub&gt;</td>
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<td></td>
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</tr>
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</table>
Tab 4.2: Indoor surfaces areas (m²) of BubbleZERO

<table>
<thead>
<tr>
<th>Area</th>
<th>Value</th>
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<tbody>
<tr>
<td>Area of m glass</td>
<td>4.935</td>
</tr>
<tr>
<td>Area of non m glass</td>
<td>4.935</td>
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<tr>
<td>Area of right façade</td>
<td>12.18</td>
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<tr>
<td>Area of gypsum board</td>
<td>4.935</td>
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<tr>
<td>Area of m glass</td>
<td>4.935</td>
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<td>12.18</td>
</tr>
<tr>
<td>Area of concrete ceiling</td>
<td>13.63</td>
</tr>
<tr>
<td>Area of non-concrete ceiling</td>
<td>13.63</td>
</tr>
<tr>
<td>Area of concrete floor</td>
<td>13.63</td>
</tr>
<tr>
<td>Area of non-concrete floor</td>
<td>13.63</td>
</tr>
<tr>
<td>Area of Panel</td>
<td>6.3</td>
</tr>
</tbody>
</table>

The second part of the heat removal from indoors, happens through convective heat transfer. The amount of heat removed through the convective cooling process is given by eq. 3 [10].

\[ q_c = 2.13 |T_p - T_a|^{0.31} (T_p - T_a) \] ……………… eq. 3

Where

- \( q_c \) is the amount of heat removed through convective cooling (W)
- \( T_p \) is the effective panel surface temperature (°C)
- \( T_a \) is the room temperature at the occupant level (1.5m from the floor) (°C)

The \( q_c \) is also calculated for each section of the panel area shown in Fig 4.2 to get a convective heat transfer distribution of the whole panel section.

The total cooling capacity of each section of the radiant panel system as shown in Fig 4.2 was then calculated using eq. 4 and the results are averaged to get the total cooling capacity of the radiant panel system.

\[ q_t = q_r + q_c \] ……………… eq. 4

Tab 4.3: Percentage of cooling capacity handled by radiation and convection at different supply chilled water temperatures to radiant panels

<table>
<thead>
<tr>
<th>Chilled water temperature (supply / return) (°C)</th>
<th>( q_c ) (W/m²)</th>
<th>( q_r ) (W/m²)</th>
<th>( q_t ) (W/m²)</th>
<th>( q_{water} ) (kW)</th>
</tr>
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<tr>
<td>15.5 / 18.5</td>
<td>14.4</td>
<td>20.5</td>
<td>34.9</td>
<td>0.37</td>
</tr>
<tr>
<td>16.5 / 19.7</td>
<td>11.1</td>
<td>19.7</td>
<td>30.8</td>
<td>0.40</td>
</tr>
<tr>
<td>17.5 / 20.4</td>
<td>8.7</td>
<td>17.6</td>
<td>26.3</td>
<td>0.37</td>
</tr>
<tr>
<td>18.5 / 21.4</td>
<td>5.8</td>
<td>14.3</td>
<td>20.1</td>
<td>0.37</td>
</tr>
</tbody>
</table>
The breakup of cooling done by radiation heat transfer and by convection heat transfer is shown in Tab 4.3 for different supply chilled water temperatures to the radiant panels. The data presented is an average calculated across the whole panel area. The flow rate of water was measured to be an average of 0.03 kg/s. The results from Tab 4.3 show that the cooling capacity of radiant panels decreases as the supply chilled water temperature to the panel increases. The average decrease in cooling capacity of the radiant panel systems is 16.6% or 5.1 W/m² with 1°C increase in chilled water supply temperature to radiant panels. Fig 4.3 shows the percentage of sensible cooling performed by radiant panels through convection and through radiation at different chilled water supply temperatures. There was no occupancy during the experiments and the chilled water supplied to the ventilation units was at 10°C. The ventilation units were operating at 40% (11.1 L/s) of the maximum ventilation capacity.

Fig 4.3: Percentage of cooling done by radiant heat transfer and convective heat transfer at different supply chilled water temperatures to panel

It can be seen from Fig 4.3 that the cooling through radiant heat transfer decreases by an average of 4.1% for every 1°C rise in supply chilled water temperature. Similarly the heat removal through convective heat transfer increases by an average of 4.1% for every 1°C increase in chilled water supply temperature. Another issue that radiant panels installed in the BubbleZERO faced was that the total capacity of radiant panels decreased due to inconsistency of panel surface temperature across the area of the panel. The surface temperature measurements of the panel surface across different sections of the panel listed in Fig 4.2 is shown in Fig 4.4a when the supply and return panel chilled water temperature was 16.5°C and 19.7°C.
From Fig 4.4a it can be observed that the panel surface is not having a uniform temperature distribution and the middle section of the panel as indicated in Fig 4.1 on average has 1.83°C higher temperature relative to the section in contact with the chilled water pipes. This is the reason for a loss in the overall cooling capacity of the radiant panels. A thermal image of the same as shown in Fig 4.4b confirms the same measurement, where it can be seen that the section of the panel in contact with the pipes having water flowing in them are at a temperature of 20°C to 21°C and the middle section is at a temperature of 22.5°C to 23°C. A further study explained at the later part of this chapter shows that a more uniform distribution of panel surface temperature could increase the cooling capacity of the radiant panels by a considerable amount.
Fig 4.4b: Panel surface temperature distribution across the area of the panel – thermal image (°C)

Fig 4.5 shows the cooling capacity of the radiant panel with respect to variation in supply chilled water temperatures.

Fig 4.5 clearly enunciates the decrease in panel cooling capacity due to an increase in supply chilled water temperature to radiant panels.
Now if the radiant panel water flow arrangement is changed to the one shown in Fig 4.6 by enhancing the number of chilled water pipe loops across the area of the panel, the capacity of the panel could increase due to a more uniform distribution of panel surface temperature.

The analysis of any enhancement in cooling capacities is shown thorough Fig 4.7a and Fig 4.7b. The study shows that the replacement of the non-uniform temperature middle section by a more uniform temperature setup, which is in contact with the radiant panel surface, could increase cooling capacity.
Fig 4.7a: Cooling capacities of the existing design and enhanced design of radiant panels

Fig 4.7b: Percentage increase in cooling capacity with different supply chilled water temperatures due to the enhanced design

It can therefore be seen that the increase in cooling capacities of radiant panels due to increased chilled water loops in the radiant panel design is more significant at lower supply chilled water temperatures. The cooling capacity could increase by up to 24.7% at a 15.5°C supply chilled water temperature with respect to the existing situation.

The next section describes the variation in indoor dry bulb temperatures when supply temperatures of chilled water are varied to the radiant panels at different ventilation rates supplied to the room.
4.1.3. Chilled Water Supply Temperature Variation in Radiant Panels

The change in indoor dry bulb temperature for different chilled water temperatures supplied to the radiant panels and ventilation rates is shown through Fig 4.8. The chilled water temperature supplied to the ventilation units were 8°C, the occupancy was 2 and the dry bulb temperature was measured at 1.5m from the floor.

![Fig 4.8: Change in indoor dry bulb temperature for varying chilled water temperature supplied to the radiant panels at different ventilation rates](image)

It can be inferred from Fig 4.8 that the dry bulb temperature in the human zone increases with the increase in ventilation rate. Also if chilled water temperature supplied to the panel decreases, the dry bulb temperature correspondingly decreases. For every 2°C reduction in chilled water temperature supplied to the radiant panels, the decrease in dry bulb temperature is 0.4°C at a ventilation rate of 11.1 L/s. This difference reduces as ventilation rate increases. The graph also shows that at higher ventilation rates, the effect of changing the temperature of chilled water supplied to the radiant panel has no effect in varying the dry bulb temperature of the space; this could be because of convective factors being more dominant in influencing dry bulb temperatures rather than the radiant factors from the panel. The cooling capacity of radiant panels at 18°C is as low as 28.5W/m². So it can be concluded that radiant panels work best when coupled with low ventilation rates.
The ventilation rate was kept constant at 11.1 L/s for the data presented in Fig 4.9. It can be inferred from Fig 4.9 that the dry bulb temperature in the human zone increases with increase in occupancy. Also if chilled water temperature supplied to the panel decreases, the dry bulb temperature decreases. The decrease in dry bulb temperature for decrease in chilled water temperature supplied to the panels is almost constant. For every 2°C reduction in chilled water temperature supplied to the radiant panels, the decrease in indoor dry bulb temperature was 0.32°C.

The next section describes the cooling capacities of ventilation units installed in the BubbleZERO and how the cooling capacities are affected by varying supply chilled water temperatures and changing ventilation rates.

**4.2. Ventilation Units Cooling Capacity**

A Dedicated Outdoor Air System (DOAS) is a separate piece of equipment used to condition (filter, cool, dehumidify) the outdoor air brought into a building for ventilation. This conditioned outdoor air is then delivered either directly to each occupied space or to local HVAC equipment in a central system serving those spaces. Treating the outdoor air separately can make it easier to verify that sufficient ventilation reaches each occupied space and this can help control high indoor humidity levels.

The ventilation units installed in the BubbleZERO were a type of DOAS that delivers air directly to the indoor space through underfloor ducts. The units perform 2 tasks; dehumidification and ventilation. The outside air having an average dry bulb temperature of 30.4°C and an average dew point temperature of 24.2°C was dehumidified in the ventilation units. The four ventilation units installed in the BubbleZERO were tested for their cooling
capacity and the variation in cooling capacity with varying ventilation rates and supply chilled water temperature was also realised in this section.

The cooling capacity of one of the 4 identical ventilation units installed at the BubbleZERO was calculated using eq. 5.

\[ Q_t = m_w \times C_{pw} \times \Delta T_w \] 

Where

- \( Q_t \) is the total sensible and latent heat removal potential of the ventilation unit (kW)
- \( m_w \) is the mass flow rate of water in the ventilation unit’s cooling coils (kg/s)
- \( C_{pw} \) is the specific heat capacity of water (4.12 kJ/kgK)
- \( \Delta T_w \) is the difference of supply and return chilled water temperature of water supplied to the ventilation units (K)

The capacity of the ventilation units was calculated for four airflow volumes of outside air; 40%, 60%, 80% and 100% of maximum air flow rate (27.8 L/s) and as shown in Fig 4.10. Another set of measurements were done with varying chilled water temperatures supplied to ventilation units as shown in Fig 4.1. The chilled water supply temperatures were varied between 8°C, 10°C and 12°C. During all the experiments there was no occupancy and the radiant panels were switched off.

Fig 4.10 shows the results of cooling capacity variation when outside air volume passing through the cooling coil is varied. The chilled water supply temperature is maintained at an average temperature of 12°C in this case.

From Fig 4.10 it is seen that the cooling capacity of the ventilation units decreases as ventilation rate increases. This is because the time of interaction of outside air with the cooling coil reduces with increase in ventilation rate. There is a 20.43% decrease in the cooling capacity when the airflow volume is increased from 40% to 100%. The decrease in cooling capacity is 0.06kW with every 20% increase in ventilation rate.
As seen from Fig 4.11, when the chilled water supply temperature to the cooling coil increases, the cooling capacity decreases. The experiments were carried out at 40% of the maximum ventilation rate and without occupancy and non-operational radiant panels. The study showed that the cooling capacity decreased by 0.026 kW for every 2°C rise in supply chilled water temperature to the ventilation units.

Fig 4.12a: Dry bulb and dew point temperatures at the diffuser level at varying capacities (influenced by varying ventilation rates) of the ventilation units
From Fig 4.12a and Fig 4.12b, it can be seen that the indoor dry bulb temperatures and dew point temperatures increase with increase in volumetric flow rate of air and decrease with decrease in supply chilled water temperature.

The airside calculations for the amount of sensible load and latent load removed by the ventilation units were carried out using the formula observed in eq. 6 and eq. 7.

$$Q_{sa} = m_a \times C_{pa} \times \Delta T_a$$  \hspace{1cm} \text{eq. 6}$$

Where

- $Q_{sa}$ is the sensible load of outside air removed by the ventilation unit (kW)
- $m_a$ is the mass flow rate of air (kg/s)
- $C_{pa}$ is the specific heat capacity of air (1.005 kJ/kgK)
- $\Delta T_a$ is the temperature difference between outside conditions and air exiting the heat exchanger (K)

$$Q_{la} = m_a \times \Delta L_a$$  \hspace{1cm} \text{eq. 7}$$

Where

- $Q_{la}$ is the latent load of outside air removed by the ventilation unit (kW)
- $\Delta L_a$ is the enthalpy difference of outside air conditions and air exiting the heat exchanger conditions (kJ/kg)
- $m_a$ is the mass flow rate of air (kg/s)

Tab 4.4 gives the values of different air flow rates used for calculating the sensible and latent loads of outside air.
Tab 4.4: Air flow rate at different ventilation rates

<table>
<thead>
<tr>
<th>Ventilation rate (% of max)</th>
<th>40%</th>
<th>60%</th>
<th>80%</th>
<th>100%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow rate of air per ventilation unit (kg/s)</td>
<td>0.012</td>
<td>0.015</td>
<td>0.018</td>
<td>0.023</td>
</tr>
</tbody>
</table>

Fig 4.13: Sensible and latent load removed from outside air with varying ventilation rates

From Fig 4.13 it is seen that the sensible load removed from outside air increases with increase in ventilation rate. However the latent load removed from outside air decreases with increase in ventilation rate. The sensible load removal increases by an average of 0.041kW for every 20% increase in ventilation rate, but the latent load decreases by an average of 0.097kW for every 20% increase in ventilation rate. The net load removed however, decreases with increase in ventilation rate as confirmed by Fig 4.10.

Fig 4.14: Sensible and latent load removed from outside air with varying supply chilled water temperatures
From Fig 4.14 it can be seen that the removal of sensible and latent load decreases with every 2°C increase in chilled water supply temperature by 0.024kW and 0.024kW respectively.

The next section focusses on the stratification that can be achieved in operational radiant panel cum decentralized systems. It shows how stratification can be affected with varying ventilation rates and varying supply water temperatures to ventilation units. The working of radiant panels also affects stratification and this is discussed in detail in the next section.

4.3. Indoor Stratification

4.3.1. Literature
An Under Floor Air conditioning (UFAD) concept applying a DOAS system works on the principle of displacement ventilation to achieve stratified air in the indoor space. But the addition of a ceiling radiant panel system can affect the stratification achieved. This section describes the various studies done to determine the stratification achieved by such a combination system and the effect of chilled ceilings or radiant panels in influencing the achieved stratification. The average radiant ceiling surface temperature is a good predictor of the temperature difference between the head (1.1 m) and ankle (0.1 m) of a seated person in the occupied zone compared to other parameters related to the fraction of total cooling load removed. Schiavon in his study [11] developed a model to predict the temperature difference between the head and ankle as a function of the average radiant surface temperature for a ratio between the cooling load and the displacement airflow rate in a combined system operating displacement ventilation and chilled ceilings. The study revealed that when smaller active radiant ceiling areas are used (e.g., for a typical radiant ceiling panel layout), colder radiant surface temperatures are required to remove the same amount of cooling load (as compared to a larger area), which causes more disruption to the room air stratification. The study [11] also observed that the room air stratification in the occupied zone decreases as a larger portion of the cooling load is removed by the chilled ceiling, increases with higher radiant ceiling surface temperatures, and decreases with an increase in the displacement airflow rate [11]. Tan in his study [12] defined η, which varies between 0 and 1 as the ratio of the zone cooling load removed by the chilled ceiling to the total room cooling load. A pure chilled ceiling had η equal to 1 and a pure displacement ventilation system had η equal to 0. The study also showed that reducing the amount of the cooling load removed by displacement ventilation i.e., increasing η, implied the likelihood of reduced stratification in the room, and this, in turn, implied a reduction in the ventilation effectiveness of the system [12]. The study also mentioned that the displacement ventilation system removed 33% of the cooling load by maintaining a temperature gradient of at least 2°C/m. Behne from his study [13] stated that good thermal comfort and air quality could be maintained when the displacement ventilation system removed at least 20% to 25% of the total cooling load. Another concern usually with such systems is the value of achieved indoor air speeds. Causone in his study [14] showed by laboratory experiments that the combination of displacement ventilation with floor cooling, under a typical European office room layout, may cause the air temperature difference between head and ankles to exceed the comfort
range specified by ASHRAE Standard 55 [15]. The study also noticed that by increasing the airflow rate and thus raising the floor temperature, the vertical air temperature differences decreased and that the draft risk did not increase significantly [14].

4.3.2. Stratification Achieved in the BubbleZERO

Different operational setups as described in Tab 4.5a and Tab 4.5b were used to determine the stratification that could be achieved in the BubbleZERO. Measurements were made for 3 days in each of these setups. The dry bulb temperatures were measured at the diffuser level (0.1m from the floor), human level (1.5m from the floor) and panel level (2.1m from the floor). Continuous monitoring of the thermal parameters was carried out at 1min intervals. The results of the experiments are summarised in Tab 4.5a and Tab 4.5b.

Tab 4.5a: Dry bulb temperatures at different heights in the space when the panels were
switched off

<table>
<thead>
<tr>
<th>Panels Off</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Dry Bulb Temperatures @ varying ventilation volumes and supply temperatures of water to ventilation units; Zero occupancy in all cases</td>
<td></td>
</tr>
<tr>
<td>12°C supply water to ventilation units</td>
<td>40%</td>
</tr>
<tr>
<td>Diffuser level (°C)</td>
<td>19.4</td>
</tr>
<tr>
<td>Human level (°C)</td>
<td>22.9</td>
</tr>
<tr>
<td>Panel level (°C)</td>
<td>23.1</td>
</tr>
</tbody>
</table>

Tab 4.5b: Dry bulb temperatures at different heights in the space when the panels were
switched on

<table>
<thead>
<tr>
<th>Panels On</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>ZERO occupancy in all cases, 12°C supply temperature of water to ventilation units</td>
<td></td>
</tr>
<tr>
<td>Dry Bulb Temperatures@ varying ventilation volumes and supply temperatures of water to panels</td>
<td></td>
</tr>
<tr>
<td>40%</td>
<td>60%</td>
</tr>
<tr>
<td>Air flow volume</td>
<td></td>
</tr>
<tr>
<td>Panel water supply</td>
<td></td>
</tr>
<tr>
<td>Diffuser level (°C)</td>
<td>16°C</td>
</tr>
<tr>
<td>Human level (°C)</td>
<td>19.4</td>
</tr>
<tr>
<td>Panel level (°C)</td>
<td>22.3</td>
</tr>
</tbody>
</table>

The results listed in Tab 4.5a and Tab 4.5b are analysed through Fig 4.15a, Fig 4.15b, Fig 4.16a and Fig 4.16b.
From Fig 4.15a and Fig 4.15b it is seen that the stratification exists between the diffuser level (0.1m) and human level (1.5m), after which the stratification is very insignificant until the panel level (2.1m). The average stratification from the diffuser level to human level is about 3.7°C with varying ventilation flow volume. The stratification achieved between the diffuser level and human level varies from 3.3°C to 3.7°C when the supply chilled water to ventilation units varies from 8°C to 12°C at the same ventilation flow volume.
Fig 4.16a: Dry bulb temperature stratification at different supply chilled water temperatures to radiant panels; with 12°C chilled water supplied to ventilation units and at a ventilation rate that is 40% of maximum.

Fig 4.16b: Dry bulb temperature stratification at different ventilation rates; supply chilled water temperatures to radiant panels is 16°C and supply chilled water supplied to ventilation units is 12°C.

From Fig 4.16a and Fig 4.16b it is seen that the stratification of the vertical space is affected due to the effect of radiant panels being switched on. The plot shows that the advantage of radiant panels to influence dry bulb temperatures at the human level is greater at low volumetric flow rates of the ventilation units; at higher volumetric flows the radiant cooling is not influential in a significant way. It can also be seen that due to the effect of the radiant panels, there is a reversal of temperature stratification at the panel level. This causes the temperature at the panel level to drop below the temperature at the human level in some cases. The influence of radiant cooling to change indoor air temperatures is increased with decrease in chilled water temperature supplied to the radiant panels.

