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HEURISTIC-ENGINEERING-STATISTICAL APPROACH FOR CHILLER OPTIMIZATION

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Ph.D

The Hong Kong Polytechnic University

2014
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HEURISTIC-ENGINEERING-STATISTICAL APPROACH FOR CHILLER OPTIMIZATION

MAK CHEUK WAI

A Thesis Submitted in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy

August 2013
CERTIFICATION OF ORIGINALITY

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__________________________  (Signed)

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Since the energy crisis occurred in 1973, the issue of energy use became the first priority for engineers and researchers to tackle this problem. In the recent two decades, ozone depletion and global warming have become two critical issues that affecting our global environment. Starting from 1990s, the Hong Kong government has been concerning about the energy use in air-conditioning systems within buildings. The government launched code of practice and guidelines for energy efficiency of air conditioning installations as well as guidelines on energy audit. Nonetheless, they were voluntary base which did not incentive to the building owners to follow. Until 2012, the government turned the codes and guidelines with
some modifications into Building Energy Efficiency Ordinance under Chapter 610 that all building services engineers and building owners shall follow the regulation. Nevertheless, some statements regarding the issue of assessing the energy performance of the centralized air-conditioning system within buildings and recommendations on optimizing the control for chiller plant are still vague. In addition, when assessing the energy performance of the chiller plant, inadequate and invalid building information and operating data were discovered in most of buildings in Hong Kong.

Notwithstanding, building energy performance or chiller performances are traditionally investigated by engineering modeling especially in the striking of the optimum point, this approach may be very successful in research level, whether it is laboratory based or simulation based. All through the past decades, there exist no successful optimization protocols in real building.

This thesis presents a protocol for assessing and optimizing individual chillers within building with supplementing the deficiency of the codes addressed by the Hong Kong government. This research starts with the concept of data mining with two approaches: (a) preliminary building energy performance assessment; and (b)
detailed chiller performance assessment.

For preliminary building energy performance assessment, the energy signature for building performance analysis based on statistical model is discussed. Using this technique, the applications of the energy signature for 20 surveyed buildings are studied. Through this analysis, the effects of the outdoor environment conditions on the building energy consumption especially the air-conditioning system for different types of functional use can be investigated. Furthermore, the survey on the availability of essential building information for establishing the building energy signature is also presented. Apart from using statistical modeling approach, a dissection of building energy consumption using engineering modeling approach in order to simulate the building energy performance for a selected commercial building is also highlighted. The results are the annual building electricity consumption breakdown, the effects of the climatic variable ($T_o^2$ and $T_oW_o$) on the building electricity consumption together with the degradations of the equipment of the centralized chiller plant within building.

For detailed chiller performance assessment, a modified refrigerant model for simulating the thermal-dynamic characteristic of R134a is firstly presented. The
accuracy of this model is significantly high when comparing the laboratory data.

Secondly, a data management protocol with 5 steps (including data acquisition, data synchronization, data conditioning, range validity conditioning and uncertainty analysis) in order to deal with the typical problems (i.e. due to data acquisition or sensor errors etc.) occurred in BMS is introduced. This protocol is a prior procedure that facilitates the next step for individual chiller performance assessment and optimization. Two chiller plants are selected for the study. Lastly, the Heuristic-Engineering-Statistical (HES) approach is developed in this study. It innovates this practice in two ways. On one hand, this study tries to identify low optimization operation zones (operating conditions) of chillers instead of highly optimized operating point or zones where the operating conditions are usually cannot be matched in real life operation, and converts these low optimized conditions into more optimized operating conditions. On the other hand, this study takes every site collected performance conditions as the operating settings in an experiment. At the completion of one annual cycle, the chiller can be thought of going through all most likely operating conditions in the building. This approach is then more realistic and applicable in the building. The HES model is then self-validated by the true operation of the chiller.
An integrated solution for individual chiller optimization involved the comprehensive working procedures. Therefore, a chiller server (electronic web-based integrated analysis scheme) for individual chiller performance optimization is established and demonstrated in this research project which is comprised of the data logic management protocol and the HES approach for individual chiller performance assessment and optimization. This chiller server is welcome by the industry and is expected to innovate the method for individual chiller performance analysis and optimization.
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NOMENCLATURE

\( A_{f,i} \)  
Area of the Building Façade Element including Wall and Glazing 
\([\text{m}^2]\)

\( A_{g,i} \)  
Area of the Glazing which is the \( i \)th Building Façade Element \([\text{m}^2]\)

\( a_0 - a_1 \)  
\( a_n \) Coefficients for Thermal Performance Line Model (Temperature Dependent)

\( b_0 - b_1 \)  
\( b_n \) Coefficients for Thermal Performance Line Model (Temperature Dependent)

\( CLF_i \)  
Cooling Load Factor for Cooling Load due to Solar Heat Gain from a Glazing which is the \( i \)th Building Envelope \([-]\)

\( COP \)  
Coefficient of Performance of Chiller \([-]\)

\( C_{pa} \)  
Specific Heat Capacity of Air \([1.02 \text{ kJ/kg/°C}]\)

\( C_{pw} \)  
Specific Heat Capacity of Water \([\text{i.e. } 4.185\text{kJ/kg/°C}]\)

\( c_0 - c_1 \)  
\( c_n \) Coefficients for Thermal Performance Line Model (Moisture Content Dependent)
\( d_0 - d_1 \)  \( d_n \) Coefficients for Thermal Performance Line Model (Moisture Content Dependent)

