HUMIDITY CONTROL IN DIFFERENT BUILDING APPLICATIONS; RESTAURANT AND OPERATION THEATRE

MOHD SYAFIQ SYAZWAN BIN MUSTAFA

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Faculty of Mechanical and Manufacturing Engineering
Universiti Tun Hussein Onn Malaysia

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For my beloved mother and father
ACKNOWLEDGEMENT

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ABSTRACT

Air conditioning (AC) in tropical climate required dehumidification of air at a low dew point temperature to meet humidity standard. This increase the required cooling energy and heating energy is needed to raise the supply air temperature to meet the room’s design temperature. This research was carried out to study energy efficiency and indoor environment factor in two different applications that is restaurant and operation theatre (OT). A method was proposed to enhance the humidity control and energy efficiency in the AC system by applying psychrometric analysis based on actual measurement. Three (3) different systems were used in this study that consists of economizer damper, desiccant wheel and heat pipe heat exchanger (HPHX). The economizer damper analysed base on fresh air intake requirement for ventilation purposes on standard ASHRAE 55 (2010). Manufacturing software by Novelaire and HPC were used to perform the psychorometric analysis for desiccant wheel and HPHX. At the restaurant, the grand total loads (Qt) of the existing AC system were 296.1 kW as to meet temperature and humidity level requirement. The Qt was reduced to 153.8 kW for economizer damper, 170 kW with heating 60.38 kW for desiccant wheel and 178.3 kW with heating 4.1 kW for HPHX. In actual measurement for the OT, the existing AC system had a grand total load Qt of 90.1 kW with heating 5.6 kW and it did not meet the humidity level requirement. However, the AC system required Qt 95.9 kW with heating 12.2 kW as to meet the humidity level requirement for an OT. By using the HPHX in the system, a reduced Qt of 81.6 kW with heating 17.7 kW was achieved, where as a Qt of 100.8 kW with heating 39.9 kW was attained by utilizing the desiccant wheel, whilst maintaining the humidity level requirement. The economizer damper was not applicable for the OT because the OT requires 100% fresh air intake. As a conclusion, damper economizer was beneficial for energy efficiency in restaurants with reducing of 48% energy used and HPHX was beneficial for energy efficiency with reduced 9.4% in OT compared with existing AC system.
ABSTRAK

Sistem penyaman udara (AC) di dalam iklim tropika memerlukan udara dikeringkan di suhu titik embun yang rendah. Ia meningkatkan tenaga sejuk dan pemanasan diperlukan untuk meningkatkan semula suhu udara. Kajian ini dilakukan untuk mengkaji aspek kecekapan tenaga dan faktor persekitaran dalam pada aplikasi yang berbeza. Penyelidikan ini mencadangkan untuk meningkatkan kawalan kelembapan dan kecekapan tenaga sistem AC dengan penggunaan analisis psychrometric berdasarkan pengukuran yang sebenar. Tiga (3) sistem yang berbeza digunakan dalam kajian ini iaitu penjimat peredam, roda pengering dan penukar haba jenis paip (HPHX), kajian ini dijalankan ke atas dua (2) aplikasi yang berbeza di restoran dan dewan bedah (OT). Penjimat peredam menganalisis keperluan pengambilan udara segar untuk orang berdasarkan standard ASHRAE 55 (2010). Perisian perkilangan dari Novilaire dan HPC digunakan untuk psychrometric analysis bagi roda pengering dan HPHX. Di restoran, jumlah beban besar (Qt) sistem AC yang sedia ada adalah 296.1 kW bagi memenuhi keperluan suhu dan kelembapan. Qt telah dikurangkan kepada 153.8 kW bagi penjimat perendam, 170 kW dengan pemanasan 60.38 kW bagi roda pengering dan 178.3 kW dengan pemanasan 4.1 kW bagi HPHX. Sementara itu, sistem AC yang sedia ada pada OT mempunyai jumlah beban besar Qt 90.1 kW dengan pemanasan 5.6 kW tetapi tidak memenuhi keperluan tahap kelembapan. Sistem AC memerlukan Qt 95.9 kW dengan pemanasan 12.2 kW untuk memenuhi keperluan tahap suhu dan kelembapan. Dengan menggunakan HPHX, Qt dikurangkan kepada 81.6 kW dengan pemanasan 17.7 kW, dan Qt roda pengering adalah 100.8 kW dengan pemanasan 39.9 kW. Peredam penjimat tidak boleh diaplikasikan bagi OT kerana OT memerlukan 100% pengambilan udara segar. Kesimpulannya, peredam penjimat memberi manfaat untuk kecekapan tenaga di restoran dengan pengurangan sebanyak 48% dari tenaga yang digunakan dan HPHX memberi manfaat untuk kecekapan tenaga dengan pengurangan sebanyak 9.4% pada OT berbanding dengan sistem AC yang sedia ada.
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LIST OF SYMBOLS

\[ \Delta h \] - Differential enthalpy (kJ/kg)
\[ \Delta t \] - Differential temperature (°C)
\[ \Delta \omega \] - Differential humidity ratio (kg/kg)
\[ A \] - Area (m²)
\[ \dot{m} \] - Mass flow rate (kg/s)
\[ \text{RH} \] - Relative humidity (%)
\[ V \] - Volumetric flow rate (m³/s)
\[ \nu \] - Specific volume (m³/kg)
\[ v \] - Velocity (m/s)
\[ \omega \] - Humidity ratio (kg/kg)
LIST OF ABBREVIATION