A heat load of the BubbleZERO is calculated in the next section to look at the various heat sources affecting the thermal characteristics of the indoor space.
4.4. Heat Load of BubbleZERO

In order to ascertain the thermal load balance of the laboratory, a heat load of the laboratory was developed. The heat load will provide information on the amount of heat entering the indoors through all facades and the amount of sensible and latent load that is added into the indoor space from other indoor sources. With knowledge of system capacities of radiant panels and ventilation units from section 4.1 and 4.2, the amount of sensible load handled by the radiant panel and the ventilation units can be determined. The latent load is only handled by the ventilation units. The use of an enhanced radiant panel system could give information on the increase or decrease of sensible load handled by the radiant panels. In this study the heat load for the worst case scenario i.e. with 4 occupants was generated. The laboratory dimensions are 6.4m (length), 4.7m (breadth) and 2.3m (height). Tab 4.6 gives information of the specifications of the laboratory, the façade dimensions and the direction of facades.

Tab 4.6: BubbleZERO laboratory size and façade specification

<table>
<thead>
<tr>
<th>Floor area (m²)</th>
<th>30.08</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ceiling area (m²)</td>
<td>30.08</td>
</tr>
<tr>
<td>West façade (Glazing - 1.2m x 2m x 3 pieces of M glass) (m²)</td>
<td>7.2</td>
</tr>
<tr>
<td>West façade (Glazing - 1.2m x 2m x 1 pieces of double glass) (m²)</td>
<td>2.4</td>
</tr>
<tr>
<td>East façade (Glazing – 1.2m x 2m x 2 pieces of M glass) (m²)</td>
<td>4.8</td>
</tr>
<tr>
<td>East façade (wood façade – 1.2m x 2m x 2 pieces) (m²)</td>
<td>4.8</td>
</tr>
<tr>
<td>North and south façade (6.4m x 2.3m x 2 sides) (m²)</td>
<td>29.4</td>
</tr>
</tbody>
</table>

The north and south facades, ceiling and floor were made of a combination of 1” XPS insulation on the outside plus 5mm container metal made of steel plus 1” wood façade on the inside. Tab 4.7 gives information on the façade material and their properties. The Resistivity (R) was calculated using eq. 8.

Resistivity (R) = Material thickness (D) / Thermal conductivity (K)……………….eq. 8

|
| Tab 4.7: Material specification of facades of BubbleZERO

<table>
<thead>
<tr>
<th>Material</th>
<th>Thickness (D) (m)</th>
<th>Thermal conductivity (K) (W/mK)</th>
<th>Resistivity (R) (m²K/W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wood</td>
<td>0.0254</td>
<td>0.13</td>
<td>0.195</td>
</tr>
<tr>
<td>XPS Insulation</td>
<td>0.0254</td>
<td>0.03</td>
<td>0.847</td>
</tr>
<tr>
<td>Steel</td>
<td>0.005</td>
<td>43</td>
<td>0.00012</td>
</tr>
</tbody>
</table>

The total resistivity (Rₜ) of the combination materials was 1.042 m²K/W. The overall U value for the north – south façade is U=1/Rₜ, which was calculated to be 0.959 W/m²K.

The heat gain through conduction was calculated using eq. 9.
\[ Q_c = U \times A \times (T_o - T_i) \] \hspace{1cm} \text{eq. 9}

Where
- \(Q_c\) is the heat gain through conduction (W)
- \(U\) is the overall conduction heat transfer coefficient (W/m\(^2\)K)
- \(A\) is the area of the façade (m\(^2\))
- \(T_o\) is the outside surface temperature of façade (°C)
- \(T_i\) is the inside surface temperature of façade (°C)

The conduction heat gain through facades was calculated and listed in Tab 4.8. The inside and outside surface temperatures on the facades were 23.4°C and 24.4°C respectively.

Tab 4.8: Heat gain through the non-fenestration North and South facades of BubbleZERO

<table>
<thead>
<tr>
<th>Heat gain through north and south facades (W)</th>
<th>28.21</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat gain through ceiling (W)</td>
<td>28.86</td>
</tr>
<tr>
<td>Heat gain through floor (W)</td>
<td>28.86</td>
</tr>
</tbody>
</table>

With respect to glazing, the west façade at the BubbleZERO was totally glazed with 75% of the façade space covered by M glass as discussed in section 1.5 of chapter 1 and 25% of the façade space was covered by a normal double glazing. Tab 4.9 shows the specification of the different glazing used.

Tab 4.9: Glazing specification of BubbleZERO

<table>
<thead>
<tr>
<th>Specification</th>
<th>U value (W/m(^2)K)</th>
<th>Shading Coefficient</th>
<th>Inside surface temperature (°C)</th>
<th>Outside surface temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>M Glass</td>
<td>0.8</td>
<td>0.345</td>
<td>24.1</td>
<td>25.6</td>
</tr>
<tr>
<td>Double Glazing</td>
<td>1.2</td>
<td>0.689</td>
<td>25.5</td>
<td>28</td>
</tr>
</tbody>
</table>

The radiant heat gain from the glazing was calculated using eq. 10 from the ETTV equation of BCA academy, Singapore [15].

\[ Q_r = 211 \times \text{WWR} \times \text{CF} \times \text{SC} \] \hspace{1cm} \text{eq. 10.}

Where
- \(Q_r\) is the total heat gain from the glazing through radiation (W)
- \(\text{WWR}\) is the window wall ratio (%)
- \(\text{CF}\) is the correction factor as prescribed by the BCA and equal to 0.7
- \(\text{SC}\) is the shading coefficient of the glazing

The conduction and radiation heat gain from west side glazing is tabulated in Tab 4.10. The conduction heat gain was calculated using eq. 9. The WWR used for M glass is 0.75 and for double glazing is 0.25.
Tab 4.10: Heat gain through west side glazing of BubbleZERO

<table>
<thead>
<tr>
<th>Heat Gain Type</th>
<th>Value (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conduction heat gain from M glass</td>
<td>8.06</td>
</tr>
<tr>
<td>Radiation heat gain from M glass</td>
<td>38.22</td>
</tr>
<tr>
<td>Conduction heat gain from double glazing</td>
<td>6.91</td>
</tr>
<tr>
<td>Radiation heat gain from double glazing</td>
<td>25.44</td>
</tr>
</tbody>
</table>

The east façade at the BubbleZERO was half glazed by M glass. The other half was a wood facade. The wood façade was 0.0127m thick and had a U value of 10.236 W/m²K. The temperature on the inside and outside of the M glass was 23.9°C and 24.5°C respectively. This side was relatively cooler than the west side, due to natural shading from trees. The inside and outside surface temperature of the fenestration was 24.5°C and 25.6°C respectively. Tab 4.11 shows the convective heat gains from the glazing and fenestration and the radiation heat gain from the glazing. The WWR used here was 0.5 and the correction factor 0.7.

Tab 4.11: Heat gain through east side glazing and fenestration of BubbleZERO

<table>
<thead>
<tr>
<th>Heat Gain Type</th>
<th>Value (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conduction heat gain from M glass</td>
<td>2.30</td>
</tr>
<tr>
<td>Radiation heat gain from M glass</td>
<td>25.48</td>
</tr>
<tr>
<td>Conduction heat gain from fenestration</td>
<td>54.05</td>
</tr>
</tbody>
</table>

The total heat gain from all facades and glazing was the summation of the heat gains shown in Tab 4.8, Tab 4.10 and Tab 4.11. So the total external heat gain was calculated to be 246.4 W. The amount of heat gain per occupant in the laboratory space is taken to be 70W of sensible load and 60W of latent [16]. No lighting load was considered as the LED lights release heat to the rear side of the LED module and this heat was directly exhausted through the central exhaust channel. The total equipment load in the laboratory was considered to be 5W/m², so for 30m² of space, the equipment load was calculated to be 150W.

In the scenario where the maximum capacity of the system was used, from Fig 4.5 of section 4.1 we can see that the maximum capacity of the radiant panel system was 33 W/m², so the total sensible cooling capacity that could be handled by 2 radiant panels installed in the laboratory making up a total area of 12m² was 396W. From Fig 4.14 of section 4.2, the maximum cooling capacity provided by each of the ventilation units was 938W, so a total of 4 units could provide a maximum cooling capacity of 3752W. Now for a maximum of 4 people in the indoor space the total internal sensible gain from humans is 280W and the total internal latent gain from humans is 240W. The equipment load is 150W. A summary of the load balance is given in Tab 4.12.
Tab 4.12: Sensible and latent heat loads of BubbleZERO

<table>
<thead>
<tr>
<th>Sensible load from facades (W)</th>
<th>246.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sensible load from humans (W)</td>
<td>280</td>
</tr>
<tr>
<td>Equipment load (W)</td>
<td>150</td>
</tr>
<tr>
<td>Sensible load of outside air (W)</td>
<td>724</td>
</tr>
<tr>
<td>Latent load from humans (W)</td>
<td>240</td>
</tr>
<tr>
<td>Latent load of outside air (W)</td>
<td>2496</td>
</tr>
<tr>
<td>Cooling capacity of the radiant panels (W)</td>
<td>396</td>
</tr>
<tr>
<td>Cooling capacity of the ventilation units (W)</td>
<td>3752</td>
</tr>
</tbody>
</table>

From Tab 4.12 it can be observed that the total indoor sensible load was 676.4W (sensible load from facades + sensible load from humans + sensible load from equipment) and the total latent load was 240W (latent load from humans). The sensible load from outside air was 724W and latent load from outside air was 2496W. Considering the cooling capacity of the radiant panels and ventilation systems, Fig 4.17 shows the percentage of sensible cooling handled by the radiant panel. Fig 4.17 also shows how this percentage increases with an enhanced cooling capacity radiant panel as discussed in Fig 4.6, Fig 4.7a and Fig 4.7b.

![Graph showing the percentage of sensible load handled by the radiant panels](image)

**4.5. Takeaway**

This chapter showed that the radiant panel’s capacity is dependent on the number of chilled water pipe loops that exist in the panel, which influence the surface temperature of the panel. More the number of loops, more uniform are the surface temperature of the panels and more is the cooling capacity of the panels. The study also showed that sensible cooling of the space by radiant panels take place using 2 heat transfer mechanisms; convective heat transfer and radiant heat transfer. The ratio of cooling done by convective heat transfer to radiant heat transfer increases with increase in supply chilled water temperatures to radiant panels. If chilled water loops are increased in the radiant panel by 50%, there could be a cooling...
capacity increase of up to 24.7%. The dry bulb temperature of the space at the human level decreases due to effect of radiant cooling from the panels, but the change is significant only at low ventilation rates. At high ventilation rates, the influence of the radiant panel to decrease dry bulb temperatures at the human level is almost insignificant. So radiant systems work best when coupled to low flows ventilation systems. The capacity of ventilation units decreases with increased ventilation rate or increased water supply temperature to ventilation units, which in turn leads to an increase in dry bulb temperature and dew point temperature at the diffuser level. The study also revealed that stratification of the indoor air is effected due to operational radiant panels. The sensible load percentage handled by radiant panels increased from 24% to 34.7% when their capacity increased from 33W/m² to 40.5W/m².
This chapter highlights the influence fluctuating outdoor environmental conditions could have on indoor measurements of thermal comfort data. An outdoor normalization matrix, setup through this study provides for necessary correction values to counter this fluctuation. A research paper in the process of being published in the Building and Environment Journal establishes an operational matrix for the feasible operation of the decentralized system in the tropics. The matrix provides for various operating scenarios of the decentralized system to achieve acceptable thermal comfort and indoor air quality along with associated best operating scenarios. The paper also addresses concerns of condensation on radiant panels when such systems are switched off overnight. A special section has been dedicated to discussing the issues of biological contamination in the tropics and how changes to system operations and maintenance procedures help solve any arising problems.

5.1. Thermal Characteristics of Indoor Spaces

Thermal comfort is a state of mind that conveys to a person, a sense of satisfaction of the thermal environment around him. Thermal comfort can therefore vary from person to person based on his physiological condition and psychological desires. The criteria to feel thermally comfortable also vary across geographical locations based on the local climatic adaptability of people. Occupants in Europe may prefer an air conditioned space that has a low amount of air draft, while occupants in the tropics usually prefer relatively higher air movements. A study by Gong, showed that the acceptable air velocity for thermal comfort was more than 0.3m/s for occupants of air conditioned spaces in Singapore [2]. ASHRAE Std. 55 [2] highlights the six primary factors that must be addressed when defining conditions for thermal comfort. The six factors are metabolic rate, clothing insulation, air temperature, radiant temperature, air speed and humidity. The standard specifies a thermal comfort range where most occupants would feel comfortable based on Predicted Mean Vote (PMV) calculations. The ASHRAE thermal comfort chart shown in Fig 5.1 and Singapore Std. SS 553 [3] is used during this study for analysing the results and arriving at conclusions with respect to thermal comfort.

Fig 5.1: ASHRAE thermal comfort chart [2]
The thermal comfort range recommended by the Singapore Std. SS 553 [3] is to have a dry bulb temperature between 22.5°C and 24.5°C, relative humidity below 70% and air movement between 0.1 m/s and 0.3 m/s. All experiments conducted during this study check for conformance with this comfort range. But ASHRAE Std. 55 [2] also provides for a trade-off to achieving thermal comfort at higher indoor dry bulb temperatures. The trade-off is to increase the air speed as shown in Fig 5.2. There has been considerable research done to validate this claim. A research by Candido showed that indoor thermal comfort was achievable at a dry bulb temperature of 26°C, but when the air velocity was increased to 0.4m/s [4]. Another research by Toftum showed that humans generally do not feel air draught at air velocities of 0.4m/s and with air velocities at 1.6m/s, an air temperature of 30°C was also acceptable as thermally comfortable [5]. So the concept of being thermally comfortable at higher indoor dry bulb temperatures can be true, but very few subjective surveys have been performed in the tropics to prove this claim.

![Fig 5.2: Achieving thermal comfort at higher dry bulb temperatures using higher air velocities [2]](image)

The issue of cold feet is a potential concern for occupants working in spaces employing Under Floor Air Distribution (UFAD) systems. Low temperature air exiting the floor diffusers causes a cooling sensation near the feet of the occupants and this situation is perceived to be uncomfortable [6] [7]. The BubbleZERO also employs a UFAD setup to supply air to the interior space and care needs to be taken to avoid this problem. So having relatively higher temperature air exiting the diffuser could solve this problem, but it could pose problems of warmer sensation at the facial level due to stratification. ASHRAE std. 55 [2] as shown in Fig 5.3, suggests a bandwidth of dry bulb temperatures of air exiting the diffuser for which a majority of the occupants would be satisfied. For an 8.5% dissatisfied rate the air exiting the diffuser should have a dry bulb temperature of at least 20°C.
Although the control on indoor thermal conditions is important, it is important to study the influence of varying outdoor environments on indoor conditions. The next section discusses the outdoor environmental conditions for a typical tropical climate like Singapore; and how variations in outdoor environments affect indoor conditions.

5.2. Role of Environmental Conditions
Maintaining a steady state indoor environmental condition during experimentation is paramount to the consistency of the experiments. Although indoor conditions could be controlled by various system changes, the changing outdoor environment could have an impact on varying indoor thermal conditions. It is therefore important to study the variation in outdoor conditions and what impact it could have on changing indoor thermal conditions. Generating a normalisation model could help correct the measured indoor parameters in tune with the fluctuating outdoor scenario. This section discusses the fluctuations taking place in the outdoor biosphere in Singapore and its influence on varying the indoor thermal measurements. An outdoor normalization model is thereafter established specifying the correction values in indoor data for corresponding outdoor variations.

5.2.1. Outdoor Environment
The outdoor environment in Singapore varies significantly during the day as shown in Fig 5.4a. The hot and humid climate in Singapore witnesses dry bulb temperatures and dew point temperatures rise during the day and fall towards the evening. The results in Fig 5.4a were plotted after measuring and averaging six months outdoor data that represent a typical sunny day in Singapore. The outdoor conditions follow a quasi linear curve. The plot shows that the outdoor conditions do not vary much from 1900 hours to 0700 hours. From Fig 5.4b it can be seen that the dry bulb temperature and dew point temperatures then rise steeply from 0700 to 0900 at 1.5°C/h and 1°C/h respectively with the morning sun. The dew point temperature then fluctuates insignificantly from 0900 to 1700; while the dry bulb temperature rises at
0.38°C/h from 0900 to 1700 hours. The dry bulb temperature and dew point temperatures then drop steeply at 3°C/h and 1°C/h from 1700 to 1900.

Fig 5.4a: Outdoor conditions varying with time in Singapore

Fig 5.4b: Magnitude of increase in outdoor dry bulb temperature and dew point temperature with time in Singapore

Fig 5.4b gives an idea of the amount of increase in outdoor dry bulb temperature and dew point temperature with respect to a base line unfluctuating temperature that may be present between 1900 hours and 0700 hours. From Fig 5.4a it can be seen that the temperatures hardly fluctuate between 1900 hours and 0700 hours, so the temperature prevailing between this times is taken as the base line temperature. Is was also measured that the outdoor relative humidity stays constant at 80% between 2100 and 0900 hours; drops from 80% to 70% from 0900 to 1700 hours at a steady rate of 1.25%/h and rises from 70% to 80% from 1700 to 2100.
hours at a steady rate of 2.5%/h. A plot of the outdoor dry bulb temperature and dew point temperature curves following the equation \( y = mx + c \) during various time intervals is shown in Tab 5.1.

Tab 5.1: Curve equations of dry bulb temperature and dew point temperature for outdoor conditions in Singapore across various time intervals

<table>
<thead>
<tr>
<th>Time</th>
<th>Dry Bulb Temperature</th>
<th>Dew Point Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>0200-0700</td>
<td>( y = 0 )</td>
<td>( y = 0 )</td>
</tr>
<tr>
<td>0700-0900</td>
<td>( y = 1.5x - 7.5 )</td>
<td>( y = x - 5 )</td>
</tr>
<tr>
<td>0900-1700</td>
<td>( y = 0.38x + 0.34 )</td>
<td>( y = 2 )</td>
</tr>
<tr>
<td>1700-1900</td>
<td>( y = -3x + 51 )</td>
<td>( y = -x + 17 )</td>
</tr>
<tr>
<td>1900-0200</td>
<td>( y = 0 )</td>
<td>( y = 0 )</td>
</tr>
</tbody>
</table>

Using the curve equations plotted in Tab 5.1 the hourly increase or decrease of outdoor dry bulb temperature and dew point temperature with respect to the previous hour is created as shown in Fig 5.5, which is another representation of Fig 5.4b.

Fig 5.5: Magnitude of increase or decrease in dry bulb temperature and dew point temperature with time (with respect to the previous hour)

The next section focusses on how indoor conditions vary with time and what curve equations indoor dry bulb temperature and dew point temperature follow.

**5.2.2. Indoor Conditions**

The variation in indoor conditions inside the BubbleZERO, unlike the outdoor variation is exponential in nature as shown in Fig 5.6. The dry bulb temperature and dew point temperature rises exponentially from 0900 to 1700 hours and drops exponentially from 1700 to 0700 hours. There is no variation in both these parameters between 0700 and 0900 hours.
The nature of curves along with the curve equations are shown in Tab 5.2. The exponential curves follow the equation $f(t) = ae^{kt}$ where $t$ is time, $a$ is the initial value of the function and $k$ is the constant variable. The parameters listed in Tab 5.2 were measured at a height of 1.5m from the floor.

Fig 5.6: Varying indoor dry bulb temperature and dew point temperature with time

The indoor relative humidity does not change significantly with respect to the change in outdoor relative humidity conditions during this time period.

Tab 5.2: Curve equations of dry bulb temperature and dew point temperature for indoors

| Exponential Rise (Dry Bulb Temperature and Dew Point Temperature) | $y = ae^{0.013t}$ |
| Exponential Drop (Dry Bulb Temperature) | $y = ae^{0.013t}$ |
| Exponential Drop (Dew Point Temperature) | $y = ae^{0.012t}$ |

As it can be seen from Fig 5.6, on two separate days (1st cycle and 2nd cycle) with the system’s operating condition being unchanged, the indoor measured data can have different values at the same time. This variation in indoor measurements is because of a corresponding variation in outdoor environmental conditions on the two separate days. If an experiment is performed to study, what influence a system parameter change can have on other system parameters as performed in section 5 of the paper described in section 5.3, the need to have a steady state experimental setup is paramount. Any fluctuations in the outdoor environment could play a role in influencing the measured indoor data. So there is a need to develop an outdoor normalization matrix, where any fluctuation in the outdoor environment can be
respectively corrected in the indoor data. The correction values would also vary with different time intervals as outdoor environmental behaviour change with changing time intervals as shown in Fig 5.4b. A parameter fit of the outdoor data and indoor data was used to understand the influence of the outdoor environment on indoor conditions. This was used to generate the normalization model for different time intervals as discussed in the next section.