\( E_{cc} \) Compressor Power Consumption of Chiller [kW]

\( E_{cc}'' \) Scaled Compressor Power Consumption of Chiller [kW]

\( E_{cooling} \) Cooling Energy [kWh]

\( EER \) Energy Efficiency Ratio [BTU/hr/W]

\( ESEER \) European Seasonal Energy Efficiency Ratio [BTU/hr/W]

\( e_0 - e_1 \)  \( e_n \) Coefficients for Thermal Performance Line Model (Energy Dependent)

\( g_{ao} \) Outdoor Air Moisture Content [kg/kg]

\( g_c \) Off-Coil Air Moisture Content [kg/kg]

\( h_{fg,0} \) Latent Heat of Evaporation of Water [kJ/kg]

\( h_L \) Specific Enthalpy at Liquid State [kJ/kg]

\( h_0 \) Specific Enthalpy at Reference Temperature [kJ/kg]

\( h_{1'} \) Specific Enthalpy of Refrigerant at the Inlet of the Compressor [kJ/kg]

\( h_3 \) Specific Enthalpy of Refrigerant at the Discharge Outlet of the Compressor [kJ/kg]

\( h_5 \) Specific Enthalpy of the Refrigerant at the Outlet of Condenser
\[ h_6 \quad \text{Specific Enthalpy of the Refrigerant at the Inlet of Evaporator [kJ/kg]} \]

\[ k \quad \text{Time Instant} \]

\[ I \quad \text{Input} \]

\[ IPLV \quad \text{Integrated Part Load Value [kW/kW]} \]

\[ LMTD_{ev} \quad \text{Logarithmic Mean Temperature Difference of Evaporator [°C]} \]

\[ LMTD_{cd} \quad \text{Logarithmic Mean Temperature Difference of Condenser [°C]} \]

\[ L_v \quad \text{Latent Heat of Vapourization of Air [2450 kJ/kg]} \]

\[ MAD \quad \text{Standard Deviation for Normally Distributed Data} \]

\[ MSHGF_i \quad \text{Maximum Solar Heat Gain Factor of the Glazing which is the } ith \]

\[ \text{Building Façade Element [W/m}^2] \]

\[ m_a \quad \text{Mass Flow Rate of Air [kg/s] / Heat Rejection Air Flow Rate via} \]

\[ \text{Condenser [kg/s]} \]

\[ max \quad \text{Maximum} \]

\[ m_{cdw} \quad \text{Condensing Water Flow Rate [kg/s]} \]

\[ mean \quad \text{Mean / Average Value} \]

\[ m_r \quad \text{Refrigerant Mass Flow Rate [kg/s]} \]

\[ m_w \quad \text{Chilled Water Flow Rate [kg/s]} \]

\[ O \quad \text{Output} \]
\( N \) Number of Samples

\( NPLV \) Non-Standard Part Load Value [kW/kW]

\( P \) Pressure [kPa]

\( PLR \) Part Load Ratio [-]

\( P_{cc} \) Chiller Electricity Power Input to the Compressor [kW]

\( P_{cc}'' \) Scaled Chiller Electricity Power Input to the Compressor [kW]

\( P_{cd} \) Condensing Pressure of Refrigerant [kPa]

\( P_{ev} \) Evaporating Pressure of Refrigerant [kPa]

\( q \) Total Design Building Cooling Load [W/m\(^2\)]

\( q_{cd} \) Heat Rejection Load of Chiller/Condenser [kW]

\( q_{f,i} \) Cooling Load due to Conduction Heat Gain via the \( i \)th Building Façade Element including Wall and Glazing [W/m\(^2\)]

\( q_{g,i} \) Cooling Load due to Solar Gain via Glazing [W/m\(^2\)]

\( q_{inf,l} \) Infiltration Load Latent Load [W/m\(^2\)]

\( q_{inf,s} \) Infiltration Load Sensible Load [W/m\(^2\)]

\( q_{lgt} \) Internal Lighting Load [W/m\(^2\)]

\( q_{occ,l} \) Occupancy Load Latent Load [W/m\(^2\)]

\( q_{occ,s} \) Occupancy Load Sensible Load [W/m\(^2\)]