AC - Air Conditioning
AHU - Air Handling Unit
ASHRAE - American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BAS - Building automation system
CLF - Cooling Load Factor for people
CO - Carbon Monoxide
CO₂ - Carbon Dioxide
ICOP - Industry Code of Practice
DF - Dehumidification fraction
DOSH - Department of Safety and Health Malaysia
EER - Energy efficiency ratio
FA - Fresh air
HPHX - Heat pipe heat exchanger
HVAC - Heating ventilation and air-conditioning
IAQ - Indoor air quality
MA - Mixing air
MVAC - Mechanical ventilation air conditioner
N - Number of people
OA - Outside air
OT - Operation Theatre
QL - Latent loads
QS - Sensible loads
Qt - Grand total loads
RA - Room air
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<td>SA</td>
<td>Supply air</td>
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<td>RSHF</td>
<td>Room sensible heat factor</td>
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<td>SMACNA</td>
<td>Sheet Metal and Air Conditioning Contractors' National Association</td>
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<td>T&amp;C</td>
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<td>VAV</td>
<td>Variable air volume</td>
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<td>VC+D</td>
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CHAPTER 1

INTRODUCTION

This chapter discusses the background of the research problem. It generally describes about the air conditioning (AC) systems and the importance of humidity control. This chapter also highlights the problem statement based on the background provided as well as the objectives, limitation and the significance of the study.

1.1 Background of Study

Air conditioning (AC) systems typically provides thermal comfort and good indoor air quality (IAQ) achieved by controlling the temperature, humidity level and cleanliness of the air distribution. Moreover, the design of the AC system involves calculation of peak cooling loads, specification of system, calculation of annual performance and calculation of cost (Sekhar and Tan, 2009). On the other hand, the cooling loads estimation is very important to ensure proper removal of building heat loads. Building heat loads can be simply defined as internal and external loads. External loads occur due to heat transfer between the building and its’ surroundings and it is affected by outdoor conditions whereas internal loads are contributed by occupants, lighting and appliances. In modern commercial buildings, the leading sources of cooling loads are internal loads (ASHRAE, 2009; Mcquiston, Parker and Spitler, 2011).

The AC system was designed with cooling coils and the ability to cool and dehumidify the air as well as to meet sensible and latent heat loads in typical
buildings or rooms. Sensible heat load arises from dry bulb temperature. The sensible loads gains directly to the conditioned space by conduction, convection and radiation. Latent loads arises from moisture generated either from internal sources or from outdoor air ventilation to maintain the IAQ requirement (Burdick, 2011). Generally, comfortable indoor temperature is between 23 to 26 degree Celsius (°C) and relative humidity (RH) level is 40% to 70% (Malaysia Standard MS1525, 2010).

1.2 Problem Statement

The AC system typically cools and dehumidifies the air to provide comfortable indoor environment by removing sensible (temperature) and latent (moisture) loads. When the latent loads are high due to ventilation, wet surface and occupants, dehumidifying leads to reheating in order to meet the required indoor temperature and humidity level for comfort and health. The fresh air ventilation present in the AC system functions to improve the IAQ in the building and in special rooms, such as operation theatres and isolation rooms, as it needs 100% fresh air ventilation, a requirement to prevent microorganisms production in the rooms. It is a challenge for an engineer to design an AC system with a large outside air intake in tropical climate. To be specific, it is a common case in Malaysia where the weather is warm and humid all year-around with the average daily temperature ranging from about 30°C to 35°C and relative humidity (RH) is about 70% to 90% (Kosravi Salman et al, 2010).

The humid air coming from the fresh air ventilation may increase the sensible and latent load corresponding to the cooling energy consumption. The cooling coil needs to cool down the air at a lower dew point temperature for moisture removal and additional heating is required to meet the temperature before it is supplied to the room. Improper analysis of the dehumidification process for the AC system may lead to high cooling energy and thus affecting the RH in indoor environment. Cooling coils in the AC system requires extra energy and this causes inefficiency in the system. In terms of IAQ, RH must follow the standards for health and comfort reasons. High RH may lead to uncomfortable and stuffy conditions, while low RH will lead to the occupants complaints such as dry nose, throat, eyes and skin (Lstiburek, 2002). High RH levels will also lead to mold and fungus growth and
affects the IAQ inside the building. These microorganisms spores travel by air and cause infectious diseases, allergies and other respiratory irritations to occupants (SMACNA, 1998). A major concern on energy conservation has led to the development of energy efficiency of the AC systems while improving indoor air quality in the buildings.

1.3 Objective of Study

This study concerns about the energy efficiency and indoor environment factor of the buildings or rooms that had been selected. The objectives are:

i. To investigate optional humidity control in tropical climates to meet the standard humidity level in two different latent heat loads applications.

ii. To compare cooling energy between the existing design and alternative supplementary design to maintain humidity standard requirement.

1.4 Scope of Study

The scope of the study began with the identification of high latent heat building or room. Latent loads are known as moisture loads that comes from outside air ventilation, humans, infiltration, equipment and appliances. Therefore, indoor monitoring was conducted in several rooms or buildings and was compared to Malaysian standard MS1525, Department of Safety and Health (DOSH) Malaysia and ASHRAE standard 55. Point of samplings followed the guideline from the Industry Code of Practice (COP) on Indoor Air Quality 2010, DOSH Malaysia. The minimum number of sampling is 1 point per 500 m². The indoor monitoring involved the air temperature, relative humidity (RH), carbon dioxide (CO) and carbon monoxide (CO₂) in the rooms that needed to be monitored. The information is required from the building owner in this study in order to understand the cooling load of the AC system that was installed include:

i. Technical data

ii. Testing and commissioning (T&C) report (if permitted)

iii. Design layout of the AC system

iv. Building automation system (BAS) daily (if any)
v. Civil and structure layout
vi. Control system
vii. Schedule maintenance
viii. Indoor air design requirement

The AC system performance was also monitored to identify the average cooling energy consumption in standard operations. The air flow rate was measured to obtain the actual total air flow rate of the AHU (field measurement). The ASHRAE recommendation for measuring air flow rate is at least 25 points for a rectangular duct (ASHRAE, 2009). The psychrometric chart was used to analyze the actual cooling energy required and to identify the latent and sensible loads respectively. The causes of both loads were rectified from the field test measurement for the AC system and indoor monitoring.