5.2.3. Outdoors’s Effect on Indoor Conditions

The fluctuation in the outdoor environment has a potential to affect indoor measurements in a significant way, so it could be beneficial to study the influence of these fluctuations on the indoor measurements. This study provides therefore for an outdoor normalizing model of dry bulb temperature and dew point temperature as shown through Fig 5.7a, Fig 5.7b, Fig 5.8a and Fig 5.8b respectively. The outdoor normalizing model gives the correction values for indoor dry bulb temperature and dew point temperature for corresponding variations in outdoor dry bulb temperature and dew point temperature respectively. The experimental conditions during these experiments were: The average chilled water supply temperature to ventilation units were 8°C. The radiant panels were switched off and there was no occupancy. The total ventilation rate was 11.1 L/s when the ventilation units were working at 40% capacity.

Fig 5.7a: Indoor dry bulb temperature correction needed for corresponding outdoor fluctuations in time intervals of 1900 hours to 0700 hours and 0700 hours to 0900 hours

Fig 5.7b: Indoor dry bulb temperature correction needed for corresponding outdoor fluctuations in time intervals of 0900 hours to 1700 hours and 1700 hours to 1900 hours
Fig 5.7a and Fig 5.7b shows the correction required in indoor dry bulb temperatures between 2 days having a variation in their outdoor condition. The first day is called a “Former Day” and the second day is called a “Latter Day”. The correction required varies with time of the day and corresponding time periods are, therefore indicated in Fig 5.7a and Fig 5.7b. The outdoor variation is the difference in outdoor dry bulb temperatures at the same time between the Latter and Former days. At a calculated outdoor variation value, the corresponding indoor correction value is added to the measured indoor data when the outdoor dry bulb temperature of the latter day is more than that of the former day; or subtracted from the measured indoor data when the outdoor dry bulb temperature of the latter day is less than that of the former day.

Fig 5.8a: Indoor dew point temperature correction needed for corresponding outdoor fluctuations in time intervals of 1900 hours to 0700 hours and 0700 hours to 0900 hours

Fig 5.8b: Indoor dew point temperature correction needed for corresponding outdoor fluctuations in time intervals of 0900 hours to 1700 hours and 1700 hours to 1900 hours

Fig 5.8a and 5.8b shows the correction required in indoor dew point temperatures between 2 days having a variation in their outdoor condition. Similar to the dry bulb temperature case, the first day is called a “Former Day” and the second day is called a “Latter Day”. The correction required varies with time of the day and corresponding time periods are, therefore indicated in Fig 5.8a and Fig 5.8b. The outdoor variation is the difference in outdoor dew
point temperatures at the same time between the Latter and Former days. At a calculated outdoor variation value, the indoor correction value is added to the measured indoor data when the outdoor dew point temperature of the latter day is more than that of the former day or subtracted from the measured indoor data when the outdoor dew point temperature of the latter day is less than that of the former day.

![Graph showing the effect of outdoor dry bulb temperature on indoor dry bulb temperature.](image)

Fig 5.9: Outdoor dry bulb temperature’s influence on indoor dry bulb temperature for time intervals of 30 minutes

Fig 5.9 shows the effect of using the normalization model on the measured value of indoor temperatures. During this study, which was done during the day; the outdoor dry bulb temperature increases by 0.38°C/h as seen from Fig 5.4a. Fig 5.6 shows that the indoor dry bulb temperature decreases by 0.44 °C/h. This decrease is measured under the influence of an increasing outdoor dry bulb temperature. So if the outdoor dry bulb temperature wouldn’t rise, the decrease in indoor dry bulb temperature would be much steeper; it is calculated to be 0.99°C/h as shown in Fig 5.9 after using the normalization model. The same was verified using measurements done between 1900 hours and 0700 hours when there was no influence of outdoor temperature variation on the indoor measured data as shown in Fig 5.4b.

The next section discusses the operational matrix that was developed for the feasible working of the decentralized system in the tropics to achieve acceptable thermal comfort and indoor air quality. The matrix also highlights the best operating scenarios of the decentralized system. The paper also discusses the quantum of rise in indoor temperatures and humidity when the decentralised system was switched off overnight.

### 5.3. Thermal Characteristics of an Integrated Decentralized System

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Characterization of Radiant Panels and Decentralized Ventilation System for achieving Thermal Comfort and Indoor Air Quality in the Tropics

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Abstract
The use of radiant panel systems to cool building interiors have been well accepted in temperate climates, but this concept is yet to gain popularity to be accepted to cool buildings in hot and humid tropical climates. One of the reasons in the minds of building operators is the fear of condensation, which shadows the energy saving potential this type of a cooling operation may have. The study here aims at showcasing the conditions under, which radiant cooling systems can operate in the tropics when coupled with a decentralised ventilation system. The study shows that condensation on radiant panels doesn’t occur overnight once the systems are switched off. The overnight indoor air temperature rise is also arrested by increase in temperature of chilled water in the panel pipes, which act as thermal barriers with the water temperatures rising by 1.4°C. An operational matrix that is generated in this research shows that such systems can be operated at higher chilled water temperatures when compared to that supplied to conventional air conditioning systems. Results showed that a reduction of 5.6 L/s of ventilation rate can bring about 25% reductions in chilled water energy demand. The best operating scenario of the decentralized system was at a ventilation rate of 5.5 L/s-person and at 12°C and 19°C supply chilled water temperatures to ventilation units and radiant panels respectively.

Keywords
Radiant panels, Condensation, Ventilation, Tropics, Indoor Air Quality, Thermal Comfort

1. Introduction
Radiant panels have been in operation since quite some time in temperate climates, but the use of such cooling systems in hot and humid tropical climates is still debated today, primarily due to fears of condensation. Although the use of this technology to cool building interiors promises a reduced energy demand as seen from many works in literature, radiant panels can only satisfy sensible load requirements of an interior space. A dehumidification system has to be coupled with such system for the successful conditioning of rooms in the tropics. The dehumidification system would then take care of latent loads and ventilation
requirements. The use of a Dedicated Outdoor Air System (DOAS) for dehumidification could be one such solution. Literature shows that a combination system consisting of radiant ceilings and DOAS reduces air moving power by 25% to 30% of the airflow required for peak cooling loads in all air systems [1]. Mumma in his study [2] showed that the annual electrical energy consumption of a pilot DOAS cum radiant panel cooling system was 42% lesser than that of the conventional VAV system with an air-side economizer. The chiller size was thus brought down by 7.6% [2]. Radiant panels thus act as a controller for dry bulb temperature and the DOAS acts as a controller for humidity and dew point temperature.

Another important issue being discussed in many technical forums today is ventilation. Building related illness was not a problem till late 1950s, when buildings were primarily ventilated by opening and closing windows. But the need to air conditioned buildings demanded a sealed design, which limited the direct interaction of the building interiors with its exterior. This brought the need for mechanical ventilation systems to ventilate building interiors. The mechanical ventilation systems, brought with itself some choices such as percentage of air circulated, control on indoor temperatures and indoor humidity, location of intakes and exhausts in buildings etc. It also brought with itself, the need for maintenance of these systems, choice of materials used etc. Some of the choices made influenced in a direct way, the air quality of the indoor spaces and poor choices led to building related illness or it went on to coin the word sick building syndrome. Sick building syndrome could create many problematic symptoms to building occupants like dry eyes, block nose, dry throat, dry skin, headache, lethargy, watery eyes, runny nose, chest-tightness symptoms, flu-like symptoms, rashes etc.

The DOAS can be integrated with radiant cooling systems in either an active way, as in a chilled beam or in a passive way as in a decentralized ventilation system, which is adopted in this study. A DOAS also does not use recirculated air, so any contaminants released inside the building are not transported to other parts of the building as done by a mechanical system, so a building-wide contamination from a localized release would not occur [3]. A concern though, with adoption of such systems is the time taken to flush out humidity from room interiors before the operation of radiant panels, for preventing condensation. A study by Zhang showed that room air needed to be dehumidified for at least 1 hour prior to the operation of ceiling panels to avoid condensation [4]. This study looks to address some of these concerns.

2. Background
The proximity of Singapore to the equator, gives it a tropical rainforest like climate without the presence of any distinct seasons. Its geographical location and maritime exposure, gives it a climate that is almost always warm and wet. High humidity levels and abundant rainfall are always a part of Singapore’s climatic scenario. This section explains the background of decentralized Heating, Ventilation and Air Conditioning (HVAC) systems that could be used to condition air in such climates.
2.1. Decentralized Systems

HVAC systems are responsible for the biggest share of consumed energy in a building. In conventional HVAC systems the fresh air and return air are mixed in the Air Handling Unit (AHU) before being delivered to the room. The AHU caters to conditioning air of both the sensible and latent load in the building. A decentralized system however works on a slightly different mechanism where the sensible cooling is separated from the latent cooling and dehumidification. In decentralized systems as the ones used in this study, the radiant panel systems take care of the sensible load while another independent ventilation system takes care of the latent load. Such a combination of a radiant systems and DOAS have been seen to offer superior energy efficiency to that offered by a conventional air-conditioning system [5] [6]. The DOAS supplies 100 % fresh air to the space, satisfying the latent load, ventilation requirement, and a part of the space sensible load [7]. The radiant cooling system that use chilled ceilings or radiant ceiling panels or active/passive chilled beams satisfy remaining of the sensible load [7]. The decentralized system used in this study consisting of ceiling attached radiant panels and floor embedded ventilation cum dehumidifying units is shown in Fig 1. This system is a part of the BubbleZERO laboratory that has been constructed in Singapore [8].

Fig 1: Schematic setup of the BubbleZERO laboratory showing the decentralized system consisting of radiant panels and ventilation units

An advantage of using radiant cooling to condition buildings is because occupants tend to perceive the air to be much cooler than reality. Hittinger, in his research showed that it is possible to maintain a higher space dry bulb temperature with chilled ceilings without having to compromise on perceived comfort; a space of 25.6°C using radiant cooling gave a perception of a space at 23.9°C without radiant cooling [9]. Mumma in his research showed that the ventilation air could remove more sensible heat since there would be about 1.7°C larger temperature rise by incorporating a radiant cooling cum underfloor DOAS with air
passes through a vertical space [10]. But one concern with such systems is of low air movements. A research showed that occupants have the tendency to feel approximately one scale cooler than still air conditions when small air movements in radiant cooling is experienced at the same level of effective temperature [11]. Studies by Scofield, Kilkis and various others have proposed different decoupling systems in the last decade as they were of the mind-set that most of the problems in all-air VAV systems were caused by the fact that a single HVAC system is requested to perform too many tasks simultaneously [12] [13].

2.2. Importance of Indoor Air Quality (IAQ)
Indoor Air Quality is an issue that is plaguing today’s building designers. The constant demand for increasing ventilation rates for better IAQ is being debated for its associated increase in energy demand. Nonetheless, the importance of maintaining good air quality in building interiors cannot be compromised. Some of the indoor contaminants are Carbon-di-Oxide (CO₂), Carbon Monoxide (CO), Formaldehyde (HCHO), Total Volatile Organic Compounds (TVOCs), Particulate Matter (PM₁₀ and PM₂.₅), Dust and Biological Contaminants (Bacteria and Fungi). Particulate matter has known to cause major respiratory problems to building occupants. Dust is usually measured in microns and inhalation of large concentrations of PM₁₀ and PM₂.₅ can cause problems in nasal tracts. Humidity levels play an important role in controlling indoor particulate concentrations. A study found that humidity levels less than 30% could lead to increased airborne dust levels as well as static electricity [14]. TVOCs are usually released from processes like printing, photocopying and from paints. So indoors with newly painted surfaces would have a tendency to have higher levels of TVOCs. Following the classification given by a WHO working group on organic indoor air pollutants [15], it has become common practice to divide organic chemicals according to boiling point ranges and to discriminate between VVOC, VOC, SVOC and POM. If a VOC mixture is analysed in indoor air, the result is often expressed as TVOC [16]. Most reported TVOC concentrations in non-industrial indoor environments are below 1 mg/m³ and few exceed 25 mg/m³. Over this range the likelihood of sensory effects like sensory irritation, dryness, and weak inflammatory irritation in eyes, nose, air ways and skin increases. At TVOC concentrations above 25 mg/m³ other types of health effects become of greater concern [15]. Sometimes the reported symptoms may be due to something completely different though and we may think it comes from the problems of VOCs. Wolkoff, in his study found that chemical reactions between oxidizable VOCs and oxidants, such as ozone and possibly nitrogen oxides, could form irritants which could be responsible for the reported symptoms [17]. Formaldehyde on the other hand is a reactive indoor pollutant that may induce airway irritation at low concentrations [18] [19]. In one experimental study, low level exposure to formaldehyde was known to cause mucous membrane irritation [20]. Burnett, in his study [21] found that CO, compared with the other major indoor gaseous pollutants, NO₂, SO₂, and O₃, was the strongest predictor of elderly patients being hospitalized for congestive heart failure [20]. Because carboxy-hemoglobin concentrations in blood could cumulative over time and prolonged exposure to low concentrations of CO could result in considerable health effects [22].
3. Methodology

The radiant panels and ventilation units which make up the decentralized system used in this research have been developed for a hot and humid climate and have been implemented in Singapore at the BubbleZERO laboratory. Evaluating the performance of this decentralized system for acceptable thermal comfort and indoor air quality is the objective of this study. The experiments conducted and presented in this study try to determine the impact of variations in chilled water temperature supplied to panels and ventilation units, ventilation rate and occupancy on dry bulb temperature, dew point temperature and relative humidity in the indoor space. The study also determines the operational limits of supply temperatures and ventilation rates in order to conform to local building codes. The thermal comfort parameters such as dry bulb temperature, relative humidity, and IAQ parameters such as concentrations of CO₂, CO, HCHO, TVOCs and particulate matter are measured at a height of 1.5m from the floor in the human breathing zone. The thermal comfort parameter of dew point temperature is measured at the radiant panel level, which is at a height of 2.1m from the floor. The dew point temperature is measured at this height as it would represent the condition of air surrounding the radiant panels. Condensation wouldn’t occur on radiant panels if the dew point temperature of air surrounding the radiant panels is lower than the surface temperature of the panels. Some of the limitations inherent to the laboratory are; the laboratory has a high surface to floor area ratio and the whole west façade is glazed. The east facade is also half glazed. These techniques of construction wouldn’t represent a real building construction technique in Singapore, so the laboratory could have more than conventional addition of heat through solar radiation. The interior of the laboratory space was divided into four equal quarters and measurements were done at the center of each quarter. Measurements of temperature and humidity were done at four spatial locations; at the center of each quarter in the laboratory. Each of these four spatial locations had one sensor at the diffuser level (0.1m from the ground), at the human level (1.5m from the ground) and at the panel level (2.1m from the ground). So in total of 12 sensors were used to carry out temperature and humidity measurements in the laboratory. The location of all 12 temperatures and humidity sensors are shown in Fig 2. Concentrations of HCHO, TVOC, CO₂, CO and particulate matter were done on a continuous measurement basis at the human level (1.5m from the ground) at all four spatial locations. Spot measurements of air velocity were performed at the human level (1.5m from the ground).
Each experiment was carried out for 8 hours during the day (9am to 5pm) and a steady state time of 16 hours was allowed between consecutive experiments. The dry bulb temperature, dew point temperature and relative humidity were monitored at 1 minute intervals on a continuous basis. The results plot an average of all four spatial locations at respective heights. Tab 1 gives information on the instruments used to collect the thermal comfort and indoor air quality data along with their respective resolution and accuracy functions.

Tab 1: Measuring instruments used to collect thermal comfort and IAQ data with their respective resolution and accuracy functions

<table>
<thead>
<tr>
<th>Measured Parameter</th>
<th>Instrument Used</th>
<th>Accuracy / Resolution / Repeatability</th>
</tr>
</thead>
</table>
| Air Velocity       | Kanomax Anemomaster Model A031 Series | Acc.: ± 2.0% of reading or ± 0.02m/s whichever is greater  
Res.: 0.01m/s |
| Surface Temperature| Fluke Dual Input Contact Digital Thermometer 54-ii | Acc.: ± 0.05% of reading |
| Mean Radiant Temperature | Testo Globe Thermometer | Acc.: ± 0.50°C |
| Dry Bulb Temperature (DBT), Relative Humidity (RH) and Dew | Sensirion Pin-type digital humidity and temperature sensors, SHT75 | Acc.: ± 0.25°C, ± 1.70% Relative Humidity  
Res.: ± 0.01°C, ± 0.05% Relative Humidity  
Rep.: ± 0.10°C, ± 0.10% Relative Humidity |
<table>
<thead>
<tr>
<th>Point Temperature (DPT)</th>
<th>Respirable Suspended Particles (RSP) PM\textsubscript{2.5}, PM\textsubscript{10}</th>
<th>Res.: ±0.1% of reading or 0.001 mg/m\textsuperscript{3}, whichever is greater</th>
</tr>
</thead>
<tbody>
<tr>
<td>DUSTTRAK™ DRX Aerosol Monitors Models 8533</td>
<td><strong>Carbon Dioxide, CO\textsubscript{2}</strong>&lt;br&gt;Carbon Monoxide, CO&lt;br&gt;Formaldehyde, HCHO</td>
<td><strong>Acc.: ±3.0% of reading or ±50 ppm, whichever is greater. Res.: 1 ppm</strong></td>
</tr>
<tr>
<td><strong>Carbon Dioxide, CO\textsubscript{2}</strong>&lt;br&gt;Carbon Monoxide, CO&lt;br&gt;Formaldehyde, HCHO</td>
<td>IAQ-CALCTM Indoor Air Quality Meters Model 7545</td>
<td><strong>Res.: 1 ppb</strong></td>
</tr>
<tr>
<td>Total Volatile Organic Compounds (TVOCs)</td>
<td>Portable Handheld VOC Monitor- ppbRAE 3000</td>
<td><strong>Res.: 1 ppb</strong></td>
</tr>
</tbody>
</table>

The results of this research investigate the interrelationships of the operating parameters of the decentralized system like occupancy, chilled water temperature and ventilation rate and how a variation in one of these elements brings about a variation in other elements to maintain indoor temperatures and humidity set points. The study uses this information to create an operational matrix for the working of the decentralized system satisfying thermal comfort and indoor air quality requirements specified by local standards. The operational combinations specified in the operational matrix have a condition of non-occurrence of condensation on the radiant panels i.e. when the dew point temperature of air surrounding the radiant panels is always lesser than the temperature of the panel surface. The final inference is to determine the limits of the operating parameters of the decentralized system for achieving thermal comfort and indoor air quality in Singapore. This would give a sense of the best case operational scenario.

4. Results of Variation in Indoor Temperatures and Humidity when the Decentralized System is switched off Overnight

Radiant system designers have always been worried about indoor temperature increase once the system is switched off over night. The perception that a large increase in indoor temperatures overnight would need the system to be switched on much in advance before occupancy commences the next day has perpetuated into resistance to adopt such systems. This section analyses the increase in indoor dry bulb temperature, dew point temperature and relative humidity after the system is switched off during the nights. The study also determines the rate of decrease of these parameters once the system is switched on in the morning. During the experimentation the system was switched off at 1900 hours and switched on at 0800 hours the next morning. The chilled water pumps to the ventilation system and radiant panel system were switched off. There was no occupancy and the ventilation units’ fans were also switched off. The ventilation units were supplied with 8°C chilled water and the radiant panels were supplied with 18°C chilled water with the average surface temperature of the panels being 21°C; the volumetric flow provided by the ventilation units was at maximum capacity of 27.8 L/s before the system was switched off and after the system was switched on. The rate of decrease in indoor temperatures would suggest the lead time needed to switch on the system before the room is occupied. The increase in dew point temperature overnight until the system turns on again would help determine if condensation would take place on the
panels overnight. Fig 3 gives details of the indoor climate during the switch off and switch on period of the systems.