\( q_{rl} \) Cooling Load of Evaporator [kW]
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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>$q_{sp}$</td>
<td>Internal Small Power or Equipment Load [W/m²]</td>
</tr>
<tr>
<td>$q_{ven,l}$</td>
<td>Ventilation Load Latent Load [W/m²]</td>
</tr>
<tr>
<td>$q_{ven,s}$</td>
<td>Ventilation Load Sensible Load [W/m²]</td>
</tr>
<tr>
<td>$R$</td>
<td>Gas Constant (81.4881629 x 10⁻³ kJ/kg/°K) / An Estimate made from Variable $V_i$ to $V_n$ (i.e. Residual Value)</td>
</tr>
<tr>
<td>$R^2$</td>
<td>Coefficient of Determination</td>
</tr>
<tr>
<td>$R_h$</td>
<td>Residual Heat Rejection Rate [kW]</td>
</tr>
<tr>
<td>$SC$</td>
<td>Sensitivity Coefficient [-]</td>
</tr>
<tr>
<td>$SC_{g,i}$</td>
<td>Shading Coefficient of the Glazing which is $ith$ Building Façade Element [-]</td>
</tr>
<tr>
<td>$s_L$</td>
<td>Specific Enthalpy at Liquid State [kJ/kg]</td>
</tr>
<tr>
<td>$s_0$</td>
<td>Specific Entropy at Reference Temperature [kJ/kg/°K]</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature [°K]</td>
</tr>
<tr>
<td>$T_{app}$</td>
<td>Approach Temperature [°C]</td>
</tr>
<tr>
<td>$T_c$</td>
<td>Critical/Reference Temperature [i.e. 374.25°K]</td>
</tr>
<tr>
<td>$T_{cd}$</td>
<td>Refrigerant Condensing Temperature [°C]</td>
</tr>
<tr>
<td>$T_{cdwr}$</td>
<td>Condensing Water Inlet Temperature [°C]</td>
</tr>
<tr>
<td>$T_{cdws}$</td>
<td>Condensing Water Outlet Temperature [°C]</td>
</tr>
<tr>
<td>$T_{chwr}$</td>
<td>Chilled Water Return Temperature [°C]</td>
</tr>
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</table>
\( T_{chws} \) Chilled Water Supply Temperature [°C]

\( T_{dis} \) Discharge Temperature of Refrigerant [°C]

\( T_{ev} \) Refrigerant Evaporating Temperature [°C]

\( T_o \) Outdoor Air Dry-Bulb Temperature [°C]

\( T_{o,db,\text{in}} \) Outdoor Air Wet-Bulb Temperature Enter into Condenser [°C]

\( T_{o,db,\text{out}} \) Outdoor Air Dry-Bulb Temperature Exit from Condenser [°C]

\( T_{op} \) Annual Operating Hours of the Chiller Plant [hr]

\( T_{o,wb} \) Outdoor Air Wet-Bulb Temperature [°C]

\( T_r \) Design Indoor Air Dry-Bulb Temperature [°C]

\( t \) Condition at a Particular Time [s]

\( U_{f,i} \) Overall Heat Transfer Value of the Building Façade Element including Wall and Glazing [W/m\(^2\)/°C]

\( V_{inf} \) Infiltration Flow Rate [m\(^3\)/s]

\( V_{inf}^\prime \) Sensible Load Factor due to Infiltration Flow Rate [kW/°C]

\( V_{inf}^\prime\prime \) Latent Load Factor due to Infiltration Flow Rate [kW]

\( V_o \) Fresh Air Flow Rate [m\(^3\)/s]

\( V_o^\prime \) Sensible Load Factor due to Fresh Air Flow Rate [kW/°C]

\( V_o^\prime\prime \) Latent Load Factor due to Fresh Air Flow Rate [kW]

\( v \) Specific Volume Obtained by Iterative Solution of the Martin-Hou
Equation at Given Temperature $T$ and Pressure $P$ [m$^3$/kg]

$v_L$ Specific Volume at Liquid State [m$^3$/kg]

$v_2$ Specific Volume of Refrigerant after Inlet Guide Vane Throttling

[m$^3$/kg]

$v_3$ Specific Volume of Refrigerant at the Discharge Outlet of Compressor

[m$^3$/kg]

$W_{com}$ Actual Work Consumed by the Compressor [kW]

$W_i$ Indoor Air Moisture Content [kg/kg]

$W_o$ Outdoor Air Moisture Content [kg/kg]

$x$ Median of the Data Sequence

$x_k$ Average of the Latest $n$ Samples of a Data Sequence

$\bar{y}$ Mean of the Sample Value

$y_i$ Sample Value

$\hat{y}_i$ Predicted Value

**Greek**

$\Delta T_o$ Difference of Mean Outdoor Air Temperature [$^\circ$C]

$\Delta T_{chw}$ Chilled Water Temperature Difference [$^\circ$C]

$\Delta r$ Uncertainty in the Estimate $R$
\( \Delta \theta_L' \) Fictitious sensible Temperature Rise due to Latent Gains [°C]

\( \delta R_h \) Uncertainty in Residual Heat Rejection Rate

\( \delta V_i \) Uncertainty in the Variable \( V_i \)

\( \eta_m \) Electro-Mechanical Loss Efficiency [%]

\( \eta_{poly} \) Polytropic Efficiency of Compressor [%]

\( \rho_a \) Air Density [kg/m\(^3\)]

\( \rho_c \) Critical Liquid Density [i.e. 512.2kg/m\(^3\)]

\( \theta_{ao} \) Outdoor Air Dry-Bulb Temperature [°C]

\( \theta_c \) Off-Coil Dry-Bulb Temperature [°C]

\( \sigma \) Standard Deviation

\( \Phi \) Auxiliary Angle [Degree]

**Subscripts**

\( a \) Air

\( cc \) Compressor

\( cd \) Condensing

\( cdw \) Condensing Water

\( chw \) Chilled Water

\( com \) Polytropic Work
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>dis</td>
<td>Discharge</td>
</tr>
<tr>
<td>ev</td>
<td>Evaporating</td>
</tr>
<tr>
<td>m</td>
<td>Electro-Mechanical Loss</td>
</tr>
<tr>
<td>poly</td>
<td>Polytropic</td>
</tr>
<tr>
<td>w</td>
<td>Water</td>
</tr>
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</table>
INTRODUCTION

Understanding the geographic, climate and demographics in a city is vitally important before analyzing the energy consumption pattern. The government’s policy regarding the energy saving issue is also incentive for engineers or researchers to explore different types of energy saving approaches. Chapter 1 is divided into nine sections. The first section is Geographic, climate and demographics in Hong Kong. The following section is energy consumption pattern in Hong Kong. The third section is Hong Kong government policy and attitude on centralized chiller plants within buildings. The fourth section is literature review on chiller plant optimization approaches while typical problems of chiller operating data from building management system (BMS) is highlighted in section five. The last two sections are aims and objectives as well as organization of this thesis.