The improvement in dehumidification is a value added to the energy efficiency for the AC system and would further improve the indoor air quality in the building.

i. Economizer damper
   - Utilizing minimum fresh air intake results in huge energy savings in humid climate. The economizer damper can be useful in controlling outside air flow rate and to minimize the fresh air load.

ii. Desiccant Dehumidifier
    - Desiccant reacts by attracting and removing moisture from one stream to another and can be cost-effective because it uses low grade thermal sources to remove moisture.

iii. Heat pipe heat exchanger (HPHX).
    - HPHX is a high performance heat transfer device which works using evaporation and condensation sections of proper working fluid in individual closed tubes. Literature review on the studies, technology and application of desiccant and HPHX dehumidifier was conducted as supplements to improve the AC system.

The target area for this study was:

i. Restaurant
ii. Hospital Operation Theatre (OT)
1.5 Limitation of Study

The limitations of this study occurred due to the following components:

i. Gathering information at the selected buildings or rooms. Some information cannot be given to a third party because it is private and confidential to the engineering department of the selected buildings.

ii. Hot and humid climate play an important role in this study for the cooling and dehumidification process in the AC system. Due to the changes in the weather cannot be controlled.

1.6 Significance of Study

The comparisons of the proposed methods in the system are beneficial to HVAC consultants and engineers in order to improve the existing system and to properly design the air-conditioning system for energy efficiency. Three methods for improving humidity had widely been studied and used in industry which are for humidity improvement for good dehumidification, energy saving and healthy indoor air quality in a tropical climate buildings.
CHAPTER 2

LITERATURE REVIEW

In this part of the chapter, it will review the current status of the several studies that had been carried out on humidity control in different building applications in term of air-conditioning system, psychrometric chart overview, humidity control, humidity to indoor air quality (IAQ), desiccant dehumidifier technology that available, heat pipe heat exchanger (HPHX), the technology for dehumidification. Others operating conditions that can be used to optimize the performance of the humidity control in building system as well as additional relevant information of the study are taken as references.

2.1 Air-conditioning System

The Air-Conditioning (AC) system is a common system providing cooling and dehumidification process for typical buildings in a tropical climate. It comprises consists of the circulation air, exhaust and fresh air for air side and blower fan, motor, filter and cooling coil for mechanical components (Carrier, 1952).

Figure 2.1 shows the air side of the AC system which is known as the air handling unit (AHU). The cooling coil in the AHU handles the cooling and dehumidification process. The air contacts with the cooling coil surface and the moisture was condensate when the air is cooled below the dew point temperature. The air changed its state from water vapour to liquid water when it contacts with the cool surface. The liquid water is then drained away for disposal.
2.2 Psychrometric Chart Overview

Figure 2.2 shows the ASHRAE psychrometric chart under sea-level pressure of 101.325 kPa. This chart uses thermodynamic properties to analyze and illustrate the condition and processes involving moist properties. The problem of high latent load, which is known as a moisture problem, can be analyzed using this chart and solutions can be made to improve the moisture problem significantly. The following are some parameters of thermodynamics shown in the psychrometric chart (ASHRAE, 2009).

i. Line “A” is the line of dry bulb temperature. It is measured using dry bulb thermometer, with the common scale in Celsius (°C) and Fahrenheit (°F).

ii. Line “B” is the humidity ratio. Humidity ratio is the ratio of mass water vapour per mass dry air conditioned measured in the scale of gram/kilogram of dry air.

iii. Line “C” is the relative humidity which is defined as the ratio of the mole fraction of water vapour in a moist air sample to the mole fraction of water vapour in a saturated moist air sample at the same pressure and pressure. The unit of relative humidity is percentage (%).

iv. Line “D” is the specific volume of a moist air mixture and is expressed in the unit of mass of dry air (m³/kg).
v. Line “E” is the wet bulb temperature which is the saturation temperature of moist air at the end of an ideal adiabatic saturation process (S. A. Sherif, 2002).

vi. Line “F” is the moist air specific enthalpy (kJ/kg) where the mixture of perfect gases equals the sum of the individual partial enthalpies of the component.

vii. Line “G” is the dew point temperature where the moisture wills condensate out from the air. When the air reaches the dew point temperature, the dry and wet bulbs are exactly the same in 100% RH.

viii. Circle “F”, the protractor in the chart shows two scales. They are the sensible/total heat ratio and the ratio of enthalpy differential to humidity ratio differential.

![Figure 2.2: ASHRAE psychrometric chart (ASHRAE, 2009)](image_url)

The relationships between temperature, moisture content and energy can be identified using this chart. The parameters involved include dry bulb temperature,
wet bulb temperature, relative humidity, specific volume, humidity ratio, specific enthalpy and dew point temperature. If two parameters are known, other properties can be obtained from the chart. Cooling and dehumidification are typically the basic processes for an AC system used in tropical climates. Both processes are able to meet indoor temperature and relative humidity by supplying air at or below than the dew-point temperature of water vapour in air (Ghali, 2008). Psychrometric chart is also used to determine and analyze energy and thermal comfort in typical buildings. Table 2.1 shows how researchers used the psychrometric chart as a tool for evaluating energy and thermal comfort in the buildings.