From Fig 3 it can be observed that the dry bulb temperature and dew point temperature increase from the system switch-off time at 1900, till 0200 hrs. These temperatures achieve a steady state condition after 0200 hours until the system is switched on again at 0800. After system switch off, the increase in dry bulb temperature and dew point temperature is 0.9°C in 7 hours, so the temperatures increase at 0.13°C/h. The rise in temperatures is lower than expected as it can be seen that the temperature of chilled water stagnant in radiant panel pipes increase by 1.4°C from the time the system is switched off to the time it is switched on. This shows that the chilled water that is stagnant in the pipes acts as thermal barriers to avoid indoor air temperatures from rising tremendously. The average panel surface temperature recorded at 0700 hours was 21.5°C. Since the dew point temperature overnight was less than the overnight panel surface temperature, there was no condensation observed on the radiant panels. Usually buildings adopting radiant systems, have 24 hour cooling cycles and it becomes important to know the increase in dew point temperature overnight when the ventilation system is switched off. We also see that the dry bulb temperature and dew point temperature decrease from 0800 to 1030 and then attain a steady state condition once the system is turned on. The dry bulb temperature and dew point temperature decrease by 1.1°C and 0.8°C respectively in 2.5 hours. The rate of decrease in dry bulb temperature and dew point temperature is therefore 0.44°C/h and 0.32°C/h respectively. This information will help building operators to schedule the switching on of air conditioning systems to achieve the indoor set point conditions before occupancy commences.
5. Results of Different Operational Scenarios of the Decentralized System

Different operational scenarios for the feasible operation of the radiant panel cum decentralized ventilation system in the tropics are described in this section. The study would highlight the limits of operation of different parameters like chilled water temperature supplied to ventilation units, radiant panels and ventilation rate. The information gathered here would then be used to generate operational matrices for the tropics and present the best operational condition achieved in terms of achieving thermal comfort and IAQ at the BubbleZERO as described in section 6. All results presented here have been normalized for variations of outdoor environmental conditions. The criteria used for thermal comfort and indoor air quality conformance are those specified by ASHRAE Std. 55 [23], Singapore Std. SS 553 [24] and Singapore Std. SS 554 [25] and are as follows: The metabolic rate of the activity indoors is between 1 to 1.3 (seated reading or seated typing). The clothing value is 0.5 to 1 (trousers and long sleeve shirt). Dry bulb temperature lies between 22.5°C to 24.5°C, relative humidity lies between 60% to 70% for PMV satisfaction of 80%. Air velocity is less than 0.2 m/s. CO₂ concentration not to exceed 1000ppm, CO concentration not to exceed 9ppm, TVOCs concentration not to exceed 3ppm, HCHO concentration not to exceed 0.1ppm, particulate matter of size 2.5µm not to exceed 35µg/m³ and concentration of particulate matter of 10µm not to exceed 50µg/m³.

5.1. Variations in Ventilation Rate

The change occurring in thermal comfort and IAQ parameters listed in section 3 for variation in ventilation rates are described in this section. The study would also highlight the minimum ventilation rate that could be achieved without compromising the conformance standards specified in section 5. The occupancy was 2. The average temperature of chilled water supplied to ventilation units and radiant panels were 8°C and 15°C respectively. The capacity of radiant panel was 34 W/m². The total cooling capacity of ventilation units was 3.72kW, 3.48kW, 3.16kW and 2.96kW at a ventilation rate of 11.1 L/s, 16.7 L/s, 22.2 L/s and 27.8 L/s respectively.
Fig 4a shows the different thermal comfort parameters with varying ventilation rates in the BubbleZERO. The results show that the average indoor dry bulb temperature, dew point temperature and relative humidity increase by 0.5°C, 0.9°C and 2.6% for every 5.5 L/s increase in ventilation rate. The temperature and humidity increase with increase in ventilation rate, as the contact time for air passing over the cooling coils is reduced with increase in ventilation rate and consequently lesser cooling and dehumidification occurs. When the ventilation rate is reduced to 5.5 L/s, the dry bulb temperature and relative humidity drop to 22.3°C and 59.9% respectively, which do to conform to the threshold limits established by standards. So a minimum ventilation rate of 11.1 L/s is established. It is also seen that at the maximum ventilation rate of 27.8 L/s, the dew point temperature reaches 17°C, which is only 1°C lower than the surface temperature of radiant panels. Considering the inaccuracy of the sensors that is used in this setup, the dew point temperature could easily go beyond the surface temperature of the radiant panel causing condensation. So dew point temperatures at the radiant panel height should be carefully monitored for high ventilation rates. The average air velocity, which was induced using an extra stand-alone mechanical fan, was measured to be 0.11m/s.
Fig 4b: Concentration of CO₂ and CO at different ventilation rates

Fig 4c: Concentration of HCHO and TVOC at different ventilation rates
Fig 4d: Concentration of particulate matter at different ventilation rates

Fig 4b, 4c and 4d shows the variation in concentrations of CO₂, CO, HCHO, TVOCs and particulate matter with varying ventilation rates. The CO₂ concentration decreases with increase in ventilation rate at a rate of 155ppm for every 5.5 L/s increase in ventilation rate. But if the ventilation rate is reduced to 5.5 L/s, then the CO₂ concentration goes beyond the safe threshold limit specified by standards. So the ventilation rate cannot be reduced further from 11.1 L/s. The CO, HCHO, TVOCs and particulate concentrations on the other hand do not vary too much and are within their respective threshold limits as there are no known sources of pollution of these gases.

5.2. Variation in Occupancy

The change occurring in thermal comfort and IAQ parameters listed in section 3 for variation in occupancy are described in this section. The total ventilation rate was 11.1 L/s. The average temperature of chilled water supplied to ventilation units and radiant panels were 8°C and 15°C respectively. The capacity of radiant panel was 34 W/m². The total cooling capacity of ventilation units was 3.72kW at a total ventilation rate of 11.1 L/s.

Fig 5a: Dry Bulb Temperature (DBT), Dew Point Temperature (DPT) and Relative Humidity (RH) at different occupancies
Fig 5a shows the results of how indoor dry bulb temperature, dew point temperature and relative humidity vary with change in occupancy when the ventilation rate and chilled water temperature supplied to the ventilation units are kept constant. The results show that the average indoor dry bulb temperature and dew point temperature increases by 1°C with every increase in occupancy by 1. The average relative humidity increases by 2.8% with increase in occupancy by 1. The study also shows that at occupancy 4, the dry bulb temperature and relative humidity are 25.2°C and 70% respectively, which exceed the threshold levels set by standards. So the occupancy of the BubbleZERO cannot be increased beyond 4 people unless the chilled water temperature supplied to the ventilation units or radiant panels is decreased below 8°C or 15°C. But doing so would only increase energy consumption. Also the temperature of chilled water supplied to the radiant panels cannot be decreased below 15°C as the dew point temperature would exceed the surface temperature of the radiant panel, hence causing condensation. The average air velocity, which was induced using an extra stand-alone mechanical fan, was measured to be 0.12m/s at all occupancies except when the occupancy was 4. In the case with occupancy 4, the dry bulb temperature was 25.2°C and thermal comfort conformance was achieved by increasing the air velocity to 0.42 m/s as per velocity correction method to achieve thermal comfort at higher dry bulb temperatures provided for in ASHRAE Std. 55 [23].

Fig 5b: Concentration of CO₂ and CO at different occupancies
Fig 5c: Concentration of HCHO and TVOC at different occupancies

Fig 5d: Concentration of particulate matter at different occupancies

Fig 5b, 5c and 5d shows the variation in concentrations of CO₂, CO, HCHO, TVOCs and particulate matter with varying occupancies. The increase in CO₂ concentration with the addition of 1 person can be averaged at 161ppm. It can be seen that the CO₂ concentration reached 975 and 1150 at occupancies 3 and 4 respectively, as the ventilation rate drops to 3.7 L/s-person at occupancy 3. So the laboratory has to operate at a minimum ventilation rate of 5.5 L/s-person in order to achieve indoor air quality and increasing ventilation rate with increasing occupancies would decrease CO₂ concentration in the indoor space. The CO, HCHO, TVOCs and particulate concentrations on the other hand do not vary too much and are within their respective threshold limits as there are no known sources of pollution of these gases.

5.3. Variation in Chilled Water Supply Temperature to Ventilation Units
This section describes the change in thermal comfort and IAQ parameters listed in section 3 for varying chilled water temperatures supplied to the ventilation units. The total ventilation rate was 11.1 L/s. Occupancy was 2. The capacity of the radiant panel was 34 W/m². The
total cooling capacity of ventilation units was 3.88kW, 3.8kW and 3.72kW at supply chilled water temperatures to ventilation units of 8°C, 10°C and 12°C respectively.

Fig 6 shows how dry bulb temperature, dew point temperature and relative humidity vary with varying temperatures of chilled water supplied to the ventilation units at a constant indoor occupancy and ventilation rate. The results show that the average indoor dry bulb temperature, dew point temperature and relative humidity increase by 0.6°C, 0.9°C and 1.8% for every 2°C increase in chilled water temperature supplied to the ventilation units. The temperature and humidity would increase with increase in chilled water temperature because the air would be cooled and dehumidified with higher temperature chilled water. The average air velocity, which was induced using an extra stand-alone mechanical fan, was measured to be 0.11m/s. The upper limit of chilled water temperature supplied to the ventilation units is 12°C as if the water temperature supplied to the ventilation units is increased by another 2°C, the indoor dry bulb temperature and humidity would not conform to comfort standards. Also the dew point temperature would reach 16.9°C, which would cause condensation on the radiant panels. The chilled water supply temperature is not decreased below 8°C due to an associated increase in energy consumption.

6. Operational Matrix for the Feasible Working of the Decentralized System at BubbleZERO
All working combinations of the decentralized system for achieving indoor thermal and IAQ comfort conforming to standards specified in section 5 are presented as an operational matrix in this section. The matrix has been generated from information gathered through section 5.1, 5.2 and 5.3. Each point on the graph of the matrix represents one operational condition at a specified chilled water supply temperature to ventilation units and radiant panels; ventilation rates and occupancy. To achieve a set point indoor dry bulb temperature at the human zone is the target. All points on the graph represent scenarios devoid of condensation (when the dew point temperature of air surrounding the radiant panel is less than the surface temperature of
the radiant panels). The operational scenarios that do not satisfy this requirement have been omitted. The average surface temperature of radiant panels is 3°C more than the chilled water temperature supplied to the radiant panels i.e. a 15°C; 17°C and 19°C chilled water supplied to radiant panels would have the limiting criteria of 18°C, 20°C and 22°C as panel surface temperature. The analysis is performed for achieving five target air temperatures within the temperature range specified by standards – 22.5°C, 23°C, 23.5°C, 24°C and 24.5°C. The limits of supply chilled water temperatures to radiant panels are 15°C and 19°C, as temperatures below 15°C would increase the chances of condensation and temperatures above 19°C would considerably reduce the cooling capacity of radiant panels. Operational scenarios of supply chilled water temperature to ventilation units less than 8°C is omitted as it only increases the energy demand. The upper limit of the ventilation units chilled water supply temperature is 12°C as an increase to 14°C chilled water does not dehumidify outdoor air sufficiently and this could be a cause for condensation occurring on radiant panels as enunciated in section 5.3. The matrix has been generated for occupancies of 1, 2 and 3. The minimum ventilation rate adopted is 5.5 L/s-person due to IAQ concerns that can prevail for lower ventilation rates as highlighted in section 5.1. The maximum ventilation rate adopted is 10 L/s-person as higher ventilation rates only increases energy demand.

6.1. To Achieve an Indoor Dry Bulb Temperature of 24.5°C
All possible operational scenarios of parameter combinations to maintain an indoor dry bulb temperature of 24.5°C are listed in Fig 7. The listed combinations lie within comfort conditions of relative humidity, air velocity and IAQ as specified by the standards listed in section 5.

Fig 7: Operational scenarios to achieve indoor dry bulb temperatures of 24.5°C at different supply chilled water temperatures to ventilation units and radiant panels; ventilation rates

The maximum dew point temperature of air that was measured was 16.4°C; hence supplying a minimum of 15°C chilled water to radiant panels wouldn’t cause condensation on the panel surface whose temperature was 18°C. The relative humidity lies between 62.2% and 66.9%
amongst all cases. There are a total of 8 combinations with which the target dry bulb temperature was achieved as listed in Tab 2.

Tab 2: Feasible operational combinations to achieve indoor dry bulb temperature of 24.5°C

<table>
<thead>
<tr>
<th>Occupancy</th>
<th>Ventilation Rate (L/s)</th>
<th>Chilled water temperature to ventilation units (°C)</th>
<th>Chilled water temperature to radiant panels (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5.5</td>
<td>12</td>
<td>19</td>
</tr>
<tr>
<td>2</td>
<td>11.1</td>
<td>10</td>
<td>19</td>
</tr>
<tr>
<td>2</td>
<td>16.7</td>
<td>8</td>
<td>19</td>
</tr>
<tr>
<td>2</td>
<td>16.7</td>
<td>10</td>
<td>17</td>
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<tr>
<td>2</td>
<td>16.7</td>
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<td>15</td>
</tr>
<tr>
<td>3</td>
<td>22.2</td>
<td>8</td>
<td>17</td>
</tr>
<tr>
<td>3</td>
<td>22.2</td>
<td>10</td>
<td>15</td>
</tr>
<tr>
<td>3</td>
<td>27.8</td>
<td>8</td>
<td>15</td>
</tr>
</tbody>
</table>

The study showed that multiple parameter combinations could achieve the same target of indoor dry bulb temperatures. It can be seen that operating the system at low ventilation rates (5.5L/s-person), can not only reduce fan energy consumption, which is not very large in the decentralized system operating DC computer fans at the BubbleZERO, but it also helps operate the system at higher chilled water temperatures. If the ventilation rate is reduced from 16.7 L/s to 11.1 L/s at occupancy 2 or from 27.8L/s to 22.2 L/s at occupancy 3, the energy demand needed to chill water is reduced by 25% (increase in supply temperatures from 8°C to 10°C). So the best operating scenarios of the decentralized system at occupancies 1,2 and 3 were at 5.5 L/s, 11.1 L/s and 22.2 L/s ventilation rate; at 12°C, 10°C and 10°C chilled water supply to ventilation units; and at 19°C, 19°C and 15°C chilled water supply to the panels respectively. If ventilation rate is to remain unchanged, then a decrease of 2°C in supply chilled water to ventilation units would maintain indoor conditions without the need to change the radiant panel temperature with the doubling of occupancy. The same would apply in reverse in case of a decrease in occupancy.

6.2. To Achieve a Dry Bulb Temperature of 24°C

All possible operational scenarios of parameter combinations to maintain an indoor dry bulb temperature of 24°C is listed in Fig 8. The combinations listed lie within comfort conditions of relative humidity, air velocity and IAQ as specified by the standards listed in section 5.
Fig 8: Operational scenarios to achieve indoor dry bulb temperature of 24°C at different chilled water supply to ventilation units and radiant panels; ventilation rates

The maximum dew point temperature of air was 15.9°C; hence even supplying a minimum of 15°C chilled water to radiant panels wouldn’t cause condensation on the panel surface whose temperature was 18°C. The relative humidity lies between 57.4% and 67.2% amongst all cases. There are a total of 3 combinations with which the target dry bulb temperature was achieved as listed in Tab 3.

Tab 3: Feasible operational combinations to achieve indoor dry bulb temperature of 24°C

<table>
<thead>
<tr>
<th>Occupancy</th>
<th>Ventilation Rate (L/s)</th>
<th>Chilled water temperature to ventilation units (°C)</th>
<th>Chilled water temperature to radiant panels (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5.5</td>
<td>10</td>
<td>19</td>
</tr>
<tr>
<td>2</td>
<td>11.1</td>
<td>8</td>
<td>17</td>
</tr>
<tr>
<td>2</td>
<td>16.7</td>
<td>8</td>
<td>15</td>
</tr>
</tbody>
</table>

It can be seen that 3 combinations of operating scenarios could achieve the target dry bulb temperature. Reducing the ventilation rate by 33.5% (16.7 L/s to 11.1 L/s) at occupancy 2 had an associated reduction in chilled water energy demand by 13.3% (increase in chilled water supply temperature from 15°C to 17°C). The best operating scenario therefore at occupancies 1 and 2 were at a ventilation rate of 5.5 L/s and 11.1 L/s; at 10°C and 8°C chilled water supply to ventilation units; and at 19°C and 17°C chilled water supply to the panels respectively.

There were however no operational combinations to achieve 23.5°C, 23°C and 22.5°C of indoor dry bulb temperatures without having to increase the ventilation rate to more than 10L/s-person or without having to supply chilled water at a temperature lower than 8°C to ventilation units, both of which are an increase in energy demand of the system. The indoor temperature can also be achieved by reducing the supply temperature to radiant panels to less
than 15°C, but this could increase the chances of condensation. There is also no sufficient reason that such low temperatures need to be achieved in office spaces to achieve thermal comfort.

7. Practical Implication
This research provides knowledge on how radiant panel systems when combined with a decentralised ventilation system operate in the tropics to achieve thermal comfort and indoor air quality. The BubbleZERO laboratory is used as a pilot project where radiant panels and decentralized ventilation systems are being tested. The study gives information to building operators on how indoor temperatures and humidity vary when such a system is switched off overnight and the lead time required to achieve indoor comfort conditions once the system is switched on in the morning, before the building is occupied. The study also highlights the limits of operating conditions for chilled water supply temperatures and ventilation rate in order to prevent condensation and conform to thermal comfort and IAQ standards. An operational matrix for the successful operation of the decentralized system is generated, which gives the best case operating scenarios and control strategies for varying occupancies.

8. Conclusion
The results from experimentation in the BubbleZERO show that the operation of radiant panel systems is feasible in the tropics when it is coupled with a decentralised ventilation system performing dehumidification. The dry bulb temperature and dew point temperature increases by 0.13°C/h when the system is switched off over night, but only for 7 hours after which the temperatures achieve a steady state. The reason for such a low increase in indoor temperatures is because the chilled water in the panel tubes acts as thermal barriers for rise in indoor temperatures with the water temperature increasing by 1.4°C in the same time. The rate of decrease in dry bulb temperature and dew point temperature however once the system is switched on is 0.44°C/h and 0.32°C/h respectively. This information would help condition indoor spaces before occupancy commences. The study shows that the dehumidification potential of ventilation units is enhanced as ventilation rates decrease. This is because of the Low Face Velocity phenomena that occur over cooling coils, wherein the face velocity of air through the coil decreases when it passes over a larger area of the cooling coil, hence dehumidifying the air to a greater extent. But it is not possible to decrease ventilation rates to less than 5.5 L/s-person as this would increase CO₂ concentrations in the room beyond safe limits. The upper threshold of chilled water temperature supplied to ventilation units is 12°C; a further increase in temperature would limit the dehumidification potential of the cooling coil, hence increasing the dew point temperature of room air and risking condensation on radiant panels. The operational matrix shows that multiple parameter combinations of the system could achieve the target indoor dry bulb temperatures. Reducing ventilation rates by 33.5% helped achieve an associated reduction in chilled water energy demand by 13.3% due to the associated increase in supply chilled water temperatures.
Acknowledgement
This work was established at the Singapore-ETH Centre for Global Environmental Sustainability (SEC), co-funded by the Singapore National Research Foundation (NRF) and ETH Zurich. We would like to thank Cheng Li for his help with sensing equipment at BubbleZERO and Kian Wee Chen for his support during the experiments. We thank Ms. Wei Yi and Dept. of Building, National University of Singapore for providing the necessary Indoor Air Quality measurement equipment.