1.1 GEOGRAPHIC, CLIMATE AND DEMOGRAPHICS IN HONG KONG
Hong Kong is situated along the southern coast of China that is surrounded by both Pearl River Delta and South China Sea. Though Hong Kong is located just south of the Tropic of Cancer with latitude 22°21’N and longitude 114°21’E, it has a sub-tropical climate and it is hot and humid throughout the year [Chan and Mak 2008]. According to Wikipedia, in summer, it is hot and humid with occasional showers and thunderstorms as well as warm air coming from the southwest. In winter, it is mild and usually starts sunny, becoming cloudier towards February; the occasional cold front brings strong, cooling winds from the north. In 2011, Hong Kong averages 1,979 hours of sunshine per year while the mean daily highest and lowest recorded air temperatures at the Hong Kong Observatory are 32.4 °C and 11.6 °C respectively [HKO 2011].
Figure 1.1  Geographic Location of Hong Kong

The territory consists of Hong Kong Island, the Kowloon Peninsula, the New Territories and over 200 offshore islands. The land area in Hong Kong occupies 1,054 km\(^2\) while 50 km\(^2\) is inland water [CIA 2010]. According to figures from Census and Statistics Department [CSD 2010], the population in Hong Kong in 2011 was 7.112 million and the usable land was 1,104 km\(^2\). The population growth from 1996 to 2011 was increased to 9.852% (see Figure 1.2). Up to June 2011, the population density of Hong Kong was 6,544 people per km\(^2\) [CSD 2011]. Because of
the high population and limited supply of land, most of the buildings are high rise in order to cope with the rapid development [Chan and Mak 2008].

![Population Growth in Hong Kong (Year 1996 - Year 2011)](image)

**Figure 1.2** Population Growth in Hong Kong (from Year 1996 to Year 2010)

Hong Kong is one of the world’s leading international financial centers. In addition, the economy in Hong Kong is dominated by the service sector which accounted for over 90% of its GDP while industry constitutes 9% [Economist 2010]. Inflation was at 2.5% in 2007. Owing to sustain the high population density and economic development of Hong Kong, energy is no doubt a significant ingredient for the development. Figure 1.3 illustrates the Hong Kong building stock for private office and commercial premises from Year 2004 to Year 2011 [RVD 2012]. It can be seen
that private commercial premises occupied a large portion of the Hong Kong building stocks for eight years.

![Figure 1.3 Hong Kong Building Stocks for Private Office and Commercial Premises (from Year 2004 to Year 2011)](image)

**1.2 ENERGY CONSUMPTION PATTERN IN HONG KONG**

**1.2.1 Energy Consumption Trend**

According to the Hong Kong Energy-Use Data 2012 [EMSD 2012], the energy consumption of the domestic sectors, commercial sectors and industrial sectors in 2011 were 39,871 TJ, 100,457 TJ and 11,104 TJ respectively (1 TJ = 1 x 10^{12} J).
Furthermore, it was also observed that the energy consumption of the commercial and domestic sectors increased 19% and 15% since 2002. As the industrial industry was decreasing and is removed to southeast mainland China, the energy consumption in 2011 reduced for about 30% in the past ten years. Obviously, the main consumer of the electricity within buildings was commercial sectors and the energy consumption of the commercial sectors has continuously increased to 15% since 2002. In 2011, the commercial sectors occupied for more than 65% of the total. Apart from that, space conditioning (i.e. air-conditioning) was about 25% of the total electricity consumption that was the second largest portion when compared with other end-users. Some research studies carried out by Lam and Chan [1994]; Lam and Li [2003]; Yu and Chow [2007] indicated that, the energy break down in percentage among various types of building services systems in commercial building, the proportion of Heating, Ventilating, Air-Conditioning and Refrigeration (HVACR) system accounted for around 50% of the total energy consumption while lighting system, internal equipment, information technologies facilities as well as vertical transportation system occupied the other half. Another research papers also reported that the operation of chiller accounts for about 60% of the electricity used for air-conditioning system which can amount to 25% - 40% of the total electricity consumption in a commercial building [Chan and Yu 2002; Deng et al. 2002]. Owing
to provide and sustain an acceptable comfort, health and safe indoor environments, energy being used within buildings for people or industrial processes, no doubt, is a must. As a result, it is vitally important to explore how to manage the energy performance of chiller plants serving for commercial buildings in Hong Kong.

Figure 1.4  Energy Consumption in Hong Kong by Different Sectors (from Year 2002 to Year 2011)
CHAPTER 1
INTRODUCTION

Figure 1.5  Annual Energy Breakdown for Commercial Sector (Year 2011)

Figure 1.6  Annual Building Energy Use Breakdown by Percentage

(Commercial Building)
1.2.2 **Relationship between Climate Factor and Energy Consumption in Commercial Sector**

In fact, the trend of the energy consumption is also influenced by the outdoor weather conditions. As illustrated in Figure 1.7, the mean outdoor air dry-bulb temperature increased in Year 2011 (April to November) when compared with other years (i.e. Year 2002 to Year 2010) [HKO 2011]. This increase is due to the global warming effect.