<table>
<thead>
<tr>
<th>Researchers</th>
<th>Methods</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zhang (2006)</td>
<td>Used psychrometric chart process and mathematical model to determine annual energy requirement and compared four systems with different strategies for dehumidification process.</td>
</tr>
<tr>
<td>C. Koranteng et al. (2011)</td>
<td>Used psychometric chart to analyze thermal comfort performance in office buildings in Ghana.</td>
</tr>
<tr>
<td>Zhou and Cheng (2011)</td>
<td>Used psychrometric chart to analyze passive energy saving strategy and portfolio analysis in Chongqing, China.</td>
</tr>
<tr>
<td>Hafizal and Goto (2011)</td>
<td>Used psychrometric chart to analyze thermal comfort of three types of residential building in Malaysia and show high humidity for all three types building.</td>
</tr>
<tr>
<td>Aziz et al. (2011)</td>
<td>Used psychomeric chart to analyze the performance of air handling unit (AHU) for energy efficiency and to meet thermal comfort levels in lecture halls in Malaysia.</td>
</tr>
<tr>
<td>Tahbaz (2011)</td>
<td>Used psychrometric chart for introducing thermal comfort index in outdoor thermal conditions.</td>
</tr>
<tr>
<td>Ratlamwala, T. A. H., &amp; Dincer, I. (2013)</td>
<td>Used psychrometric chart to define energy and exergy for three different approached by increasing ambient temperature, heating with humidification and evaporative cooling.</td>
</tr>
<tr>
<td>Robert and Ezenwa (2014)</td>
<td>Used psychrometric chart to study of influence of an evaporative cooler on turbine inlet air and how turbine responds to reduced ambient temperature.</td>
</tr>
<tr>
<td>Lee et al (2015)</td>
<td>Used psychrometric chart to carry out theoretical study of the fog removal process in LNG Ambient Vaporizer. The LNG Ambient Vaporizer in Incheon area was used in our study. Vaporizer in Incheon area was used in our study.</td>
</tr>
</tbody>
</table>

### 2.2.1 Cooling and Dehumidification Process using Psychrometric Chart

Figure 2.3 illustrates the psychrometric chart cycle for cooling and dehumidification processes in a tropical climate. The outside air (OA) or fresh air (FA) conditions was identified by the frequencies of outdoor condition (outdoor climate) occurrence. This
is because the OA influences the cooling load estimation where heat flows from outdoors to indoors (high temperature to low temperature). The design condition of room air (RA) should comply with human comfort and health requirement. The room air (RA) design conditions in Malaysia are 22°C to 26°C and relative humidity is 40% to 70%. The mixing air (MA) must lie between the points OA and RA and is determined by the ratio of air flow rate. Equation 2.1, 2.2 and 2.3 shows the calculation to identify mixing air (ASHRAE Fundamental 2009).

\[ \omega_{MA} = \frac{\dot{m}_{PA}\omega_{PA} + \dot{m}_{RA}\omega_{RA}}{\dot{m}_{PA} + \dot{m}_{RA}} \]  
\[ \dot{m}_{PA} = \frac{V_{PA}}{v_{PA}} \text{ (in psychrometric chart)} \]  
\[ \dot{m}_{RA} = \frac{V_{RA}}{v_{RA}} \text{ (in psychrometric chart)} \]

Where \( \omega = \text{humidity ratio (g/kg)} \) and \( \dot{m} = \text{mass flow rate (kg/s)} \). \( V = \text{volume flow rate (m}^3\text{/s)} \) and \( v = \text{specific volume (m}^3\text{/kg)} \) as determined from the psychrometric chart.

Figure 2.3: Cooling and dehumidification processes of an AC system (ASHRAE, 2009)
Sensible and latent loads need to be determined in a cooling load estimation. Both of the loads are required in order to calculate room sensible heat factor (RSHF). The RSHF will show how cold and dry the supply temperature needs to be in order to meet the desired temperature and RH level in the building (ASHRAE, 2009). RSHF can be expressed using equation 2.4.

\[
\text{RSHF} = \frac{\text{room sensible heat load}}{\text{room total heat load}} \tag{2.4}
\]

The slope of RSHF was plotted on the protractor in psychrometric chart. The line passing through from RA and parallel to the RSHF line on the protractor was drawn to intersect with the saturation curve (100% RH). This is known as the apparatus dew point temperature (ADP) and the supply air must lie somewhere on the RSHF line. The coil by-pass factor need to be considered because a very small portion of the air passes through the coil without being cooled or bypassed and this is about 14% of the supply air (Haines and Myers, 2010). The total cooling energy can also be determined by taking the heat contents i.e. enthalpies from points MA and SA. After a complete air-conditioning cycle process, the actual sensible loads \((Q_S)\), latent loads \((Q_L)\) and total loads \((Q_T)\) can be expressed using equation 2.5, 2.6 and 2.7 (ASHRAE Fundamental 2009):

\[
Q_S = 1.21 \times \text{sensible heat factor} \times V \times \Delta t \tag{2.5}
\]

\[
Q_L = 3.01 \times \text{latent heat factor} \times V \times \Delta \omega \tag{2.6}
\]

\[
Q_T \text{ (total load)} = Q_S + Q_L \text{ or } 1.2 \times V \times \Delta h \tag{2.7}
\]

Where:

- \(V\) = Volumetric flow rate (m\(^3\)/s),
- \(\Delta t\) = Differential temperature (°C)
- \(\Delta \omega\) = Differential humidity ratio (kg/kg)
- \(\Delta h\) = Differential enthalpy (kJ/kg)
2.2.2 High Latent Loads Process in Psychrometric Chart

Estimation of cooling loads is the amount of energy needed or must be removed to meet the comfortable environment required in the building. Building loads can be simplified into internal and external load. External loads are mainly due to heat transfer between the building and its surrounding which is affected by outdoor conditions and internal loads are contributed by occupants, lighting and appliances. There are two types of load to be met under the AC system that are sensible and latent loads.

Table 2.2 summarizes the factors that influence the sensible and latent loads. Sensible heat load arises from dry bulb temperature heat transfer. Latent load arises from moisture generated either from internal sources or from outdoor air ventilation to maintain the IAQ requirement. Murphy (2006) advocated that the impact of geographical location and high outside air ventilation in humid climate may give high indoor relative humidity. Chen et al. (2004) believed high latent loads may influence roots and mold problem. It can be avoided by proper design and maintenance of the air-conditioning system. Chong and Low (2006) found the latent loads occur are due to weather conditions, occupants, wet surface and water leakages in the building.