References


5.4. Biological Contamination of the Indoor Environment

Building related illness was not a problem till late 1950s, when ventilation of buildings was mainly catered to by opening and closing of windows. But as buildings began to be air conditioned, it demanded a sealed design. This then brought the need for mechanical ventilation systems to ventilate building interiors. These mechanical ventilation systems, brought with itself some choices such as percentage of air circulated, control on indoor temperatures and humidity and location of intakes and exhausts in buildings. It also brought with itself, the need for maintenance of these systems, choice of materials etc. The choices made in these factors influence in a direct way, the air quality in indoor spaces and poor choices have led to problems and building related illness or it has gone on to coin the word sick building syndrome. Sick building syndrome could create problematic symptoms to building occupants like dry eyes, block nose, dry throat, dry skin, headache, lethargy, watery eyes, runny nose, chest-tightness symptoms, flu-like symptoms, rashes etc. The National Institute for Occupational Safety and health (NIOSH) lists biological contamination as the third most serious indoor air health threat preceded only by poor ventilation and chemical contamination [8]. The most serious building related illness is hypersensitivity pneumonitis, which is an acute immune reaction where the lung alveoli are filled with and are damaged by inflammatory cells linked to the exposure from fungi and bacteria [9] [10].

Another indoor pollutant is moisture and this can catalyse a series of biological contamination sources. Uncontrolled moisture addition or un-regulated humidity in room air could not only cause thermal discomfort, but also be responsible for non-maintenance of indoor air quality. Increased humidity in a space is known to cause a spectrum of biological diseases. The undisclosed sources of indoor humidity pollution could be through infiltration or diffusion through building materials [11] or from stored humidity released from building materials of the room as detailed in Chapter 2. Inadequate dehumidification performed by an under sized air conditioning systems could also be one of the factors causing such problems. A study by Keiko showed that the increase in indoor relative humidity from 60% to 75.5% at favourable temperatures above 25°C caused the growth of xerophilic fungi in a rapid way, which could damage parts of the lungs [12]. Biological contamination concentrations in excess of 1000 CFU/m3 in indoor spaces indicate the need for investigation and possible remediation [13].

Most people discuss CO₂ as being the one big polluter to indoor air quality, but tests done show that indoor CO₂ concentrations will reach 5000ppm only when the ventilation rate is 1 L/s and 30,000 ppm when the ventilation rate is 0.2 L/s [14]. A number of peer review studies of occupant symptoms show that there is no significant relationship between the prevalence
of symptoms and CO₂ concentrations [15], but some indoor air quality investigators also associate indoor CO₂ concentrations from 600ppm to 1000 ppm or higher with perceptions of stuffiness, irritation and other discomfort [16]. So while CO₂ is used more as an indicator to check indoor air quality levels, it is more important to keep a check on concentrations of other indoor air pollutants, which could be more hazardous and life threatening, especially biological contamination. A study by Fang observed that the impact of temperature and humidity on the perception of air quality decreases with increasing air pollution levels, while the influence of pollution on perceived air quality decreases with increasing air temperature and humidity [17]. This study tries to realise how indoor biological contamination occurs and the mitigation techniques that can be used to avoid such problems. The BubbleZERO laboratory was used as a test bed to conduct the experiments listed in the following sections.

5.4.1. Experimental Methodology
The biological pollutant concentrations in the space were measured at a height of 1.5m from the floor, which represents the human breathing zone. Tab 5.3 gives information on the various instruments used in conducting these measurements along with their accuracy functions. Accuracy describes how close the measured value is to the true quantity; repeatability is the ability to obtain the same measured value under unchanged conditions, and resolution is the smallest increment which can be detected by an instrument. The biological measurements were done for a constant set-point dry bulb temperature and relative humidity. Spot measurements of biological count were performed at one location in each quarter of the laboratory as shown in the methodology of section 5.3. The biological measurements in the diffuser bowl were done inside 6 diffusers as shown in figure 6 of section 3.3. The bacteria plates were incubated at 35°C for 48 hours and the fungi plates were incubated at room temperature for 5 days.

Tab 5.3: Measuring instruments and their accuracy functions

<table>
<thead>
<tr>
<th>Parameter Measured</th>
<th>Instrument Used</th>
<th>Accuracy Functions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Biological Contamination (Bacteria and Fungi)</td>
<td>Agar plates and Anderson N6 Single Stage Bio aerosol Sampler</td>
<td>NA</td>
</tr>
<tr>
<td>Temperature, Relative Humidity, Dew Point</td>
<td>Sensirion Pin-type digital humidity and temperature sensors, SHT 75</td>
<td>Acc.: ± 0.25°C, ± 1.70%Relative Humidity, Res.: ± 0.01°C, ± 0.05%Relative Humidity, Rep.: ± 0.10°C, ± 0.10%Relative Humidity</td>
</tr>
</tbody>
</table>

5.4.2. Experimental Results
This section discusses the concentration levels of indoor biological contaminants such as bacteria and fungi for different operational scenarios. The indoor air quality is said to be good
if the concentration levels of these pollutants are below a threshold level of 500 CFU/m³ as specified by Singapore Std. SS 554 [18]. All operational combinations were having an average indoor dry bulb temperature of 24°C and 67% average relative humidity. The air velocity, which was mechanical induced using an external stand-alone fan, was measured to be 0.11 m/s. The metabolic rate in the indoor space varied from 1 to 1.1 (seated reading or seated typing criteria). The clothing value was 0.61 (trousers and long sleeve shirt).

The bacteria and fungi concentrations were measured using an Andersen’s sampler for different operational scenarios inside the laboratory, at locations outside the laboratory, and inside diffusers. An assumption that the air that was being drawn indoors through the ventilation units already had the possibility of having an inherent level of bacteria and fungi concentration in them before it enters the room through the underfloor diffuser was realised during this study. These inherent bacteria and fungi concentrations could be associated to the pollutants being added to air when it passes through heat exchangers, ducting and chilled water loops in the dehumidifying units. The moist environment at these places could have moss, bacteria and fungi growing on them as shown from Fig 5.13, and if maintenance is not regularly done, a significant amount of bacteria and fungi could be added to the outside air before it is supplied into the room. The indoor space itself could add again, bacteria and fungi concentration to the room from emissions of moist building materials. So biological concentration measurements done for different operational combinations have to take into consideration the addition of inherent contamination from all known sources before coming to a judgement on the contamination increase or decrease due to variation in occupancy or ventilation rate.

![Graph showing bacteria concentration varying with occupancy and ventilation rates](image_url)

Fig 5.10a: Bacteria concentration matrix for varying occupancy and ventilation rates; at the diffuser level and room level
From Fig 5.10a and Fig 5.10b, it can be observed that the bacteria and fungi concentration at the diffuser level, for any changes in ventilation rate is constant at zero occupancy, i.e. irrespective of the ventilation rate; there is a constant amount of bacteria and fungi that is being added to the outside air. Thus the average bacteria and fungi concentration level at the diffuser is 281 CFU/m$^3$ and 430 CFU/m$^3$. The bacteria and fungi concentration in the room however was averaged to be 352 CFU/m$^3$ and 495 CFU/m$^3$. This means that there could be an addition of bacteria and fungi to the air from emissions of moist building interiors.

The measured bacteria concentration levels with occupancy 1 and 2 were 495 CFU/m$^3$ and 702 CFU/m$^3$ at 11.1 L/s ventilation rate and the measured fungi concentration levels with occupancy 1 and 2 were 695 CFU/m$^3$ and 794 CFU/m$^3$ at 11.1 L/s ventilation respectively. When the ventilation rate was increased from 11.1 L/s to 16.7 L/s, the bacteria concentration reduced from 702 CFU/m$^3$ to 415 CFU/m$^3$ and the fungi concentration reduced from 794 CFU/m$^3$ to 594 CFU/m$^3$ at occupancy of 2. Similarly the bacteria concentration reduced from 676 CFU/m$^3$ to 423 CFU/m$^3$ and the fungi concentration reduced from 697 CFU/m$^3$ to 597 CFU/m$^3$ at occupancy 3 as ventilation rate was increased from 16.7 L/s to 22.2 L/s respectively. From these values though it could not be concluded that for occupancy 1,2 and 3 it is needed to have a ventilation of 11.1 L/s, 16.7 L/s and 22.2 L/s respectively for achieving conformance of indoor air quality with Singapore Std. SS 554 [17], which is much higher than the minimum ventilation rate required. From Fig 5.12a and Fig 5.12b, it will become clear how ventilation rates in a better and cleaner setup could be much lesser than what was achieved here.
Fig 5.11a: Indoor bacteria concentrations added from diffusers and room

Fig 5.11b: Indoor fungi concentrations added from diffusers and room

Fig 5.11a, Fig 5.11b shows that the bacteria and fungi concentration already inherent in air supplied at the diffuser level is 281 CFU/m$^3$ and 430 CFU/m$^3$. So the amount of bacteria and fungi concentration added to outside air between the outside and diffuser outlet is 210 CFU/m$^3$ 421 CFU/m$^3$ respectively. This gives us a very important conclusion to our assumption that the quantum of contamination that can occur in the ducts and heat exchanger setup is quite pertinent to determine the final concentration of biological pollutants in the room. This claim is also validated by pictures taken of ducts and heat exchanger components as shown in Fig 5.13. It can also be seen that the amount of bacteria and fungi concentration at the room level, without any occupancy is 352 CFU/m$^3$ and 495 CFU/m$^3$. So the bacteria and fungi added in the room air between the diffuser outlet and 1.5m from the room floor is 71 CFU/m$^3$ and 65 CFU/m$^3$ respectively. This shows that biological contamination can also occur from moist room materials which have a favourable condition for the growth of bacteria and fungi. So the total amount of bacteria and fungi concentration added inherent to the laboratory setup was 281 CFU/m$^3$ and 486 CFU/m$^3$. 

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140
Fig 5.12a: Bacteria concentration excluding additions from room interiors and ducting for various operational scenarios in occupancy and ventilation rate

Fig 5.12b: Fungi concentration excluding additions from room interiors and ducting for various operational scenarios in occupancy and ventilation rate

It can be observed from Fig 5.12a and Fig 5.12b that the elimination of inherent bacteria additions in a different setup that is much cleaner can restrict concentration of bacteria to 214 CFU/m$^3$, 421 CFU/m$^3$ and 395 CFU/m$^3$ for occupancy of 1, 2 and 3 at 11.1 L/s, 11.1 L/s and 16.7 L/s ventilation rate. Similarly the fungi concentration can be restricted to 209 CFU/m$^3$, 308 CFU/m$^3$ and 211 CFU/m$^3$ for occupancy of 1, 2 and 3 at 11.1 L/s, 11.1 L/s and 22.2 L/s ventilation rate. This shows that a decrease of 56.7%, 40% and 41.5% respectively from the measured value of bacteria concentration and 69.9%, 61.2% and 69.7% reduction in fungi concentration can occur with proper maintenance of the system. These percentages give a sense of how much the concentration of bacteria and fungi can be reduced if the inherent additions are removed through better construction and maintenance techniques. So ventilation rates need not be increased to satisfy indoor air quality requirements, but better maintenance
procedures need to be put in place. The results also show that ventilation of 5.5 L/s-person is enough to satisfy indoor air quality requirements if inherent addition of biological pollutants is taken care of by proper maintenance procedures.

From Fig 5.13 it can be observed that the reasons for biological contamination of outdoor air as explained in Fig 5.11a and Fig 5.11b occur from heat exchanger coils, filters and the ducting setup. The moisture in the outside air condenses when it comes in contact with the chilled water that is flowing through the chilled water coils. This moisture trickles down the coil fins and into the condensate basin below. This process causes a build-up of moisture bubbles in between the fins of the heat exchange. The air in this region is relatively cool and this cool and moist environment is a favourable condition for the growth of bacteria and fungi. The high rate of moisture transfer in this region also influences the filters in the region to become moist and this could be an additional source of biological growth and pollution. Fig 5.13 also shows a wet floor in the heat exchanger housing, filter units that are damp and cold and the duct setup with rust formation on the inside. All this could lead to the inherent biological pollutant’s addition into the air stream. Regular maintenance of these components could help solve this problem.

**5.5. Takeaway**

The takeaway from this chapter is that fluctuating outdoor conditions could have an influence on the consistency of the measured indoor conditions data. So outdoor environmental fluctuations must be studied and appropriate correction values to the indoor measured data need to be incorporated using the normalization model. Building managers must take care to
monitor indoor dew point temperatures to have a corresponding control of chilled water temperatures supplied to radiant panels. An increase in chilled water temperatures in panel pipes when the system is switched off overnight ensures a low rise in indoor air temperatures. The rate of decrease in indoor dry bulb temperature and indoor dew point temperature is 0.44°C/h and 0.32°C/h once the system is switched on. The study realised that many operational combinations of the radiant panel cum decentralized ventilation system is possible in the tropics to achieve indoor thermal comfort and indoor air quality. The upper limit for temperature of chilled water supplied to ventilation units is 12°C after which it becomes difficult to control condensation on radiant panels due to inadequate dehumidification achieved at the ventilation units. The ventilation rate cannot be decreased below 5.5 L/s-person keeping in view an increase in CO₂ concentration beyond safe limits. The research indicates that there is a significant amount of bacteria and fungi added to the outdoor air stream as it passes through the heat exchanger; filter cum ducting setup and from room materials. Such a situation can lead to unnecessary increase in ventilation rates to satisfy IAQ requirements. Regularly followed maintenance procedures of cleaning heat exchangers, replacing filters and fumigating ducting can prevent this problem and ventilation systems can function at reduced ventilation rates.
Chapter 6

Energy Demand
This chapter discusses the energy demand of the decentralized system. The study looks at the energy consumed by various components of the system and investigates areas where this demand can be reduced. Energy demand values for enhanced system designs are also determined through this study. An energy demand comparison of the decentralised system with a conventional building operating in the tropics is performed to look at possible increase in system energy efficiencies.

6.1. Energy Demand
Buildings account for 20% to 40% of a country’s total final energy consumption and its amount has been increasing at the rate 0.5% to 5% per annum in developed countries [1]. Air conditioning processes consumes most of this energy in buildings. The continuous talk and materialization of research and development in the field of mechanical and thermal engineering is therefore needed to reduce building energy demands in a significant way. Countries with abundant natural resources have less incentive for energy savings, but in Singapore, a place devoid of natural resources realises the cost of wasting energy. The need for more efficient use of energy is therefore greatly welcomed. The increasing percentage of air conditioned buildings in Singapore will only call for an increase in energy demand, and the long term sustainable growth of a country is only possible when efficient use of its energy is implemented. The potential for energy conservation in existing buildings of Singapore seems high when the distribution of energy performance of office buildings in Singapore is looked at, which varies from 100 kWh/m²-year to 469 kWh/m²-year [2]. Benchmarking studies conducted by Lee [2] examined all available independent building parameters influencing building energy consumption, including building height, age, storey height, occupancy rate and building system parameters, such as the chiller’s COP etc. Their correlation with the total building energy consumption based on the regression analysis of the collected data was investigated. Unfortunately, the results showed that there was no conclusive correlation drawn between total energy consumption and the building parameters except the gross floor area (GFA).

Most buildings in Singapore employ a central plant type cooling system that employs the conventional mixed ventilation concept with chillers supplying chilled water at 8°C to the Air Handling Unit (AHU). Radiant cooling processes though slowly being accepted as an alternative cooling strategy by the increasing adoption of chilled beams, chilled ceilings or radiant panels is still seen sceptically due to fears of condensation. The advantages of using radiant cooling techniques are many. It is generally believed that radiant systems have the advantage of lowering the pumping energy required to move heat in a water-based system in comparison to energy used to run fans in a central system. A study by Mumma predicted that the annual electricity energy consumption of a radiant system coupled with a Dedicated Outdoor Air System (DOAS) reduced energy consumption by 42% in comparison to a standard conventional VAV system [3]. Radiant cooling techniques have also been slated to provide 50% more heat transfer through radiation [4]. Although some theoretical studies of radiant slab cooling [5] and [6], and some field measurements [7] have been conducted, the
combined energy consumption and cooling capacity of operating buildings have seldom been reported.

Another area for reducing the energy demand in radiant cooling systems occurs due to delivery of higher chilled water temperatures to radiant ceiling panels. The temperature of chilled water delivered to the radiant systems are in the order of 15°C to 19°C in comparison to 6°C to 8°C water delivered in conventional air conditioning systems. This, in turn, allows the chiller evaporator temperature to rise and this improves the overall efficiency. Another advantage of such systems is that they tend to reduce the heat dissipated by ventilation fans within the conditioned space and the outdoor air volume required for cooling. Heat transfer through radiation directly cools occupants, which could allow slightly higher building air temperatures, hence decreasing building cooling loads. Overall, radiant ceilings reduce cooling energy consumption by 15% to 20% [8]. A study comparing the energy consumption of a conventional VAV system with a radiant ceiling cum DOAS found that the radiant ceiling with DOAS could realize annual blower electrical power reduction of 25% [8]. The DOAS enabled approximately a 20% reduction in outdoor air volume too [8]. Another study by Stetiu [9] estimated the HVAC energy savings in cold moist areas to be 17% to 42% [9]. The next section shows a study done to estimate the time needed for condensation to occur on radiant panels supplied at different chilled water temperatures. A note on the sensible load and latent load split at different supply chilled water temperatures to radiant panels is also discussed.

6.2. Energy Split between Sensible and Latent Load

A New Approach to Cool Buildings in the Tropics: Low Exergy Mechanism

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Keywords: Tropical Climate, Exergy, High Performance Building, Sustainability, Energy

1. Introduction

Air Conditioning systems consume most of the building’s energy in the tropics. The general practice to cool spaces is by using large amounts of air with low density and low heat capacity. We explore low exergy cooling mechanisms which reduce dependence on air as a temperature control medium, and focus on increasing building performance. Energy savings can be enhanced if the latent demand is separated from the sensible demand. A system using this concept has been implemented for the heating case in Switzerland, and a similar cooling system is being studied in Singapore as a part of the research at the Future Cities Laboratory.
A decentralized system operates wherein the latent demand is handled by an independent ventilation system and the sensible demand is satisfied using a radiant cooling system. The challenge is the risk of condensation. We try to analyse if this new mechanism can overcome some of these challenges.

2. Materials and Methods

An experimental laboratory Bubble ZERO (Zero Emission Research Operation) was setup to study the decentralized system’s operation in Singapore. The laboratory’s design used two 20-foot containers to represent a work space of 6m x 5m x 2.8m when the containers are combined.

The same radiant panel system as in Zurich was used to provide the sensible cooling through a process of radiation and convection. Radiant panels usually activate the thermal mass of the structure using it as additional cooling surface area; to test the extended impact of such a phenomena concrete was poured into one part of the ceiling and floor. Additionally the concrete floor also demonstrates how the decentralized ventilation system could be integrated into the building’s structure.

Singapore’s building standards limit the Envelope Thermal Transfer Value (ETTV) to 50 W/m² for the façade surface in air conditioned non-residential buildings (Building Construction Authority 2008) with 40 W/m² as the limit for the highest level Green Mark certification (Building Construction Authority 2010). Test facades using M glass which gets its name from the spectral transmittance curve and having a selectivity function greater than 2 (I. Mack, 2009) were used after shipping from Zurich. The glazing has a U-value of 0.9 W/m²K. Internal gains for four occupants occupying 30m² area were considered at 80W/person, the total equipment load was taken as 20 W/m² and outdoor air intake was quantified at 30m³/hr. The use of radiant panel integrated LED lighting eliminated lighting loads as the panel’s design has a central shaft which exhausts the heat of the LED directly to the outside. These data were used to estimate the cooling load for our experimental space (Meggers, 2012).

3. Results and Discussion

We consider the operation with a dew point of 18°C that could be used without risking direct condensation. A 5 kW custom chiller was used with the decentralized ventilation system and the radiant cooling panels, as shown in Fig 1.
A theoretical calculation for the cooling energy demand of the system for 4 people is given in Table 1.

Table 1: Cooling energy demand (Meggers, 2012)

<table>
<thead>
<tr>
<th>Cooling coil / dew point</th>
<th>Latent load</th>
<th>Sensible load</th>
</tr>
</thead>
<tbody>
<tr>
<td>8°C / 10°C</td>
<td>2 kW</td>
<td>3 kW</td>
</tr>
<tr>
<td>12°C / 14°C</td>
<td>1.6 kW</td>
<td>3.4 kW</td>
</tr>
<tr>
<td>16°C / 18°C</td>
<td>1.2 kW</td>
<td>3.8 kW</td>
</tr>
</tbody>
</table>

The decentralized system was operational for the first time in the tropics in January 2012 and initial results of time taken for condensation to occur on radiant panels were noted and summarised in Table 2.