By correlating the monthly energy consumption of the commercial sectors and the mean outdoor air dry-bulb temperature for ten years data, it can be easily observed that the trend of the energy consumption is directly proportional to the outdoor air dry-bulb temperature. This phenomenon can be explained by the demand of the energy used in air-conditioning system within buildings. As the outdoor air temperature increases, the energy consumption of the air-conditioning system is also increased. Reported by BBC News with title “Hong Kong Air-Con ‘is too cold’” [BBC 2005], increasing the temperature in offices across the territory by just 1.0°C would reduce carbon dioxide emissions by 2.5 Million tons a years.
Figure 1.7 Monthly Mean Outdoor Air Dry-Bulb Temperature Profile for Hong Kong (from Year 2002 to Year 2011)
Figure 1.8  Energy Consumption of Commercial Sectors Against Mean Outdoor Air Dry-Bulb Temperature (from Year 2002 to Year 2011)
1.3 HONG KONG GOVERNMENT POLICY AND ATTITUDE ON CENTRALIZED CHILLER PLANT WITHIN BUILDINGS

1.3.1 Electrical and Mechanical Services Department (EMSD)

Early in 1990s, the Hong Kong government has aware the issue of global warming which impacts on our global environment. In 1994, an Energy Efficiency Office (EEO) was established with the aim at providing the technical expertise and the drive for energy efficiency and conservation programmes [EMSD 1998a]. The office firstly launched the Code of Practice (CoP) for Energy Efficiency of Air Conditioning Installations (EE-AC) in 1998 [EMSD 1998a]. The code sets out the minimum design requirement on energy efficiency of air conditioning installations. It forms a part of set of comprehensive Building Energy Codes (BECs) that addresses energy efficiency requirements on building services installations. With regard to the issue of chillers, this code covers minimum requirements on coefficient of performance (COP) of chillers with standard rating conditions. Other than the code, the Guidelines on Energy Efficiency of Air Conditioning Installations was also issued [EMSD 1998b]. The guidelines listed the recommendations on operation of attended air conditioning installations including: daily/weekly routine operation and
maintenance duties, monthly routine services and repair as well as annual overhaul
for chillers. With the rapid technology development, the code and the guidelines
have been revised in many times (i.e. 2005, 2007) [EMSD 2005; EMSD 2007a]. In
the same year, EMSD also promoted “Hong Kong Energy Efficiency Registration
Scheme for Building” [EMSD 2007b]. Nevertheless, although the code and
guidelines have been updated in two versions and energy efficiency registration
scheme was promoted, they are voluntary base while it was not incentive to
encourage the private sectors to use the code and guidelines as well as to participate
the scheme. Later on, the Hong Kong government amended the ordinance and
enforced the code becoming mandatory [EMSD 2013a]. In 2012, a Building Energy
Code (BEC) covering the air conditioning installations, electrical installations,
lighting installations and lift & escalator installations, under the Building Energy
Efficiency Ordinance (BEEO) Cap. 610, was addressed. The minimum requirements
regarding the COP of chillers were stringent [EMSD 2012].

Reported by the Water Suppliers Department (WSD), surplus of fresh water (from
Dongjiong Guangdong province) on average in the past decade was recorded [Chan
2009]. As it is obviously understood that using water-cooled chiller is more efficient
than adopting air-cooled chiller since this type of chiller offers lower condensing
temperature and hence reduces the thermal lift of a chiller [Calm 2007], the EMSD started to promote “A Pilot Scheme for Wider Use of Fresh Water in Evaporative Cooling Towers in Energy Efficient Air Conditioning Systems in Designated Areas (for non-domestic buildings)”. This pilot scheme was firstly announced in 2000. At that time, only six designated areas were granted. Two years later, the designated areas were extended to twenty-eight. Up to 2006, there were fifty-seven designated areas were covered. Until 2010, the designated areas have been extended to one hundred and seven under a new scheme called “Scheme for Wider Use of Fresh Water in Evaporative Cooling”.

Energy Saving using Water-Cooled System instead of Air-Cooled System:

As a cooling tower give a lower condensing pressure than an air-cooled condenser resulting in a better coefficient of performance (i.e. more energy efficient), using water-cooled system instead of air-cooled system can have energy saving with ranges from 25% to 40% depending on the complexity and types of air-conditioning systems [EMSD 1998a].

Overview on Hong Kong and International Standards for Gauging Chiller Efficiency:

The overview and summary on Hong Kong and International Standards for standard rating conditions and minimum of chiller efficiency are presented in Appendix A.
In 2004, “Guidelines on Energy Audit” was published which enables the building services engineers or researchers to understand conducting energy audit for buildings (especially for centralized chiller plants within buildings) in a systematic and efficient way. The second revised version was issued in 2007. Since 21 September 2012, BEEO also covers the requirements on energy audit. Before this, a “draft for Code of Practice for Energy Audit in Buildings” was firstly introduced in 2010 and the finalized version was issued in 2012. As emission of greenhouse gas from buildings have been taken awareness, EMSD collaborated with the Environmental Protection Department (EPD) published the “Guidelines to Account for and Report on Greenhouse Gas Emissions and Removals for Building (Commercial, Residential or Institutional Purposes) in Hong Kong” in 2010 [EMSD 2008a].