<table>
<thead>
<tr>
<th>Latent heat load</th>
<th>Sensible heat load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air infiltration through crack, windows and doors</td>
<td>Solar heat gain through walls, roof and windows</td>
</tr>
<tr>
<td>Equipment and appliances</td>
<td>Equipment and appliances</td>
</tr>
<tr>
<td>Occupants</td>
<td>Lights</td>
</tr>
<tr>
<td>Ventilation</td>
<td>Infiltration (hot air)</td>
</tr>
<tr>
<td>Weather</td>
<td>Occupants</td>
</tr>
<tr>
<td></td>
<td>Ventilation</td>
</tr>
<tr>
<td>-</td>
<td>Weather</td>
</tr>
</tbody>
</table>

Figure 2.4 illustrates the air conditioning process in high latent loads buildings. This process requires very low dew point temperature to handle the latent loads. In this case, a heating system is required to be added. The low dew point temperature without reheat may lead to over cooling in the building. The
The recommended level of dew point for a typical office building is between 51°F and 57°F (11°C and 14°C), while the level for libraries and museums is between 46°F and 54°F (8°C and 12°C) (Saidi and Vazirifard, 2007).

![ASHRAE Psychrometric Chart](image)

Figure 2.4: AC system process in high latent loads building (Saidi and Vazirifard, 2007)

### 2.3 Humidity Control

The humidity level in the building is quantified by relative humidity (RH) where RH was the ratio of the amount of water vapour in the air to the amount of water vapour air can hold at that temperature (Elovitz, 1999; Mandayo et al., 2015; Standard, 2010).

Table 2.3 shows the recommended relative humidity parameter in the building. The range of RH is specified for occupied and unoccupied buildings. When the humidity level is below the range (40%), humidification is required where a
water source is used for the humidifying process (Halim and Man, 2006). The wet surface and dirt can cause the growth of microorganisms and result in poor indoor air quality (S.K.Wang, 2001). Humidification by injecting steam can be advantageous to avoid wet surface. While at high humidity levels (>60%), dehumidification process is necessary. The air is going cooled until the dew point to dehumidify the air.

Table 2.3: Recommended relative humidity parameter (55.1-2010, 2010; Department of Occupational Safety and Health Ministry of Human Resources)

<table>
<thead>
<tr>
<th>Standard</th>
<th>ASHRAE standard 55 - 2004</th>
<th>DOSH (ICOP IAQ) -2010</th>
<th>Malaysia Standard (MS 1525)-2007</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relative Humidity (RH)</td>
<td>40 – 60 %</td>
<td>40 - 70 %</td>
<td>40 - 70 %</td>
</tr>
</tbody>
</table>

2.3.1 Humidity Control System

Typically in tropical climates where the weather is warm and humid all year-around, dehumidification process is required for humidity control. Diekman et al. (2009) discussed three types of dehumidification approaches in typical AC systems.

Figure 2.5 illustrates the 3 approaches for dehumidification process in the AC system. The first approach (a) utilized cooling coils to cool and condense the water vapour. The second approach (b) utilized cooling and hot gas reheating to meet sensible and latent loads and the third approach (c) utilized heat pipe heat exchanger where heat pipe transporting pre-cooled and reheat for the AC system. Murphy (2006) studied the strategies for dehumidification process using the psychrometric chart based on space dry bulb temperature for constant volume. The strategies involved in rainy day, fan speed adjustment, face and bypass damper and variable air volume (VAV) systems. The VAV system is a more effective strategy to control humidity and if improperly designed and controlled may lead to high humidity level in a building space.
Chua et al. (2007), on the other hand, investigated the control strategies to meet humidity during part load condition. The comparison of control chilled water flow, bypass air, variable air volume, run-around coil and low face velocity was made. The result showed chilled water control strategy gave the highest indoor humidity while variable air volume (VAV) gave effective dehumidification performance. In addition, Sekhar and Tan (2009) investigated the indoor humidity control by modulating the chilled water flow rates. Performance evaluation of various coil selection was conducted at various operating conditions using simulation. It is shown by increasing the effective surface area of the coil and reducing air flows (VAV) to improve the dehumidifying performance.

Due to warm and humid climates, keeping it to a minimum and optimizing the fresh air intake may control indoor humidity level, indoor air quality and reduce energy in the building. Table 2.4 showed how the researcher optimized the fresh air intake to meet the indoor air quality (IAQ) requirement while reducing energy in the AC system. Thus, it benefits the IAQ and is an energy saving opportunity in typical buildings.
<table>
<thead>
<tr>
<th>Year</th>
<th>Authors</th>
<th>Method</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>1998</td>
<td>Harry J. Saver</td>
<td>Outside air economizer damper was modulated to maintain minimum fresh air intake using simulation by set point of supply temperature.</td>
<td>It shows that control of the economizer damper can save the amount of energy and comply to ASHRAE standard 62.1</td>
</tr>
<tr>
<td>1998</td>
<td>George J. Berbari</td>
<td>Outside air was controlled and additional run around coil was used to treat the outside air intake and analyzed potential energy saving.</td>
<td>It shows the run around coil resulted in energy saving in hot and humid climates and improved indoor relative humidity significantly.</td>
</tr>
<tr>
<td>2006</td>
<td>Gang Wang and Ming sheng Liu</td>
<td>Method to optimize the outside air intake to minimize total cost of mechanical cooling and steam humidification.</td>
<td>Optimal curve was made and optimized the outside intake attaining 27% energy saving.</td>
</tr>
<tr>
<td>2011</td>
<td>Joo et al.</td>
<td>Conducted optimum ratio of makeup and recirculation air in Korea to improve IAQ and energy saving. Consideration involved weather condition, mixing air and IAQ.</td>
<td>Result shows the ratio between make up to recirculation air was 1:2 and met the requirement for IAQ and reduced energy.</td>
</tr>
<tr>
<td>2012</td>
<td>Gang Wang and Li Song</td>
<td>Steady state energy consumption model was identified for air handling unit system (AHU) under economizer damper to meet temperature and humidity level in the building.</td>
<td>Optimal supply air temperature was evaluated based on modelling and resulted in energy saving up to 90% under certain space load. The economizer may be controlled using the temperature sensor.</td>
</tr>
<tr>
<td>2014</td>
<td>Ular Palmiste</td>
<td>Explored and calculated free cooling potential of direct airside economizer in Esonia by conducted temperature bin weather data</td>
<td>Free cooling presented in yearly, monthly and hourly. The highest airside economizer operation from October to April</td>
</tr>
<tr>
<td>2016</td>
<td>Baojie Mu et al.</td>
<td>evaluated of energy efficiency by experimental using performance of anti-windup extremum seeking control towards chilled water and direct expansion air side economizer</td>
<td>By controlled outdoor air damper, chilled water minimized the cooling coil load and direct-expansion system minimized power consumption.</td>
</tr>
</tbody>
</table>
Table 2.4: Optimized fresh air intake in AC system (continue)