Table 2: Time taken for condensation

<table>
<thead>
<tr>
<th>Dew point</th>
<th>Time taken (min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>16°C</td>
<td>15</td>
</tr>
<tr>
<td>18°C</td>
<td>30</td>
</tr>
<tr>
<td>20°C</td>
<td>Unseen (&gt; 60 min)</td>
</tr>
</tbody>
</table>

We were able to achieve 70.39% relative humidity at 21.92°C as air box supply conditions. The room conditions however were 30°C and 61% relative humidity due to the fact that the critical mass would take more time to cool down.

4. Conclusions
The use of radiant cooling systems has not become popular due to the risk of condensation. We see from our results that high temperature supply temperatures can help prevent condensation, but if the system is equipped with positive pressurization and low infiltration
then these systems can be used to provide the same human comfort but at a lower operational energy consumption. The alternative mechanism of using a decentralized system divides the energy needs for the latent and sensible demands. Low temperature cooling may be needed to dehumidify, but high temperature cooling can be carried out using radiant systems for over half the loads as shown in Table 1. Development of new integrated low exergy systems with radiant panels, ventilation supply, CO₂ optimized exhaust channels will lower energy demand. Exergy demand is minimized using large temperature lifts thereby reducing the energy losses. This directly translates to an enhanced performance.

As the laboratory arrived in September 2011, at the time this work is presented, the physical setup was completed and preliminary experiments were taking place. We anticipate further tests to be conducted through June 2012 whose results will be presented at the conference. Our goal is to continue to reduce the primary energy demand for operation such that the supply of renewable energy can meet the demand economically, and we will achieve our overall goal of zero emission operation.

5. References
Baldini, L. and H. Leibundgut, 2005, Increasing the effectiveness of building ventilation systems through use of local waste air extraction. CLIMA 2005. Lausanne, Switzerland, REHVA.
6.3. Energy Analysis of BubbleZERO

This section analysis the energy demand of the decentralised system installed in the BubbleZERO along with any enhancements that could be made to make the system more efficient. The energy demand of the system and of individual components installed in the BubbleZERO was calculated for different supply temperatures of chilled water to the panels and ventilation units. The same was also calculated at different ventilation rates. The results obtained were compared for COP and kW/ton values for three different and possible operational combinations. The three different combinations studied were the existing setup of the ventilation units, chillers and radiant panels; for an enhanced system setup of the radiant panels, which has an increased capacity as described in Fig 4.6 with a higher compressor COP; and the third case was for a high performance system using high performance compressors. Finally the COP and kW/ton values of the different setups were compared with a standard office building using conventional central plant systems in Singapore. Fig 6.1a and Fig 6.1b shows the amount of electrical power consumed by the existing chillers at different supply chilled water temperatures to ventilation units and to radiant panels. The old chiller in further discussions would represent the radiant panel chiller and the new chiller would represent the ventilation units’ chiller. Electrical power meters were installed on different electrical circuits of the BubbleZERO to measure the electrical power demand of each component of the chiller. The electrical power meters installed on the old chiller’s electrical circuits measured separately the electrical power demand of the compressor, pumps and chiller. The power consumed by electronics as shown in Fig 6.1a was calculated as a difference between the total chiller electrical power minus the compressor electrical power. The electrical power meters installed on the new chiller’s electrical circuits measured separately the electrical power demand of the chiller, ventilation units and pumps. The compressor power of the new chiller was determined from the compressor manufacturer’s catalogue. The supply and return chilled water temperatures and flow rate of water were measured using installed temperature and flow meters at the manifold junction as shown in Fig 1.8b.

Fig 6.1a: Electrical power consumed by the old chiller at different supply temperatures of chilled water to radiant panels
Fig 6.1b: Electrical power consumed by the new chiller at different supply temperatures of chilled water to ventilation units

From Fig 6.1a and Fig 6.1b it is clear that the chiller electrical power demand decreases with increase in supply chilled water temperatures. The decrease in chiller electrical power demand is 16.55% and 9.2% with every 2°C and 1°C increase in chilled water temperatures supplied to ventilation units and radiant panels respectively. Tab 6.1a and Tab 6.1b shows the component distribution of electrical power demand for the radiant panel chiller and ventilation units’ chiller. Both compressors in the chiller were reciprocating type and were working at a COP of 2 as determined by the chiller manufacturer’s catalogue.

Tab 6.1a: Distribution of electrical power demand among different components of the old chiller in the existing setup of BubbleZERO

<table>
<thead>
<tr>
<th>Old chiller</th>
<th>Electrical power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>709.9</td>
</tr>
<tr>
<td>Pumps</td>
<td>24.8</td>
</tr>
<tr>
<td>Electronics</td>
<td>39.5</td>
</tr>
<tr>
<td>Chiller Total Power</td>
<td>749.4</td>
</tr>
<tr>
<td>Total Power</td>
<td>774.2</td>
</tr>
</tbody>
</table>

0.77 kW

Tab 6.1b: Distribution of electrical power demand among different components of the new chiller in the existing setup of BubbleZERO

<table>
<thead>
<tr>
<th>New chiller</th>
<th>Electrical power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ventilation Units</td>
<td>97.3</td>
</tr>
<tr>
<td>Pumps</td>
<td>108</td>
</tr>
<tr>
<td>Compressor</td>
<td>4576</td>
</tr>
<tr>
<td>Chiller pump, Fan and Other Electronics</td>
<td>1668.8</td>
</tr>
<tr>
<td>Chiller Total Power</td>
<td>6244.8</td>
</tr>
<tr>
<td>Total Power</td>
<td>6450.1</td>
</tr>
</tbody>
</table>

6.45 kW
From Tab 6.1a it can be seen that the compressor in the old chiller consumes 94.72% and 91.69% of the chiller electrical power and total electrical power respectively. In the ventilation units system, the compressor electrical power is 73.27% and 70.94% of the chiller electrical power and total electrical power. So this shows that the compressor is the heart of the chiller system, demanding the maximum electrical power and an enhancement of the compressor system would improve system performances as realised in Fig 6.2a.

Tab 6.2a: Cooling capacity of the radiant panels at different supply temperatures of chilled water for the existing setup of the BubbleZERO

<table>
<thead>
<tr>
<th>Panel water supply temperature (°C)</th>
<th>Panel cooling capacity - Existing design of both panels (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15.5</td>
<td>740</td>
</tr>
<tr>
<td>16.5</td>
<td>800</td>
</tr>
<tr>
<td>17.5</td>
<td>740</td>
</tr>
<tr>
<td>18.5</td>
<td>740</td>
</tr>
</tbody>
</table>

Tab 6.2b: Cooling capacity of ventilation units at different supply temperatures of chilled water and at different airflow volumes for the existing setup of the BubbleZERO

<table>
<thead>
<tr>
<th>Ventilation units water supply temperature (°C)</th>
<th>Ventilation units cooling capacity (W)</th>
<th>Ventilation units’ airflow volume (%)</th>
<th>Ventilation units cooling capacity (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>3920</td>
<td>40</td>
<td>3.72</td>
</tr>
<tr>
<td>10</td>
<td>3840</td>
<td>60</td>
<td>3.48</td>
</tr>
<tr>
<td>12</td>
<td>3720</td>
<td>80</td>
<td>3.16</td>
</tr>
<tr>
<td></td>
<td></td>
<td>100</td>
<td>2.96</td>
</tr>
</tbody>
</table>

Tab 6.2a shows the variation in cooling capacity of the radiant panel system at different supply temperatures of chilled water to radiant panels. Tab 6.2b shows the cooling capacity of ventilation units at different supply airflow volumes and at different supply chilled water temperatures to ventilation units for the existing system installed at the BubbleZERO. These values were used to estimate the COP values as shown in Fig 6.2a, Fig 6.2b and Fig 6.2c.

If the compressors as discussed in Tab 6.1a and Tab 6.1b are changed to a more efficient one, having a COP of 2.45 for the radiant panel systems and 2.5 for the ventilation units systems, the electrical power demand will vary accordingly as shown in Tab 6.3a and Tab 6.3b. The compressors were selected from catalogues of industry manufacturers based on the required cooling capacities handled by the radiant panels and ventilation units as determined in Tab 6.2a and Tab 6.2b. The radiant panel system in this study was also assumed to be at an enhanced cooling capacity as enunciated in Chapter 4.
Tab 6.3a: Distribution of electrical power demand among different components of the old chiller in the enhanced setup of the BubbleZERO

<table>
<thead>
<tr>
<th>Old chiller</th>
<th>Electrical power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>244</td>
</tr>
<tr>
<td>Pumps</td>
<td>24.8</td>
</tr>
<tr>
<td>Electronics</td>
<td>39.5</td>
</tr>
<tr>
<td>Chiller Total Power</td>
<td>283.5</td>
</tr>
<tr>
<td>Total Power</td>
<td>308.3</td>
</tr>
</tbody>
</table>

0.31 kW

Tab 6.3b: Distribution of electrical power demand among different components of the new chiller in the enhanced setup of the BubbleZERO

<table>
<thead>
<tr>
<th>New chiller</th>
<th>Electrical power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ventilation Units</td>
<td>97.3</td>
</tr>
<tr>
<td>Pumps</td>
<td>108</td>
</tr>
<tr>
<td>Compressor</td>
<td>1691</td>
</tr>
<tr>
<td>Chiller pump, Fan and Other Electronics</td>
<td>578</td>
</tr>
<tr>
<td>Chiller Total Power</td>
<td>2269</td>
</tr>
<tr>
<td>Total Power</td>
<td>2474.3</td>
</tr>
</tbody>
</table>

2.47 kW

With the enhanced setup, the compressor would consume 86.07% of chiller electrical power and 79.14% of total electrical power in the radiant panel system. In the ventilation units system the compressor electrical power demand will change to 74.53% of chiller electrical power and 68.34% of total electrical power. Tab 6.4 shows how the electrical power consumption of the chiller would vary with respect to different supply chilled water temperatures to radiant panels and ventilation units in the enhanced setup. The data in this table has been plotted by scaling down the measured values shown in Fig 6.1a and Fig 6.1b considering the ratio of COPs in the existing setup and enhanced setup.

Tab 6.4: Radiant panels and ventilation units’ electrical power demand for varying supply temperatures of chilled water and airflow volumes in the enhanced setup of the BubbleZERO

<table>
<thead>
<tr>
<th>Chilled water supplied to ventilation units (°C)</th>
<th>Electrical power of new chiller (W)</th>
<th>Chilled water supplied to radiant panels (°C)</th>
<th>Electrical power of old chiller (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>5100</td>
<td>15.5</td>
<td>706.4</td>
</tr>
<tr>
<td>10</td>
<td>4272</td>
<td>16.5</td>
<td>624</td>
</tr>
<tr>
<td>12</td>
<td>3548.8</td>
<td>17.5</td>
<td>571.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>18.5</td>
<td>528</td>
</tr>
</tbody>
</table>

Now if a hypothetical high performance system with a compressor having a COP of 6 is considered; the electrical power demand situation could change accordingly as shown in Tab 6.5a and Tab 6.5b.
Tab 6.5a: Distribution of electrical power demand among different components of the old chiller in the high performance setup of BubbleZERO

<table>
<thead>
<tr>
<th>Old chiller</th>
<th>Electrical power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>100</td>
</tr>
<tr>
<td>Pumps</td>
<td>24.8</td>
</tr>
<tr>
<td>Electronics</td>
<td>39.5</td>
</tr>
<tr>
<td>Chiller Total Power</td>
<td>139.5</td>
</tr>
<tr>
<td>Total Power</td>
<td>164.3</td>
</tr>
</tbody>
</table>

0.16 kW

Tab 6.5b: Distribution of electrical power demand among different components of the new chiller in the high performance setup of BubbleZERO

<table>
<thead>
<tr>
<th>New chiller</th>
<th>Electrical power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ventilation Units</td>
<td>97.3</td>
</tr>
<tr>
<td>Pumps</td>
<td>108</td>
</tr>
<tr>
<td>Compressor</td>
<td>700</td>
</tr>
<tr>
<td>Chiller pump, Fan and Other Electronics</td>
<td>578</td>
</tr>
<tr>
<td>Chiller Total Power</td>
<td>1278</td>
</tr>
<tr>
<td>Total Power</td>
<td>1483.3</td>
</tr>
</tbody>
</table>

1.48 kW

The compressor electrical power demand in the high performance setup will change to 71.68% of chiller electrical power and 60.86% of total electrical power in the radiant panel system and 54.77% of chiller electrical power and 47.19% of total electrical power in the ventilation units system. Tab 6.6 shows the electrical power demand of the chiller at different supply chilled water temperatures to the radiant panels and ventilation units in the high performance setup.

Tab 6.6: Radiant panels and ventilation units’ electrical power demand for varying supply temperatures of chilled water and airflow volumes in the high performance setup of the BubbleZERO

<table>
<thead>
<tr>
<th>Chilled water supplied to ventilation units (°C)</th>
<th>Electrical power of new chiller (W)</th>
<th>Chilled water supplied to radiant panels (°C)</th>
<th>Electrical power of old chiller (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>1593.75</td>
<td>15.5</td>
<td>220.75</td>
</tr>
<tr>
<td>10</td>
<td>1335</td>
<td>16.5</td>
<td>195</td>
</tr>
<tr>
<td>12</td>
<td>1109</td>
<td>17.5</td>
<td>178.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>18.5</td>
<td>165</td>
</tr>
</tbody>
</table>
The COP of the system as described through Fig 6.2a, Fig 6.2b and Fig 6.2c is calculated as the ratio of cooling capacities in kW over electrical power demand in kW. The kW/ton values of energy efficiency are calculated as the ratio of electrical power demand in kW over cooling capacities in tonnes of refrigeration. The COPs and kW/ton values for the existing setup was calculated using data presented through Fig 6.1a, Fig 6.1b, Tab 6.2a and Tab 6.2b. The enhanced setup COPs and kW/ton values were calculated using the data presented through Tab 6.2a, Tab 6.2b and Tab 6.4. The high performance setup COPs and kW/ton values were calculated using data presented in Tab 6.2a, Tab 6.2b and Tab 6.6.

Fig 6.2a, Fig 6.2b and Fig 6.2c show the COPs of radiant panel system and ventilation system for different operational scenarios of the existing setup, enhanced setup and the high performance setup in the BubbleZERO.

![Graph showing COPs of radiant panels](image)

**Fig 6.2a:** Comparison of COPs between the existing setup, enhanced setup and high performance setup of the old chiller for different supply chilled water temperatures to radiant panels.
Fig 6.2b: Comparison of COPs between the existing setup, enhanced setup and high performance setup of the new chiller for different supply chilled water temperatures to ventilation units.

From Fig 6.2a, Fig 6.2b and Fig 6.2c it can be seen that the COP of the radiant panel system and ventilation units increases with increase in supply temperatures of chilled water to radiant panels and ventilation units respectively. The COP of the ventilation system however decreases with increase in ventilation rate. The COP of a high performance system employing a high performance compressor increases by 4 times to the system installed currently at the laboratory. The issue is that a high performance compressor having a COP of 6 is seldom available for low temperature lift systems.
A comparison of the above results with buildings employing conventional central air-conditioning systems are made next. The energy demand of a typical office building in Singapore was estimated out of literature studies. A sample room space from the UWC Tampanies building resembling the same size as that of the BubbleZERO was extracted [10] and the cooling capacity of the space was determined from BMS data. Since UWC Tampanies has a very high efficient system, the chiller energy demand wouldn’t represent a typical office building in Singapore, so 4 typical office buildings from Singapore; namely Ocean Financial Centre, Tampanies Grande, Fuji Xerox Towers and The National Library Building were selected from the research done by UWC, Tampanies [10] to ascertain the energy consumption of typical office buildings in Singapore. The power consumed by typical office buildings in Singapore is averaged to be 153 kWh/m²–year from research performed in [10]. The technical details of the typical office building energy demand are given below.

Area: 32.5 m²
Cooling capacity: 30.3 kW or 8.65 ton

\( \text{kW/ton} \) 1.42
\( \text{COP} \) 2.45

Table 6.7 gives the summary of COP and kW/ton values of the different systems in Singapore as discussed before.

<table>
<thead>
<tr>
<th></th>
<th>COP</th>
<th>kW/kW</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Radiant system</td>
<td>Ventilation system</td>
</tr>
<tr>
<td>Existing system in the BubbleZERO</td>
<td>1.1</td>
<td>0.83</td>
</tr>
<tr>
<td>Enhanced system in the BubbleZERO</td>
<td>1.4</td>
<td>1.04</td>
</tr>
<tr>
<td>High performance system in the BubbleZERO</td>
<td>4.5</td>
<td>3.35</td>
</tr>
<tr>
<td>Typical office in Singapore</td>
<td>NA</td>
<td>2.45</td>
</tr>
</tbody>
</table>

The best of compressors are not currently installed in the BubbleZERO, so making a comparison between efficiencies of the installed system in the laboratory and a typical central system is not reasonable. Compressors handling low temperature lifts and for small capacities if available to form a high performing setup could increase COPs to 3.93 from 2.45 in conventional systems installed in buildings, which is an efficiency increase of 60%.
6.4. Takeaway
The takeaway from this chapter is that the compressor plays a large role in determining the energy demand and COPs of chiller systems. In decentralized systems, the electrical power needed to supply pumps and fans are greatly reduced in comparison to a conventional central plant system. The study also reveals that higher COPs can be attained with higher chilled water temperatures, but cooling capacities of radiant systems must be high to achieve an overall high COP. It is seen that a decentralized system in comparison to a conventional system can perform better and can be more efficient, but the efficiency is highly dependent on compressor efficiencies.
Chapter 7
Ventilation of Indoor Spaces
This chapter discusses the various techniques used to ventilate buildings. A paper published in the International Sustainable Development Research Conference discusses the conventional ventilation techniques and how demand control ventilation can possibly reduce the amount of ventilation. A novel CO\(_2\) adsorption system that can be used to satisfy indoor ventilation requirements is also discussed with its pro and cons. Concerns of increase in concentration of other indoor air pollutants with the use of such an adsorption machine is also investigated through this study.

7.1. Different Ventilation Techniques

Buildings nowadays are designed to be sealed and such designs would require an effective ventilation system to maintain indoor air quality. Ventilation usually is the process where the exhaust air from air conditioned buildings is mixed with a percentage of outside air, and after undergoing filtration and conditioning; the air is re-supplied into the indoor space. In some other systems employing the concept of Dedicated Outdoor Air Supply (DOAS), only outdoor air is used to ventilate buildings, thus eliminating recirculation. Such a process also increases energy demand needed to condition a larger amount of outside air. In temperate climates, where humidity levels are relatively low, ventilation energy involves mainly cooling or heating of outside air during summers and winters respectively. But in the tropics, where humidity levels are comparatively higher, the ventilation process would additionally involve the process of dehumidification, a more energy intensive process. Various standards have been framed to indicate the ventilation rate for an indoor space based on the activity in the space. The ventilation rate is usually calculated on the basis of occupancy in the room or on the basis of floor area of the room or a combination of both. Singapore Std. SS 554 [1] mentions the minimum ventilation rate of an office space to be 5.5 L/s-person. There have been various debates occurring continuously in the industry on reducing ventilation rates, as it is seen to have a direct impact on reducing energy demand of buildings, but often such ideas are faced with stiff resistance from Indoor Air Quality (IAQ) experts citing the need to maintain adequate ventilation from a health point of view. A study by Seppanen showed the large extent of energy consumed by ventilation in heating, cooling, dehumidifying and humidifying of air [2]. Reducing the supply rate of outdoor air can reduce energy consumption, but it has to be limited to a point where the gaseous pollutants like Carbon – di – oxide (CO\(_2\)), Carbon Monoxide (CO), Formaldehyde (HCHO), Total Volatile Organic Compounds (TVOCs), dust, particulate matter (particulate size of \(\mu_{10}\) and \(\mu_{2.5}\)) and biological contamination (bacteria and fungi primarily) are within their permissible concentrations. Therefore, the ventilation rates designed for buildings should maintain a balance between energy consumption and ventilation for known benefits of comfort and health [3].