In 2002, the EEO also launched the “Guidelines on Application of Central Control and Monitoring Systems” [EMSD 2002]. Regarding the centralized chiller plant, chiller plant optimization features, including chiller sequencing control and reset the chilled water supply temperature, were briefly discussed.

With the advanced technology development in recent decades, EMSD also completed three consultancy studies on Water-Cooled Air Conditioning Systems in
Hong Kong and the studies are “Territory-Wide Implementation Study of Water-Cooled Air Conditioning Systems in Hong Kong”, “Implementation Study for a District Cooling Scheme at South East Kowloon Development” and “Implementation Study for Water-Cooled Air Conditioning Systems at Wan Chai and Causeway Bay – Investigation” [EMSD 2013b], which were finished from 2002 to 2005. EMSD intends to promote the use of district cooling system in Hong Kong. Indeed, a lot of research studied revealed that using district cooling system would be more significantly energy efficient comparing with traditional air-cooled or water-cooled chiller systems. Starting from 2007, EMSD also promoted the use of oil-free air-cooled chiller in government buildings. Oil-free air-cooled chiller allows more energy saving during its part load operation when compared with traditional air-cooled chiller [EMSD 2008b].

1.3.2 Architectural Services Department (ASD)

Owing to maintain the standard and level of local support services as well as to enhance the servicing quality of chillers used in the air-conditioning installations in government buildings, ASD addressed “Requirements for Certification Scheme for Servicing Quality for Chillers in Air-Conditioning Installations in Government Buildings of the Hong Kong Special Administrative Region” in 2007 [ASD 2007].
Table 1.1  Timeline for Hong Kong Government Policy and Attitude regarding Centralized Chiller Plant within Buildings