<table>
<thead>
<tr>
<th>Year</th>
<th>Authors</th>
<th>Method</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>2016</td>
<td>Bin Yan et al.</td>
<td>Investigated the uncertainties of outdoor air control pattern and mapped to building performance simulation in term of energy consumption and ventilation requirement of a particular HVAC system type.</td>
<td>Simulated results show that the uncertainty in outdoor air control can lead to 17% difference in cooling consumption and 43% difference in heating consumption compared with using standard outdoor air control settings in Energy Plus simulation.</td>
</tr>
</tbody>
</table>

### 2.4 Humidity to Indoor Air Quality (IAQ)

Relative Humidity (RH) should meet the standard requirement for health and comfort reasons. RH is related to comfort, mold growth, respiration illness and impact to occupant’s performance and ability (Bayer et al, 2000; Choo et al, 2015).

Figure 2.6 illustrates the optimum relative humidity for every building for health and comfort reasons. The optimum RH range shown is 30% to 60%. Low RH in the building can cause occupants to experience dryness and irritation of eyes and airways (Bayer et al., 2000; Ismail and Deros, 2010). In high RH, the occupant may feel uncomfortable and stuffy (Lstiburek, 2002). Hamimah et al. (2010) highlighted that high RH gave small effect on thermal sensation but in long term will effect microbial growth. Morse et al. (2007) defined the causes of high humidity in the school. High humidity was found due to ventilation and infiltration air, oversized equipment and improper operation of the AC system. Due to high humidity, the symptom of mold growth was found during walk-around inspection. This study found the AC system has difficulty to maintain indoor humidity under the range due to the climate.
Figure 2.6: Optimum relative humidity (RH) range (Thammanoon et al, 2010).

Ghasemkhani and Naseri, (2008) monitored the IAQ for a kitchen in Tehran. They discovered that 90% of the people felt comfortable with the recommended range for temperature and RH. RH below 30% resulting in dry air affecting eyes, skin and mucous membrane while RH above 60% supported the growth of pathogens or allergens. Marzuki et al. (2010) did monitoring at schools in Malaysia and highlighted the range of RH did not comply with the standard range (refer Table 2.3) due to the hot and humid climate.

Additional effect from high RH is condensation. Condensation occurs when the air cools until the dew point and moisture condenses out from air. Condensation contributes to a variety of problem such as damaged wood, paper and fabric and steel can become rusty (Elovitz, 1999). High relative humidity and condensation may lead to microorganism growth such as mold and fungal. Moulds spores travel through the air and moulds growth and survives on surfaces are determined by micro environmental condition of both material and indoor space such as the high humidity, temperature and, radiation and light (Aibinu et al, 2009; Nematchoua et al, 2015).
Figure 2.7 illustrates the mold growth in hotels. Harriman, Lstiburek, & Kittler (2000) highlighted the American Hotel & Motel Association estimated its members spending more than USD 68 million each year dealing with mold growth. This microorganism problem causes infectious diseases, allergies such as rhinitis and asthma, pneumonias, cancers, immune disorders, mucous membrane inflammation and other respiratory irritations to occupants (SMACNA, 1998).

2.4.1 Health Care Buildings and Hospitals

Hospital buildings are complex indoor facilities with variable uses and functions. Therefore, the AC system should be used to keep good IAQ for patients and medical staff to meet comfort and health reasons. For surgery in operation theatres (OT), the requirement of fresh air intake is 100% and exhaust must also be 100% considering it is in a critical area (Leung and Chan, 2006).

Table 2.5 shows the recommended design criteria for various places in health care facilities. The room must also follow the required room pressure control. Room differential pressure is focused on air balancing, proper maintenance of the AC system and continuous checking of the filter. It is recommended to add the
differential room pressure monitoring (Leung and Chan, 2006). In tropical climates, the influence of humid air may increase the AC system energy that is required to remove moisture before it is supplied to the OT. Murphy (2006) discusses how the temperature and humidity requirements of surgery rooms impact the design of the HVAC system, and presents system alternatives that can meet both of these requirements. The design used the rule of thumb which hugely impacts to the energy and is unable to meet the requirements.