Air filtration can be done in 2 ways, one by diluting the polluted air or return air with unpolluted air or Treated Fresh Air (TFA) as performed in conventional ventilation techniques or by capturing air pollutants by specialized filters through methods of adsorption, absorption or disintegration of pollutants. Many works in the literature describe filters (e.g. HEPA H14, carbon filter, G3/4 pre-filter and F7/8 fine-filter) that can capture most pollutants such as VOCs, dust and odours etc. [4] [5], but none of these capture CO\(_2\), which could be a hazardous pollutant after certain concentrations.
Climeworks, a spinoff company of the Swiss Federal Institute of Technology (ETH, Zurich) has recently developed a novel CO$_2$ capture device that can be integrated with an Air Handling Unit (AHU) to efficiently capture CO$_2$ from air. Research from the use of this device in combination with filters claim to increase IAQ; reduce outdoor air supply for ventilation and save cooling energy demand of buildings [6]. The CO$_2$ capture technology used in this process has a cyclic adsorption of CO$_2$ and desorption of CO$_2$ process on a patented celluloid sorbent material [6]. The two main chemical reactions that take place on the solid sorbent material are as follows:

Under dry conditions, solely carbonates are formed according to equation 1.

$$2(RNH_2) + CO_2 \leftrightarrow RNHCO_2^- + RNH_3^+$$……………eq.1

Under humid conditions, additional bicarbonates are formed according to the equation 2.

$$RNH_2 + CO_2 + H_2O \leftrightarrow RNH_3^+ + HCO_3^-$$……………eq. 2

In the adsorption process, the CO$_2$ in air is chemically bound to the sorbent’s surface. Once the sorbent is saturated, the CO$_2$ is desorbed by heating the sorbent to 95°C, thereby delivering output air with lower levels of CO$_2$ as the main product and high-purity gaseous CO$_2$ as a by-product that can be used for other purposes. The CO$_2$-free sorbent can be re-used for many adsorptions and desorption cycles.

CO$_2$ concentration in outdoor air can be presently averaged at 390 ppm, which is about 100 to 400 times lower than that contained in flue gases [3]. The theoretically minimum work (in the general understanding of thermodynamic work) required to separate CO$_2$ from an air stream, as given by the Gibbs free energy change of un-mixing, is only a factor of about 3 larger than that required to separate it from flue gas streams [3]. The most crucial aspect of air capture is the large mass flow rate of air per mass of CO$_2$ captured, which practically excludes any approach that requires heating, cooling or pressurizing of air because of the associated energy penalty. Also in contrast to liquid scrubbing systems, solid sorbent materials for CO$_2$ adsorption [7] [8] offer reduced thermal mass and resistance to corrosion [9]. Among them, amine-functionalized materials with high specific surface area are promising for air capture because of their high selectivity at low concentrations [10], tolerance to moisture due to the chemical rather than physical nature of the sorbent–adsorbate interaction [11] and stability [12].

CO$_2$ itself is not such a dangerous indoor contaminant and many techniques have been developed to monitor CO$_2$. But other gaseous contaminants like CO, HCHO, and TVOCs etc. can cause more harm to the human body. The reason CO$_2$ is used as an indicator for IAQ in comparison to other pollutant concentrations is because of its relative inexpensiveness to measure. Demand control ventilation techniques that have been developed can be another mechanism to maintain the balance between energy demand and ventilation in traditional ventilation systems. In a research done by Mysen [13], 157 Norwegian classrooms were analysed for energy use over different ventilation systems: Constant Air Volume (CAV), CO$_2$ sensor based demand-controlled system (DCV-CO$_2$) and infrared occupancy sensor
based demand-controlled system (DCV-IR). Their results showed that DCV-CO$_2$ and DCV-IR reduced the energy use due to ventilation in the average classroom by 38% and 51%, respectively, in comparison to the corresponding energy for a CAV system [13]. So designers are already developing control strategies for reducing ventilation to building interiors, but a completely different strategy of using a CO$_2$ adsorption technique to ventilate buildings could be useful in the long run to even eliminate ventilation. The next section describes the mathematical concepts used in conventional ventilation techniques and what improvements are plausible in such techniques.

7.2. Conventional Ventilation Mechanism

Low Exergy approach to maintain Indoor Air Quality in tropical buildings

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Keywords: CO$_2$, Emissions, Tropical Climate, Exergy, Indoor Air Quality

Abstract

More than a third of the primary energy generated in the world is consumed by buildings for their sustenance using processes like heating, cooling, lighting and other appliances. This primary energy mainly generated by burning of fossil fuels is responsible for a large percentage of global CO$_2$ emissions and consequently the problem of climate change. The understanding, importance and materialisation of the concept of exergy which focuses on the quality of an energy source could address this problem and mitigate CO$_2$ emissions. Exergy based building design looks at incorporating Low-Ex systems to specific building processes and optimise energy flow. One way is to separate latent demand from the sensible demand. A system using this concept has been implemented for the heating case in Switzerland, and a similar system is being created in Singapore as a part of the research at Future Cities Laboratory to manage the cooling case.

A traditional air conditioning design employs fixed air changes per hour to remove indoor pollutants of which CO$_2$ is the most discussed. But if this removal process is performed in a different way i.e. by adsorption or decomposition or another process, the dependence on heavy air changes and the energy associated with it will be reduced. Although some minimum air changes would be required to address issues of other indoor air pollutants. The added advantage is the associated reduction in the dehumidification load which is also critical to the energy consumption of the building.
Background & Introduction

Singapore experiences three types of climates all around the year. As Singapore is an island located almost on the equator it experiences a hot and humid climate with usually hot daytimes and unpredictable rains. The winter environment however is experienced inside its buildings. The main reason for this type of a situation is the psychometric process employed to air condition buildings in Singapore. Over cooling is performed to remove humidity from Singapore air, but this process is not coupled with the thermodynamic process of reheating due to regulations, leading to air of 18 °C – 19 °C being blown into the room. This new approach of low exergy building system design could be the answer to achieve higher performances and solve this problem. Exergy basically focuses on the quality of energy and the potential efficiency increase in energy utilization processes. E.g. in a boiler operation. The energy efficiency increase cannot be quantified but the increase in energy efficiency of the thermodynamic process is close to 100%. The exergy efficiency for the same process however is only about 8% which is comparatively very low. The reason for such a variation is that a large quantity of high quality energy is consumed to produce this low quantity of low quality energy. Exergy can be more clearly understood as the maximum work that can be extracted from an energy flow process primarily caused by the change in the state of systems. The exergy content in the flow expresses the quality of the energy source. Therefore evaluating thermodynamic performances and efficiencies using exergy analysis gives a better and clearer picture. This process largely involves the calculation of exergy of heat and cold, at a temperature below or above the reference temperature of the environment.

The strategy of mixing ventilation is adopted in traditional air conditioning systems. This process works on the concept of dilution of air and its impurities to achieve the space indoor air quality and thermal comfort. Such a system is generally referred to as an overhead air distribution system in the tropics and is usually ducted above a false ceiling. This type of an arrangement therefore demands a higher floor to ceiling height in the building. An under floor air distribution system on the other hand employs the concept of displacement ventilation, achieving thermal comfort through thermal stratification of the surrounding air. This type of a system would also require reduced ceiling to floor height due to the elimination of ducting.

Different leading standards like ASHRAE and Singapore Standard (SS) quote a definite quantity of outside air that must be supplied into the buildings based on the area of the room and the assumed number of people in the room. The aim is to remove indoor air pollutants like CO₂, VOCs, Bio effluents, dust etc.; CO₂ being the major one. Conditioning Singapore’s outdoor air means the employment of high energy consuming processes of dehumidification. To avoid this problem, we look at techniques to remove CO₂ in a more efficient way.
Methodology

Exergy
Rant coined the word Exergy and its concept in 1950. It was then used as a tool to optimize thermal power plants [6], but not long ago it has interested engineers to apply the same in the field of sustainable design and building technology [9][11]. Exergy can be better understood as a balance between entropy and energy; combination of the two laws of thermodynamics. It defines in a better way the potential of the system to produce useful work in a specific environmental condition. Exergy quantifies the net potential of a system as influenced by both quantity of energy available and the temperature (quality) available relative to the system’s surroundings. [4][5][11]. Energy as such is a combination of two elements; i.e. the exergy and the anergy. In mathematical expression it is E = Ex + A.

In general cold exergy can be calculated using this equation:

\[ \text{Ex} = Q^* \left(1 - \frac{T_0}{T}\right) \]  

where \( T_0 \) is the reference temperature, \( Q \) is the energy associated and \( T \) is the temperature of the system in observation.

CO₂ Extraction
Ventilating a space with outside air is done to limit air pollutants like CO₂, CO, SO₂, CH₄, VOCs, dust, biological organisms etc. within their respective threshold levels. The ventilation requirement is developed on this basis. The much discussed and important carbon-di-oxide demands most of the focus among all air impurities and is the cause for large volumes of air changes in ventilation designs. Such large air movements are responsible for major energy consumptions as the outside air needs to be cooled and dehumidified especially in the tropics. This causes a decrease in exergetic efficiency. To overcome this problem we would primarily need to couple ventilation requirement to the dynamic occupancy of the space. We would then need to look at more novel techniques like absorption or adsorption or dissolution etc. to remove the impurities or CO₂. By employing such a methodology the chance of reducing the outdoor air intake would significantly increase. Our focus can then be on removing the other pollutants like VOCs, Methane, Formaldehyde, bio effluents, dust etc. which could have a smaller impact on the overall outdoor air requirement to ventilate the space hence reducing primary energy use and increasing exergetic efficiency without compromising indoor air quality. Some such technologies have been used in the past but never in the building domain. The use of Scrubbers in submarines, solid oxide fuel cells in space crafts or chemical adsorbent based systems is an area of high level research. A solid fuel cell based system could have a CO₂ reduction potential of more than 30% compared to the conventional system [8]. This process converts chemical energy released from the reaction of air and a fuel mixture into electrical energy. Electro chemical reduction of oxygen molecules at the cathode of the fuel cell cause oxygen to diffuse and produce on the anode. These oxygen ions then
react with the fuel and release electrons which circulate through an electrical circuit providing the gaseous product of the reaction.

CO₂ scrubber technology has been extensively used in submarine operations and ventilation requirements since world-war 2. As submarines have to be under water for a long period of time, it becomes more important to concentrate our effort on indoor air quality. One of the processes used in scrubbers is the use of Mon ethanolamine (popularly called MEA), which can reduce the CO₂ levels in submarines to 0.5% or even as low as 0.2% of the pressure, which is within the safe levels; British findings proclaiming a continuous exposure to 1% pressure of CO₂ can change the pH of blood and interfere with the body’s ability to retain essential metabolic salts [7]. The chemical property in amine solutions to absorb and desorb CO₂ could be a possible solution to remove CO₂. The plant’s operations is based on the ability of cool amine solution to readily absorb CO₂, while the hot amine solution gives up most of its CO₂ content. The amine solution used in the scrubber consists of water, a chelating agent and monoethanolamine, HO–CH₂–CH₂–NH₂. When CO₂ is absorbed or desorbed in aqueous MEA solution two reactions takes place [3]

\[
2RNH₂ + CO₂ \rightleftharpoons RNHCOO⁻ + RNH₃⁺ \cdots \cdots \cdot \text{eq. 2} \\
RNHCOO⁻ + CO₂ + 2H₂O \rightleftharpoons 2HCO₃⁻ + RNH₃⁺ \cdots \cdots \cdot \text{eq. 3}
\]

Where R is HOCH₂–CH₂.
The MEA solution with higher CO₂ content is referred to as rich MEA solution and the solution containing lower amounts of CO₂ is referred to as lean MEA solution.

The property of CaO to absorb and desorb CO₂ is another area of investigation. Standard humid calcium oxide absorbs CO₂ and produces CaCO₃ as a by-product in an exothermic chemical reaction CaO + CO₂ → CaCO₃. If this heat is tapped, it can be used for other heating purposes or can also be redirected to act as reheat in the dehumidifying and conditioning process of the incoming air. The CaO can then be regenerated in an endothermic process later. The endothermic reaction is CaCO₃ → CaO+CO₂ which is also the regeneration process [1].

In another enzyme based CO₂ extraction process employed by NASA, an enzyme based liquid membrane bioreactor is designed for CO₂ capture. The concept of selectivity for CO₂ is high for particular enzyme quantities i.e. 1400:1 for CO₂ v/s N₂ and 866:1 for CO₂ v/s O₂ [13].This is a high selectivity function and can act as a good approach to extract CO₂. NASA has already tested this type of a reaction based CO₂ removal process in a 180 day spacecraft prototype where the measured values of CO₂ was 0.307kPa of pressure, the upper limit being 1.067 kPa as a safe concentration [13], but NASA’s own maximum allowable concentration being 0.709 kPa [12]. The American Society of Heating, Refrigerating, and Air Conditioning Engineers, ASHRAE, reports that people complain of stuffy air at CO₂ concentrations of 0.101 kPa [2].
Results and Discussion
An experimental laboratory called Bubble ZERO (Zero Emission Research Operation) was
setup in September 2011 in Singapore using two combined together twenty feet containers.
This lab will be used to study the low exergy system consisting of radiant panels,
decentralized ventilation systems and a customised chiller’s operation in Singapore. The
laboratory represents a work space of 6 m x 5 m x 2.8m. This project was started through
collaboration between ETH, Zurich and National Research Foundation, Singapore in Future
Cities Laboratory, Singapore. It is aimed to look at adaptation techniques and use of Low-Ex
building systems already incorporated in Zurich to achieve enhanced conditioning of air and
thermal comfort in Singapore.

Fig 1: Bubble ZERO research lab schematic

The present basis for designing ventilation requirements is by following a simple mass
balance of air equation [14]

\[ V_o = \frac{N}{C_s - C_o} \quad \text{eq. 4} \]

Where \( V_o \) = outdoor air flow rate per person;
\( N = \text{CO}_2 \) generation rate per person;
\( C_s = \text{CO}_2 \) concentration in the space;
\( C_o = \text{CO}_2 \) concentration in outdoor air.

In the present design formula, the dynamic nature of occupancy is not registered. Also the
focus is only on the mass balance of \( \text{CO}_2 \). It is very important that the volumetric balance of
the gases is considered. As outdoor air has a fixed amount of \( \text{CO}_2 \) already present in it, the
room \( \text{CO}_2 \) level is only affected by the occupant’s input of \( \text{CO}_2 \). This leads to the formulation
of a differential equation in order to study the relationship between mass balance and the
volumetric balance based on occupancy dynamism. Considering the mass balance of \( \text{CO}_2 
flow we have a generalized partial differential equation:

\[ \frac{dv}{dt}_{\text{out}} \cdot \frac{dm}{dv}_{\text{out}} + \frac{dN}{dt}_{\text{person}} \cdot \frac{dm}{dN}_{\text{person}} - \frac{dv}{dt}_{\text{ext}} \cdot \frac{dm}{dv}_{\text{ext}} = \frac{dm}{dt}_{\text{tot}} \quad \text{eq. 5} \]
where the subscript “out” represents the property in outdoor air (the volumetric variation and density), subscript “person” represents the amount of CO\(_2\) exhaled by people. The term \(\frac{dN}{dt}\) represents the variation in the number of people with time and \(\frac{dm}{dN}\) represents the variation in the amount of CO\(_2\) with occupancy. If we incorporate the approach of adsorption or absorption of CO\(_2\) in this process we come up with a third parameter which has to be subtracted from the initial equation as it is removed by this process. That is the amount measured in the exhaust or the safe limit for operation. \(\frac{dv}{dt}\) and \(\frac{dm}{dv}\) would represent the volume of air sample monitored in a time interval and the amount of CO\(_2\) that is present in that sampling volume respectively. If we need to maintain the CO\(_2\) within prescribed limits, the CO\(_2\) removal process will only happen when this criteria is satisfied;

\[
\left(\frac{dm}{dv}\right)_{\text{room}} > \left(\frac{dm}{dv}\right)_{\text{ext}} \quad \text{eq. 6}
\]

or when the amount of CO\(_2\) in the room exceeds the CO\(_2\) level in the exhaust channel. And the rate of CO\(_2\) removal is determined by this equation.

\[
\left(\frac{dm}{dv}\right)_{\text{room}} \cdot \left(\frac{dv}{dt}\right)_{\text{ext}} - \left(\frac{dm}{dv}\right)_{\text{out}} \cdot \left(\frac{dv}{dt}\right)_{\text{out}} \quad \text{eq. 7}
\]

When we look into the overall density of CO\(_2\) in the space, we can represent it in such an equation

\[
\left(\frac{dm}{dv}\right)_{\text{room}} = \left(\frac{dm}{dv}\right)_{\text{out}} + \text{RQ} \cdot \left(\frac{dm}{dN}\right)_{\text{out}} \cdot \frac{N}{V} \quad \text{eq. 8}
\]

N: number of people in the room and V is the volume of the room (m\(^3\))

where the density of CO\(_2\) in the room is dependent on the density of CO\(_2\) from outside air which is usually constant and the variation in the occupancy density and its associated CO\(_2\) mass. The term RQ would represent the respiratory co-efficient which is the volumetric ratio of CO\(_2\) produced to oxygen consumed.

CO\(_2\) extraction mechanisms will be investigated in greater detail during the course of my PhD to get quantifiable data. The impact, reducing outdoor air intake and in turn energy consumption with the employment of such an adsorber or absorber system will be studied. As some studies show the influence air movement in a room may have on enhancing thermal
comfort. The incorporation of such a system will have an associated decrease in air flow. So the impact of such a process on thermal sensation and thermal comfort will also be studied. CFD analysis will be performed further to see how the movement of CO₂ and H₂O takes place inside the space and how it influences air flow.

**Conclusion**
Maintaining indoor air quality of a space is very essential but there can be better ways of achieving the same using lesser energy and efficient processes. Considering dynamic occupant density, volumetric CO₂ variations and relationships between mass flow rate of CO₂ and density flows of CO₂, we can optimize the ventilation design process to a greater extent. Associated problems of low air movements, replenishment of adsorption or absorption materials and affinity of the gases towards the adsorption process are things that need more investigation. Such a process can also better control mechanisms. Coupling such a CO₂ removal mechanism with other low exergy technologies like the decentralized ventilation and radiant cooling systems can reduce further the energy consumption of the building and increasing exergetic efficiency which directly translates into enhanced performance. The successful use of this technique in other domains brings hope that this type of a process can also be incorporated into the building domain. It is though important to study what impact this process will have on other air impurities present in the room and how the reduction in outdoor air flow affects the removal of these impurities like VOCs, Methane, CO, SO₂, biological impurities, dust etc. It is also important that the impact on thermal comfort is unaffected. We anticipate further analysis and tests to be conducted in our research laboratory Bubble ZERO. Our goal is to continue to reduce the primary energy demand for operation such that the supply of renewable energy can meet the demand economically, and to achieve our overall goal of zero emission operations.

**References**
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7.3. Ventilation by CO₂ Adsorption

An experimental setup was established at the Climeworks laboratory in Zurich to test the effective removal of CO₂ from air using a CO₂ adsorption machine. Outside air was drawn into the inlet of the machine and passed through a cellulose membrane material, which adsorbs CO₂. There were 2 operational cycles that took place in the CO₂ adsorption process. First was the adsorption cycle, wherein the celluloid material adsorbed CO₂ till a saturation point. The saturated celluloid material was then put through a desorption cycle, wherein the celluloid material was heated to about 100°C. Heating of the celluloid material caused it to delink itself from the adsorbed CO₂, which was then exhausted out and the air was cleaned of CO₂. The amount of energy that was consumed during one cycle of adsorption and desorption was calculated to be 2000kWh per ton of adsorbed CO₂. The celluloid membrane at the laboratory could adsorb only 1kg of CO₂ in one adsorption cycle, so 2 kWh of energy was consumed per cycle of adsorption and desorption. Climeworks was also of the opinion that in a real building scenario the energy consumption could be estimated to be of the order 2500 kWh/ton of CO₂ adsorbed due to higher losses present in a real case scenario. The pressure drop over the adsorber structure during the adsorption process was 100 Pa. Fig 7.1 shows the concentration of CO₂ in the incoming and exiting air stream of the celluloid material. The CO₂ concentration measurements were done just before the air entered the cellulose material and just after the air exited the cellulose material. CO₂ concentrations were measured by Climeworks for about 3 hours of the adsorption cycle, after which an extrapolation of the nature of the graph was done to achieve the saturation time.
Fig 7.1: CO₂ concentrations of air at the inlet and exit of the celluloid material

From Fig 7.1 it is clear that the CO₂ concentration of air is reduced significantly by the celluloid material. As the celluloid material reaches CO₂ saturation, the CO₂ concentration of air at the outlet comes at par with the inlet CO₂ concentration. The material then needs to undergo a desorption process to remove the adsorbed CO₂ from the membrane material. It can also be seen that 1kg of CO₂ can be adsorbed in about 4 hours. Desorption process would take another 4 hours. So a cycle of adsorption and desorption would roughly take 8 hours.