<table>
<thead>
<tr>
<th>Year</th>
<th>Document</th>
<th>Requirement</th>
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</table>
| 1998   | Code of Practice and Guidelines on Energy Efficiency of Air Conditioning Installations | (a) Minimum COP requirements with Standard Rating Conditions are set for chillers based on survey results and the Air Conditioning and Refrigeration Association of Hong Kong.  
(b) Since very little data on part-load COP are available from manufacturers and there are also arguments on the adoption of Integrated Part Load Values (IPLV) for all building types in U.S. Therefore, part-load COP is not recommended in the code.  
(c) Operation of Attended AC Installations (Daily/Weekly Routine Operation and Maintenance Duties, Monthly Routine Services and Repair as well as Annual Overhaul) is included. |
| 1998   | Hong Kong Energy Efficiency Registration Scheme for Buildings             | (a) With the compliance of four sets of building energy code of practice (COP)  
(b) COP for Energy Efficiency of Air Conditioning, Electrical and Lift & Escalator Installations (1998 to 2000 version)  
(c) The updated version is 2007 version                                                                                                     |
| 2000   | A Pilot Scheme for Wider Use of Fresh Water in Evaporative Cooling Towers in Energy Efficient Air Conditioning Systems in Designated Areas | (a) For Non-Domestic Buildings in Six Designated Areas (in Year 2000); and  
(b) Extended to 28 Designated Areas (in Year 2002)  
(c) Extended to 57 Designated Areas (in Year 2006)                                                                                          |
| 2002 – 2005 | Executive Summary of the Territory-Wide Implementation Study for Water-Cooled AC systems in HK; Executive Summary of the South East Kowloon Development District Cooling System; and Executive Summary of Implementation Study for Water-Cooled AC System at Wan Chai and Causeway Bay - Investigation | -                                                                                                                                                                                                                     |
| 2004   | Guidelines on Energy Audit                                               | For Whole Building.                                                                                                                                                                                          |
| 2005   | Guidelines on Application of Central Control and Monitoring System       | (a) Chiller Optimization (for starting up and switching off sequence to maintain the optimum efficiency) is promoted; and  
(b) Raising the Chilled Water Supply Temperature is promoted.                                                                                   |
<table>
<thead>
<tr>
<th>Year</th>
<th>Document</th>
<th>Requirement</th>
<th>Detail</th>
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<tbody>
<tr>
<td></td>
<td></td>
<td>(b) Standard Rating Conditions for Air-Cooled and Water-Cooled Water Chillers</td>
<td>(b) Standard Rating Conditions for Water-Cooled Water Chillers are based on condenser water inlet and outlet temp, as well as the chilled water inlet and outlet temp.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(c) Minimum COP for Air-Cooled and Water Chiller with Reciprocating, Centrifugal and Screw or Scroll Compressors</td>
<td></td>
</tr>
<tr>
<td>2005</td>
<td>Guidelines on Performance-Based Building Energy Code</td>
<td>Examples for adopting total building energy budget approach are listed.</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(a) Standard Rating Conditions for Air-Cooled and Water-Cooled Water Chillers.</td>
<td>(b) The Minimum COP for all types of chillers are changed when compared with 2005 version.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(b) Minimum COP for Air-Cooled and Water-Cooled Water Chiller with Reciprocating, Centrifugal and Screw or Scroll Compressors.</td>
<td></td>
</tr>
<tr>
<td>2007</td>
<td>Guidelines on Energy Efficiency of Air Conditioning Installations</td>
<td>Same as 1998 version on AC equipment.</td>
<td>-</td>
</tr>
<tr>
<td>2007</td>
<td>Requirements for Certification Scheme for Servicing Quality for Chillers in Air-Conditioning Installations in Government Buildings of the Hong Kong Special Administrative Region</td>
<td>Intending to uphold the standard and level of local support services and to improve servicing quality of chillers used in the air-conditioning installations in Government buildings.</td>
<td>-</td>
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<tr>
<td>2007</td>
<td>Performance-Based Building Energy Code</td>
<td>Same as 2005 version.</td>
<td>-</td>
</tr>
<tr>
<td>2007</td>
<td>Performance-Based Building Energy Code</td>
<td>Same as 2005 version.</td>
<td>-</td>
</tr>
<tr>
<td>2007</td>
<td>Guidelines on Performance-Based Building Energy Code</td>
<td>Same as 2005 version.</td>
<td>-</td>
</tr>
<tr>
<td>2007</td>
<td>Guidelines on Energy Audit</td>
<td>Same as 2004 version.</td>
<td>-</td>
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<tr>
<td>2007</td>
<td>Promotion of Adopting Oil-Free Air Cooled Chiller</td>
<td>-</td>
<td>-</td>
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<tr>
<td>2008</td>
<td>Scheme for Wider Use of Fresh Water in Evaporative Cooling Towers for Energy-Efficient Air Conditioning Systems</td>
<td>(a) With further review in 2010.</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(b) Extended to 107 Designated Areas (up to 2012)</td>
<td></td>
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<tr>
<td>2010</td>
<td>Guidelines to Account for and Report on Greenhouse Gas Emissions and Removals for Buildings (Commercial, Residential or Institutional Purposes) in Hong Kong</td>
<td>(a) It is a guideline on Carbon Audit.</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(b) It embraces direct emissions &amp; removals, energy indirect emissions and other indirect emissions.</td>
<td></td>
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<tr>
<td>2010</td>
<td>Draft - Code of Practice for Energy Audit in Buildings</td>
<td>For Whole Building.</td>
<td>-</td>
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<tr>
<td>Year</td>
<td>Document</td>
<td>Requirement</td>
<td>Detail</td>
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<tr>
<td>2012</td>
<td>Code of Practice for Energy Efficiency of Building Services Installation – Energy Efficiency Requirements for Air Conditioning Installations</td>
<td>Minimum AC Equipment Efficiency&lt;br&gt;(a) Standard Rating Conditions for Air-Cooled and Water-Cooled Water Chillers.&lt;br&gt;(b) Minimum COP for Air-Cooled and Water-Cooled Water Chiller with Reciprocating, Centrifugal and Screw or Scroll Compressors.</td>
<td>(a) The Standard Rating Conditions are same as 2005 version but Sea Water-Cooled Water Chiller is also taken into account and furthermore, water side fouling factor for evaporator and condenser are also covered.&lt;br&gt;(b) The Minimum COP for Air-Cooled Water Chiller is the same as 2005 version while for Water-Cooled Water Chiller is tightened when compared with 2005 version.</td>
</tr>
<tr>
<td>2012</td>
<td>Code of Practice for Building Energy Audit</td>
<td>Same as Draft - Code of Practice for Energy Audit in Buildings version.</td>
<td>Energy Metering Requirements on Chiller and Chiller Plant (cooling capacity ≥ 350kW) regarding power consumption and cooling load as well as coefficient of performance (chiller compressor power, circulation pumps of condensers or cooling towers, condenser fans or radiator fans).</td>
</tr>
</tbody>
</table>
Figure 1.9  Timeline for Hong Kong Government Policy
1.4 LITERATURE REVIEW ON TYPICAL CHILLER
OPTIMIZATION APPROACHES AND APPLICATIONS

Before reviewing the recent research works focusing on chiller or chiller plant optimization approaches, it is important to understand the definition and rationales of the optimization and how to achieve optimization. After that, extending the optimization methods to chiller plants can then be easily managed. This section embraces: What is Optimization, Classifications of Supervisory Control and Optimization Methods, Applications on Typical Chiller and Chiller Plant Supervisory Control and Optimization Approach as well as Deficiency of Typical Local/Global Optimization Methods for Real Chiller Plants.

1.4.1 What is Optimization

In mathematics, statistics, empirical sciences, computer science or management science, mathematical optimization is the selection of a best element, with regard to some criteria, from some set of available alternatives [Marti-Herrero 2013]. In general, an optimization problem involves minimizing or maximizing a real function by systematically choosing input values from within an allowed set ad computing the value of the function. It includes searching the best available values of some objective functions given a defined domain, involving a variety of different types of objective functions and different types of domains. In the field of engineering, a lot of computational optimization techniques for engineering systems are well developed [Nelles 2001]. Wang and Ma [2008] further elaborated the classification and application of the optimization techniques used in HVAC supervisory control.
1.4.2 Classifications of Supervisory Control and Optimization Methods

In the recent decades, optimization supervisory control methods in HVAC industry have been well developed. In general, there are four types of the optimization supervisory control method classified and they are: (a) Model-Based Supervisory Control, (b) Model-Free Supervisory Control, (c) Hybrid Supervisory Control and (d) Performance Map-Based Supervisory Control [Wang and Ma 2008].