Table 2.5: Recommended design criteria in health care facilities (ASHRAE, 2007)

<table>
<thead>
<tr>
<th>Function Space</th>
<th>Pressure Relation to Adjacent Areas</th>
<th>Minimum Changes of Outside Air per Hour</th>
<th>Minimum Total Air Changes per Hour</th>
<th>All Air Recirculated within Room Units</th>
<th>Air Recirculated Within Room Units</th>
<th>Relative Humidity, %</th>
<th>Design Temperature, °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surgery and Critical Care</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Operating room (class B and Positive C surgical)</td>
<td>Positive</td>
<td>4</td>
<td>20</td>
<td>No</td>
<td>30 to 60</td>
<td>17 to 27</td>
<td></td>
</tr>
<tr>
<td>Operating/ surgical cystoscopic rooms²⁻⁴⁻¹</td>
<td>Positive</td>
<td>4</td>
<td>20</td>
<td>No</td>
<td>30 to 60</td>
<td>20 to 23</td>
<td></td>
</tr>
<tr>
<td>Delivery room²</td>
<td>Positive</td>
<td>4</td>
<td>20</td>
<td>No</td>
<td>30 to 60</td>
<td>20 to 23</td>
<td></td>
</tr>
<tr>
<td>Recovery room⁹</td>
<td>- *</td>
<td>2</td>
<td>6</td>
<td>-</td>
<td>30 to 60</td>
<td>24 ± 1</td>
<td></td>
</tr>
<tr>
<td>Critical or intensive care (burn or intermediate)</td>
<td>Positive⁴</td>
<td>2</td>
<td>6</td>
<td>-</td>
<td>30 to 60</td>
<td>21 to 24</td>
<td></td>
</tr>
<tr>
<td>Newborn intensive care</td>
<td>Positive⁵</td>
<td>2</td>
<td>6</td>
<td>-</td>
<td>30 to 60</td>
<td>22 to 26</td>
<td></td>
</tr>
<tr>
<td>Treatment room⁹</td>
<td>Positive⁶</td>
<td>2</td>
<td>6</td>
<td>-</td>
<td>30 to 60</td>
<td>21 to 24</td>
<td></td>
</tr>
<tr>
<td>Nursery suite</td>
<td>Positive</td>
<td>5</td>
<td>12</td>
<td>-</td>
<td>30 to 60</td>
<td>24 to 27</td>
<td></td>
</tr>
<tr>
<td>Trauma room¹⁻²</td>
<td>Positive</td>
<td>5</td>
<td>12</td>
<td>-</td>
<td>30 to 60</td>
<td>22 to 26</td>
<td></td>
</tr>
<tr>
<td>Trauma room (crisis or shock)</td>
<td>-</td>
<td>3</td>
<td>15</td>
<td>-</td>
<td>30 to 60</td>
<td>22 to 26</td>
<td></td>
</tr>
<tr>
<td>Anaesthesia gas storage</td>
<td>Negative</td>
<td>-</td>
<td>8</td>
<td>Yes</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>GI endoscopy</td>
<td>-</td>
<td>2</td>
<td>6</td>
<td>-</td>
<td>30 to 60</td>
<td>20 to 23</td>
<td></td>
</tr>
<tr>
<td>Bronchoscopy</td>
<td>Negative</td>
<td>2</td>
<td>12</td>
<td>Yes</td>
<td>No</td>
<td>30 to 60</td>
<td>20 to 23</td>
</tr>
<tr>
<td>Emergency waiting rooms</td>
<td>Negative</td>
<td>2</td>
<td>12</td>
<td>Yes</td>
<td>-</td>
<td>30 to 60</td>
<td>22 to 26</td>
</tr>
<tr>
<td>Triage areas</td>
<td>Negative</td>
<td>2</td>
<td>12</td>
<td>Yes</td>
<td>-</td>
<td>-</td>
<td>21 to 24</td>
</tr>
<tr>
<td>Radiology waiting rooms</td>
<td>Negative</td>
<td>2</td>
<td>12</td>
<td>Yes¹</td>
<td>-</td>
<td>-</td>
<td>21 to 24</td>
</tr>
<tr>
<td>Procedure room (class A surgical)</td>
<td>Positive</td>
<td>3</td>
<td>15</td>
<td>-</td>
<td>30 to 60</td>
<td>21 to 24</td>
<td></td>
</tr>
</tbody>
</table>

Table 2.6 depicts the result of the comparison that has been proposed. It is shown that the series of desiccant wheel system is energy saving and the humidity level in the surgery room. The desiccant system is able to supply the dried air without lowering the coil temperature. Basic calculation and psychrometric chart analysis was involved to analyze the system in order to meet the temperature and humidity level required in the surgery room.

Nunelly et al. (2004) investigated the design of humidity control inside the operation theatre. The proposed desiccant wheel system was introduced for the existing air handling unit (AHU). Some benefits include dried AHU and meeting the
low dew point up to 20°F. The addition of desiccant led to energy saving and the RH met the requirement in the operation theatre.

T Sookchaiya, Monyakul, and Thepa (2008) compared two strategies to control temperature and humidity in the hospital. Case 1 involved inverter control speed for the compressor motor of the AC system and the addition of ultrasonic humidifier along with an electrical heater and case 2 used a thermostat for controlling the compressor of the AC system without RH control. Case 1 showed better control strategies to meet the temperature and humidity as compared to case 2.

Table 2.6: Comparison of AC system and design (Murphy, 2006)

<table>
<thead>
<tr>
<th></th>
<th>Space RH</th>
<th>Cooling Capacity</th>
<th>Leaving Coil DB</th>
<th>Reheat Capacity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cool and Reheat (Single</td>
<td>60%</td>
<td>3 tons (10.6 kW)</td>
<td>48°F (9°C)</td>
<td>8,500 Btu/h (2.5 kW)</td>
</tr>
<tr>
<td>Cooling Coil</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cool and Reheat (Two</td>
<td>60%</td>
<td>3 tons (10.6 kW)</td>
<td></td>
<td>8,500 Btu/h (2.5 kW)</td>
</tr>
<tr>
<td>Cooling Coils In Series</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Upstream Cooling Coil</td>
<td></td>
<td>1.8 tons (6.3 kW)</td>
<td>55°F (13°C)</td>
<td></td>
</tr>
<tr>
<td>Downstream Cooling Coil</td>
<td></td>
<td>1.2 tons (4.2 kW)</td>
<td>48°F (9°C)</td>
<td></td>
</tr>
<tr>
<td>Series Desiccant Wheel</td>
<td>55%</td>
<td>2.1 tons (7.4 kW)</td>
<td>52°F (11°C)</td>
<td>0 Btu/h (0 kW)</td>
</tr>
</tbody>
</table>