An experiment was done in the BubbleZERO laboratory to measure the increase of other indoor air pollutants like CO, HCHO, TVOCs and particulate matter when the ventilation system was switched off. Four people occupied the laboratory during the experimentation. The rise of CO, HCHO, TVOC and particulate matter were measured using spot measurement techniques using instruments listed in Tab 1 in the methodology of section 5.3 for a period of 4 hours at intervals of 15 min. An average of 8 spot measurements done in the human zone (1.5m from the floor) in the four quarters of the laboratory as shown in Fig 2 in the methodology of section 5.3 was used to arrive at an average value.
Fig 7.2a: Rise in concentration of CO$_2$ with time in the BubbleZERO laboratory

Fig 7.2b: Rise in concentration of CO with time in the BubbleZERO laboratory
Fig 7.2c: Rise in concentration of HCHO with time in the BubbleZERO laboratory

Fig 7.2d: Rise in concentration of TVOCs with time in the BubbleZERO laboratory
Fig 7.2e: Rise in concentration of particulate matter (µ2.5 and µ10) with time in the BubbleZERO laboratory

Fig 7.2 (b-e) shows the rise of indoor air pollutants (CO, HCHO, TVOCs, particulate matter of µ2.5 and µ10) with time when no ventilation system is running and when the space is having 4 people. The CO₂ data that has been plotted in Fig 7.2a is taken as a hypothetical situation considering a working CO₂ adsorption machine installed at the BubbleZERO. Knowledge of the rate of CO₂ adsorbed by the machine from Fig 1 is used to estimate the probable decrease in CO₂ concentration levels in the BubbleZERO for this hypothetical scenario. The aim of the experiment was to ascertain if indoor air quality can be maintained when a CO₂ adsorption system coupled with the decentralized ventilation system is used to ventilate indoor spaces. The impact on energy demand is also discussed.

The operational scenario was such that air purification is performed by the CO₂ adsorption machine for the first 4 hours, during the machine’s adsorption cycle. During the desorption cycle of the machine, the decentralized ventilation system ventilates the space. If such a combination system is able to maintain indoor air quality of the space, then energy consumed by the decentralized ventilation system could be reduced by half as the ventilation system need not run for 4 hours, when the adsorption cycle of the CO₂ adsorption system is in progress. An 8 hour time period was chosen, as regular office working hours are usually 8 hours a day. However this type of a setup would only be energy efficient if the CO₂ adsorption machine consumed lesser energy than the energy consumed by the decentralized ventilation system in 4 hours. Also the CO₂ adsorption machine is a much simpler building system, which could allow optimizing building construction. Smaller buildings with smaller technical systems could perform better with coupled PV panels being outside the city transmitting the energy demand for the building.

It can be seen from Fig 7.2 (a-e) that even when the ventilation system is switched off during the adsorption cycle of the CO₂ adsorption machine, the indoor CO₂ concentration is
maintained within safe limits. All other indoor pollutants are also maintained within safe limits set by the Singapore Std. SS 554 [1]. The rise in concentration levels of carbon monoxide and particulate matter is not of much concern as there are no known sources of addition of these pollutants, but the rise of formaldehyde and TVOC can be concerning, especially due to the operation of computers inside the laboratory which can release TVOCs. HCHO can be released by the wood facades and plywood that make up the laboratory construction. The safe limit for TVOC concentration is 3ppm and in 4 hours the limit had already reached 1.455. Similarly the safe limit of formaldehyde is 0.1ppm and in 4 hours the concentration had already reached 0.068ppm. So the need for a coupled ventilation system that can be switched on for 4 hours during the desorption cycle of the adsorption machine was justified. But using air purifiers to remove formaldehyde and TVOC could further reduce the use of the ventilation system, this was not tested though during the experimentation.

The acceptance of this ventilation theory could still be challenged for control of biological pollutants contamination. Inherent high levels of biological contaminants in the BubbleZERO laboratory due to the nature of its construction as explained in Chapter 5 prevented the experimentation procedure from studying the influence of biological contaminants. Nevertheless using of HEPA filters and other air purification systems could restrict the rise of biological contamination too. The rate of decrease in oxygen levels was also not measured in this experimentation, and doing so would only strengthen further the stability and reliability of this type of a ventilation setup.

From chapter 6 it can be seen that the energy required to operate the four ventilation units installed at the BubbleZERO laboratory was 2.95 kWh for a period of 8 hours. The energy required to operate the CO₂ adsorption machine was 2kWh for a period of 8 hours. So a combination of CO₂ adsorption machine operating for 8 hours (1 cycle of adsorption and desorption) and ventilation system running for 4 hours could demand 3.475 kWh of energy, which is 17.79% more than operating the ventilation system for 8 hours. This though, does not account for the decrease in chiller energy consumption that could be achieved by the combination system due to reduced dehumidification load on the ventilation system during the adsorption cycle of the CO₂ adsorption machine.

If two CO₂ adsorption machines were used together or if the mass of the CO₂ celluloid adsorbent was increased to a point where the CO₂ adsorbent machines along with other air purifiers are responsible for providing complete air quality of the indoor space, then the need for external ventilation could be eliminated, but this is quite a debatable theory, which needs more concrete and rigorous experimentations to be done along with subjective measurement studies. If this is possible, then the liability of installing large dehumidification systems, which demand a lot of energy especially for conditioning tropical air, can be addressed more efficiently and buildings could behave like submarines or space crafts, where the influence of outdoor air would become irrelevant. The achievement of safe limits of oxygen concentrations though needs to be still studied for such systems.
7.4. Takeaway

This chapter has looked at the adaptability of CO$_2$ adsorbing machines to maintain indoor air quality in buildings. The possibility of translation of technology used to maintain indoor air quality in space crafts and submarines to buildings was carried out through some basic experimentation. The study showed that CO$_2$ concentration levels could be maintained within safe limits of IAQ using a CO$_2$ adsorbing machine. Eliminating ventilation for a known period of time, allowed other indoor air pollutants like CO, HCHO, TVOC and particulate matter concentrations to increase, but not beyond the safe limits specified by industry standards. The concentrations of formaldehyde and TVOCs showed a tendency to rise more steeply than other indoor pollutants. The experimentation showed that outdoor ventilation could be eliminated for 4 hours without pollutant concentrations rising above their safe limits, but a conventional ventilation system was needed to provide ventilation thereafter. The possibility of prolonging this time could be achieved by the use of air purifiers to arrest the rise of other indoor pollutants. From an energy efficiency viewpoint, the energy required in operating such a combination ventilation system is more than that required to operate only a decentralized ventilation system. The use of this technology although possible to ventilate buildings, needs introspection on decreasing energy demand in order to be accepted to ventilate buildings. The need to develop and monitor a more detailed human study and acceptance criteria for the use of such systems is required along with subjective perception analysis and acceptance surveys. Only then could such a ventilation mechanism be seen as an alternative to provide indoor air quality to buildings in future.
f. Conclusion & Future Research

It can be concluded that a combined radiant cooling and decentralized ventilation system can condition building interiors in the tropics for temperature and humidity without the fear of condensation. Condensation would only occur if the dew point temperature of air surrounding the radiant panels goes beyond the surface temperature of radiant panels. This could happen if the air tightness of buildings is not good enough to prevent infiltration, which can be responsible for the addition of moisture from building exteriors thus increasing the dew point temperature of indoor air. Positive pressurization of the indoor space is a solution that can prevent such a phenomenon in leaky buildings. Experiments also testify that the radiant panel’s operation is only feasible in the tropics when it is coupled with another dehumidification system that conditions outdoor air used to ventilate indoor spaces. Indoor humidity addition could also occur from moisture realised by building materials in the room. It is therefore important to consider the influence of stored humidity released from room interiors during the design of radiant panel systems.

This study showed that radiant cooling takes place using two thermodynamic processes; namely radiant heat transfer and convective heat transfer and the percentage share of cooling achieved by both mechanisms depend on the temperatures of room surfaces and temperatures of radiant panel surfaces. The ratio of cooling performed by convective heat transfer to radiant heat transfer increases with increase in supply chilled water temperature. Also a radiant panel’s cooling capacity is dependent on the uniformity of surface temperature of the radiant panel, which in turn depends on the number of chilled water pipe loops that exist on the panel. More the number of loops, more uniform are the surface temperatures of the panel surface and more will be the cooling capacity. It was found that the cooling capacity could increase by up to 24.7% by increasing the cooling coil length by 50%. Another investigation showed that, only a percentage of the total sensible load in the room was handled by radiant panels (about 24% to 34.7% with respect to this experimental setup), so a coupled ventilation system still needs to operate to take care of the remaining sensible load. It was found that fluctuating outdoor conditions have an influence on the consistency of the measured indoor conditions data and an outdoor environment normalization model that was developed in this study could be used to employ appropriate correction values to the indoor measured data. Increase in indoor temperature and humidity overnight when the system was turned off was not very significant in such decentralized systems as any associated increase in dry bulb temperature was consumed by stagnant water in the chilled water pipes, which acted as an thermal barrier for room air temperature. This was seen as a pleasant surprise for radiant systems contrary to popular belief that rooms could heat up overnight causing a longer purge out period during the next morning.

This research also realised that many operational combinations of the radiant panel system and decentralized ventilation system was feasible in the tropics conforming to local and international guidelines. The operational combinations were based on different operating
criteria of ventilation rate, supply chilled water temperatures to the panels and ventilation units and occupancy. The best operating condition achieved was for panel supply temperatures of 15°C and ventilation unit supply temperatures of 12°C and at a ventilation rate of 5.5 L/s-person. The indoor pollutant concentrations of CO, HCHO, TVOC, CO₂ and particulate concentration of PM₂.₅ and PM₁₀ were maintained in their respective safe limits specified by local and international standards for all feasible operational combinations. A scopic investigation of bacteria and fungi concentrations found that biological pollutants maybe added to the outdoor air stream as it passes through the heat exchanger; filter cum ducting setup and from building materials in the room. Usually an increase in biological concentrations would recommend an increase in ventilation rate to counter the same, but this study found that regularly followed maintenance procedures of cleaning heat exchangers, replacing filters and fumigating ducting could avoid such problems and restrict ventilation rates to a minimum specified by industry standards. It was also seen that stratification of air in rooms could be influenced by working radiant panels and perimeter heat gain from glazing. Under Floor Air Distribution (UFAD) systems could be exposed to experiencing category 2 leakages and care must be taken to seal floor and tile joints in a plenum supply UFAD system and duct joints in a ducted supply UFAD system. Ducted UFAD systems however could have relatively less leakage when compared to plenum supply UFAD systems. From energy demand considerations, this research showed that the compressor plays the largest role in determining the energy demand and COP of the chiller and hence the overall system. In a decentralized system, the power needed to operate pumps and fans could be miniscule in comparison to a conventional central plant system, but higher COPs could only be attained by using highly efficient compressors. Higher chilled water temperatures supplied to radiant panels, uniform panel surface temperatures that increase cooling capacity of the radiant systems etc. could only enhance the efficiency of the system.

The study found that a novel approach to ventilate building interiors, by incorporating a CO₂ adsorbing machine could maintain indoor air quality of the room within safe limits for 4 hours after which a conventional ventilation system would be needed to provide ventilation. The reason for such an occurrence is because other indoor air pollutants like CO, HCHO, TVOC and particulate matter concentrations tend to increase, especially formaldehyde and TVOCs whose concentrations have a tendency to rise more steeply than other indoor pollutants. The possibility of prolonging this time could be achieved by the use of air purifiers to arrest the rise of other indoor pollutants. But the energy required to operate such a combination ventilation system was more than that required to operate a conventional ventilation system. So although this technology could have a plausibility to ventilate buildings in future, a through introspection needs to be done on decreasing energy demand of such systems. The need to perform a subjective human response survey will further develop to realise the acceptance of such ventilation theories. Only then could such ventilation mechanisms, similar to that which is adopted in submarines and space crafts be seen to ventilate buildings. As a future research area, the use of liquid desiccant systems to dehumidify outdoor air is a promising avenue. If this is possible, then the need for the use of 8°C chilled water for dehumidification can be eliminated. The cooling of spaces can then be
performed by radiant panels and the dehumidification of outdoor air by desiccant systems. This could then act as a more efficient system in terms of exergy.
g. References

Chapter 1


Chapter 2


Chapter 3


**Chapter 4**


**Chapter 5**


**Chapter 6**


Chapter 7


h. Resume

I believe in working with dignity, integrity and sincerity to face the challenges put forth against me by the ever changing dynamic work atmosphere.

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Phone: +91 9742342880
Email: rupesh_iyengar@yahoo.com
Nationality: Indian
Age: 27 years

Education:

1. ETH Zurich, Switzerland (Sept 2011 – Oct 2014)

   Master of Science (Building Performance and Sustainability)
   CAP: 4.64 / 5 – Gold Medalist

3. Visveswaraiah Technological University, Belgaum, India (Sep 2005 – Jun 2009)
   Bachelor of Mechanical Engineering from BMS College of Engineering, Bangalore
   First Class with Distinction – 78.7 % (University Topper in Refrigeration and Air conditioning)

Work Experience:

1. Researcher, Singapore ETH Center, Singapore (Sept 2011 – Oct 2014)
   Setup of “BubbleZERO” – A million dollar thermal and IAQ laboratory project funded by the National Research Foundation, Singapore within the Future Cities Laboratory

2. Research Staff, ETH Zurich, Switzerland (May 2011 – Jul 2011)
   Performed Energy Modeling and CFD Modeling of BubbleZERO and studied Low Exergy systems

3. Graduate Student Researcher, NUS, Singapore (Sep 2010 – Apr 2011)
   Worked on a collaborative project with Concordia University – “Experimental investigation of cooling load dynamics and user acceptability evaluation of Under Floor Air Distribution (UFAD) systems in hot and humid climate”
4. **Services Consultants, Bangalore, India** (Jul 2009 – Jun 2010)
   Designed energy efficient MEP systems for green building projects and LEED compliant designs, tender making and site inspections

**Research Publications (First Author):**


3. The feasibility of performing high temperature radiant cooling in tropical buildings when coupled with a decentralized ventilation system, *HVAC & R Research Journal*, 2013

4. Decoding India’s first Net ZERO energy showroom: Puma Sustainable store, Bangalore, *APCBE Conference*, 2013, Manila, Philippines


6. High performance building design strategy to achieve resilience towards climate change, *Healthy Buildings Conference*, 2012, Brisbane, Australia

7. A new approach to cool building spaces and maintain indoor air quality in tropics, *Healthy Buildings Conference*, 2012, Brisbane, Australia

8. The win-win opportunities for optimizing building energy consumption and IEQ, *Healthy Buildings Conference*, 2012, Brisbane, Australia


**Research Publications (Second Author):**


4. The potential of low exergy building systems in the tropics - Prototype evaluation from the BubbleZERO in Singapore, IAQVEC Conference, 2013, Prague, Czech Republic

**Awards and Honorary Positions:**

1. **2014 YEA award for exceptional service to ASHRAE, Seattle, USA**
   Presented at the ASHRAE Annual Conference for YEA related work in South – East Asia

2. **2013 Emerson Cup Award, Chennai, India**
   Project titled “Puma Sustainable Store, Bangalore” was judged as the best new building design

3. **2013 Bry – Air Award, Mumbai, India**
   Project titled “Puma Sustainable Store, Bangalore” was judged as the most energy efficient new building design

4. **2013 ASHRAE Presidential Award of Excellence, Jakarta, Indonesia**
   Presented at the ASHRAE region 13 conference for student activities related work in South – East Asia

5. **2013 Leadership U Award, Denver, USA**
   Secured a travel grant to present at the ASHRAE Annual Conference

6. **2012 Chairperson @ Healthy Buildings Conference, Brisbane, Australia**
   Responsibility of session on “Sustainability and Green Buildings – Construction, Design and Material”

7. **2011 IHG Design Competition @ IGBBC Conference, Singapore**
   Project titled “Resilient to Climate Change” was judged as the second best design project in South – East Asia

8. **2010 Bry – Air Award, Delhi, India**
   Project titled “Design, Fabrication & Testing of a Grooved Heat Pipe” received a National Commendation Award

9. **2009 Best Outgoing Student Award, Bangalore, India**
   A commendation presented by the Mechanical Engineering Department @ BMS College of Engineering

10. **2006 American Alumni Association Award, Bangalore, India**
    Scholarship for best academic performance @ BMS College of Engineering
**Academic Involvement:**

1. Conducted a webinar on “Is greening buildings sustainable, moving beyond the green buzz”, 6th Apr 2013, BMSCE, Bangalore, India
2. Invited to give a technical talk on “ZERO emission building design research and Puma sustainable store design” at Mapua Institute of Technology, 7th Nov 2013, Manila, Phillipines
3. Invited to give a technical presentation on on “Is greening buildings sustainable, moving beyond the green buzz” at Ong-Ong Architects group, 9th Jan 2013, Singapore
4. Secured a travel grant to attend ASHRAE Annual Conf. 2014 at Seattle, USA
5. Attended a workshop on “Commissioning for LEED and Sustainable design”, Nov 1st and 2nd 2010 at BCA academy, Singapore
8. Attended ASHRAE annual conference on “Sustainability”, 25th to 29th June 2011, Montreal, Canada
10. Attended ASHRAE CRC and conference of Region 13, 18th to 20th Aug 2011, Hong Kong, China
15. Attended ASHRAE CRC and conference of Region 13, 15th to 17th Aug 2014, Taipei, Taiwan
17. Invited to visit George Fisher piping solutions facility at Schaffhausen, Switzerland in October 2008
18. Invited to visit Grundfos Pumps facility in Bjerringbro, Denmark in October 2009
19. Invited to visit Armstrong Pumps facility in Toronto, Canada in June 2011
20. Invited to visit Hitachi HVAC plant at Guangzhou, China in August 2008
21. Invited to visit Climaveneta R&D plant at Venice, Italy in June 2012

**Social Involvement / Achievements:**

1. YEA Regional Coordinator – ASHRAE Region 13, 2014-2017
2. Secretary for ASHRAE Singapore Chapter, 2014-2015
3. Chair, YEA for ASHRAE Singapore Chapter, 2013 - 2014
4. Chair, Student Activities for ASHRAE Singapore Chapter, 2012-2013
5. President of student chapter (2011) – ASHRAE NUS student branch, Singapore
7. President of student chapter (2007, 2008, 2009) - ISHRAE BMSCE student branch, Bangalore, India
8. Selected as a Flying (Pilot) of the Indian Air Force in the 4SSC (M) F (P) course in 2009
9. Co-Founded Index Workshop Inc. – A sustainable design company in India, April 2011.
11. Singapore coordinator BMSCE International Alumni Association since 2013
12. Received a certificate of appreciation and momento for being the organizing team leader in the venue and hospitality committee during ACREX 2008
13. Won three silver medals in VTU swimming competition in 2007 along with the overall runner-up championship in swimming
14. Won 3rd Place in Swimming at Inter faculty games held at National University at Singapore, 3rd Sept 2010
15. Performed ramp shows and modeled for various brands.
17. Black belt in Taekwondo-Recognized as champion in the all India Taekwondo competition organized by Indian Association of Martial Arts.
18. Performed a sky walk at 233mts at the Macau tower, Macau.
19. Completed a course in Astrology from Indian astrological foundation.
20. Can speak English, Hindi, Kannada, Tamil, Sanskrit, Marathi, German (elementary), Mandarin (elementary)

References:

1. Dr. Hansjürg Leibundgut (Supervisor)
   Professor, Chair of Building Systems, ETH Zurich
   Inst. f. Technologie in der Arch., HPZ G 36, John-von-Neumann-Weg 9, 8093 Zurich
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2. Dr. Chandra Sekhar (Co – Supervisor)
   Professor, Department of Building, National University of Singapore
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3. Daniel C. Pettway (Industry)
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