2.5 Desiccant Dehumidifier Technology

The desiccant is a chemical dehumidification that may be in either solid or liquid types. There are 3 basic concepts for desiccant dehumidifier which are sorption, desorption and cooling. Figure 2.8 shows the process of the desiccant dehumidifier. The process starts in point 1, where the desiccant is in low surface vapour pressure compared with the air pressure. Desiccant is now in cool and dry form and attracting the moisture in the air. At Point 2, the desiccant vapour pressure increases and equalizes to the surrounding air pressure. Desiccant cannot attract any more moisture because there are no difference in the pressure between the surface and the vapours in the air. Desiccants need to regenerate by increasing the desiccant vapours pressure. Therefore, heating is required. At Point 3, desiccant is in dry form and the vapours pressure is too high to collect moisture in the air.

To restore the desiccant to low vapours pressure, the desiccant needs to be cooled and it will continue being circulated in the dehumidification system. There are
many types of desiccant dehumidifier such as liquid spray tower, solid packed tower, rotating tray, rotating honeycomb and etc. After comparing all the desiccant dehumidifiers, the chosen one is the rotating honeycomb-desiccant wheel because it may reach high flow rates and is low cost compared with the others (Harriman, 2002). Figure 2.9 shows the rotating Honeycombe (desiccant wheel).

Figure 2.8: Desiccant dehumidification process (Harriman, 2002)

Figure 2.9: Honeycombe desiccant (Harriman, 2002)
2.5.1 Energy Saving using Desiccant Dehumidification

Several studies have been conducted using desiccant wheel in several places and countries. Charles (2000) presented the cycle using a desiccant to move moisture within the air handler system of an AC system. The moisture captured by the desiccant from the saturated air leaving the cooling coil (supply) and delivered to the inlet duct (return). Some of the moisture absconds with the exhaust air but most of it goes back to the coil. The cooling coils will cool the air back and condense as a moisture removal. This system may reduce 16.4% of energy usage and increases the moisture removal. Another researcher, Bellia and Mazzei (2000) examined the AC system with desiccant wheel on an Italian climate. For retail store applications energy saving obtained was 22% and in theatres was between 23% and 38% when compared with the conventional AC system.

While Dhar and Singh (2001) studied desiccant hybrid AC system and compared the performance of four hybrid cycle including a new proposed cycle for typical hot-dry and hot-humid weather conditions. By lowering heating consumption and lowering compressor work, energy savings up to 10.63% was possible. Mazzei, Minichiello and Palma (2002) investigated the summer operating cost between hybrid desiccant and traditional system using three software codes based on Italian weather conditions. Operation savings up to 35% was obtained and thermal cooling was reduced by 52% compared to traditional systems.

In addition, Jia, Dai, Wu, and Wang (2006) studied and experimented between hybrid desiccant air conditioning system and vapour compression system. The hybrid desiccant air-conditioning economized 37.5% electricity power and increased evaporator sensible heat factor. Capozzoli, Mazzei, Minichiello and Palma (2006) compared a conventional heating ventilation and air-conditioning (HVAC) system to a hybrid system with chemical dehumidification for supermarket application. Their study used the simulation and analyzed annual operating cost and found that the payback period to be about one year. Moreover, operating cost savings is about 30% compared with existing HVAC systems in Italian supermarkets.

Ghali (2008) had simulated the transient performance of hybrid desiccant vapours compression system for Beirut ambient conditions. The study compared that with conventional vapours compression system in the entire cooling season. The
annual running savings for hybrid system was USD 418.39 and payback was achieved in less than five (5) years. Sayegh, Hammad and Faraa (2011) evaluated and compared two methods of improving dehumidification in air conditioning and the impact of the desiccant performance on the hybrid system performance. The two methods are refrigeration cycle with rotary desiccant (hybrid) and heat exchanger cycle. Both of the systems were compared with the conventional refrigeration system at different air flow rates and different conditions. The result showed that the coefficient of performance and cooling effect were close but the hybrid system achieved lower sensible heat ratio (SHR) and dew point temperature compared to conventional cooling systems.

Zhang and Lee (2011) investigated the performance of two hybrid systems in some four area of China. One was composed of vapours compression and desiccant (VC+D) cooling system, and the other was composed of vapours compression, desiccant and direct evaporative cooler (VC+D+EC) cooling system. Both of the systems were compared with conventional vapours compression. Energy consumption of the whole load using VC+D was saved by 4.21% and Hangzhao, 11.14% in Chengdu, 11.64% in Shenyang and 16.31% in Xi’an. VC+D+EC save 2.87% in Hangzhao, 11.04% in Chengdu, 10.93% in Shenyang and 22.74% in Xi’an. Later, Guojie, Chaoyu, Guanghai and Wu (2012) developed and analyzed two stage rotary desiccants in ship AC systems. The study started with investigating the amount of steam consumption available for rotary desiccant wheel. The new marine rotary desiccant air-conditioning can save up to 33.4% of the energy consumption compared with the traditional marine AC.

2.6 Heat Pipe Heat Exchanger (HPHX) Technology for Dehumidification

A heat pipe is a high performance heat transfer device which functions by evaporation and condensation sections of proper working fluid in individual closed tubes.

Figure 2.10 shows working fluid in evaporator section and condenser section. Heat is absorbed by evaporator section and working fluid inside of the heat pipe vaporizes, flow to the opposite or condenser end. The condenser section will release the energy and the fluid will re-evaporate and flow back to the evaporator section.
REFERENCES