1.4.2.1 Model-Free Supervisory Control Method

For Model-Free Supervisory Control, no simulation or mathematical model, in order to simulate the performance of the HVAC equipment or system based on a set of input candidates and hence to seek for solutions (i.e. optimum points), is required. Henze and Schoenmann [2003] stated that expert systems and reinforcement learning approaches can be applied. Basically, they are based on a set of rules defined in the knowledge base and information obtained from the system or attempt to improve its performance based on the results of pervious actions. As claimed by Wang and Ma [2008], the drawback of adopting expert system is that this system is seriously influenced by the richness of the knowledge database since the rules are static and threaten significant errors outside their domain of expertise while reinforcement learning control takes very long time to make the controllers to learn. The controller performance is very sensitive to the selection of the state-action, learning parameters etc.

1.4.2.2 Model-Based Supervisory Control Method

Unlike Model-Free Supervisory Control, simulation or mathematical model(s) is/are required in Model-Based Supervisory Control. In fact, this type of control method
has been widely adopted in research areas [Wang and Ma 2008]. The characteristics of each type model used in HVAC system, especially chiller plants, are discussed in section 1.4.3.2. The main purpose of using such models is to act as a problem in order to seek for the solution(s) (e.g. maximum COP or minimum energy consumption etc. in HVAC system) by simulating various sets of input candidates (i.e. like the system operated under different working conditions). After determining different outputs by simulating various sets of input candidates, optimization technique(s) together with a set of constraint(s) is/are required in order to find out the optimal solution(s) [Nelles 2001].

1.4.2.3 **Hybrid Supervisory Control Method**

With making use of the merits of Model-Free Supervisory Control and Model-Free Supervisory Control method, both methods may also be associated together in order to solve the HVAC system problems. Wang and Ma [2008] classified this approach as Hybrid Supervisory Control Method.

1.4.2.4 **Performance Map-Based Supervisory Control Method**

Alternative approach called Performance Map-Based Supervisory Control Method is also adopted in the HVAC industry. This method either uses the simulation or laboratory test results over the range of expected operating conditions in order to draw a performance map to illustrate the operating performance of the system. Based on this map, control settings giving the maximum or minimum values are therefore identified.
1.4.2.5 Optimization Methods

As discussed in Model-Based Supervisory Control Method, optimization technique(s) is/are required in order to search for the optimal solution(s) from an optimization problem. In HVAC system problems, global or local optimums exist in determining the solutions. This will affect in selecting the optimization technique(s). On the other hand, computational efficiency and memory demand of each type of optimization technique should also be taken into consideration.

Nelles [2001] categorized the optimization technique(s) into two main types and they are namely: (a) Linear Optimization Technique and (b) Non-Linear Optimization Technique. Linear Optimization Technique is simple and straightforward that is used to solve a unique local optimum optimization problem while Nonlinear Optimization Technique is applied to solve the optimization problems with many local optimum exist and thus, it is more complex and sophisticated when compared with the former one. In fact, Nonlinear Optimization Technique can be further sub-classified as (a) Nonlinear Local Optimization Technique and (b) Nonlinear Global Optimization Technique. Wang and Ma [2008] addressed that the major difference between these two techniques is the Nonlinear Local Optimization Technique always leads to a local not global optimum. The details of each type of optimization techniques together with their advantages and disadvantages can be found in Nelles [2001] and Wang and Ma [2008].
1.4.3 Applications on Typical Chiller and Chiller Plant Supervisory Control and Optimization Approach

As discussed in previous section, Model-Based Supervisory Control Method has been widely adopted in research field. In general, modeling the energy performances/characteristics of the chiller system or equipment is the first step to achieve chiller plant optimization target. With understanding the energy performances of the chiller system or equipment, the optimal settings of the system or equipment under different operating conditions can then be determined. Therefore, this sub-section is divided into two parts: (a) Chiller Model Development and (b) Chiller Plant Optimization Approach Applications.

1.4.3.1 Chiller Model Development

In recent decades, a lot of models, (i.e. physical, semi-empirical and empirical based), for the prediction of the energy use or the performance of the chiller with its associated equipment have been well developed. The concepts, merits and demerits among three types of models are summerized in Table 1.2.
### Table 1.2 Classifications of Types of Modeling Methods for Chillers

<table>
<thead>
<tr>
<th>Type</th>
<th>Concept</th>
<th>Advantage</th>
<th>Disadvantage</th>
</tr>
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</table>
| Physical Model        | (a) Entirely based on the first principle (i.e. mass, momentum and energy balance relations) approach with a set of mathematical equations to describe the behaviour (i.e. performance) of the equipment.  
(b) A physical relationship interacts among all components inside a chiller is maintained.  
(c) No training data (i.e. collection of the actual measured data) is required. | (a) The accuracy of the prediction of the output (i.e. behaviour of the equipment) is excellent.  
(b) Suitable for both steady-state and transient-state conditions prediction with high accuracy. | (a) A serious of algebraic mathematical equations (with iterative procedures) to solve the equations may be needed incurring complex and computationally intensive effort.  
(b) Input details of the equipment (e.g. the constituent materials, configurations and dimensions of the internal components, operating conditions within the equipment etc.) are not readily available from the proprietary manufacturers.  
(c) Impractical method. |
| Empirical Model       | (a) Appropriate mathematical form/function, deterministic input variables and output for a model are required.  
(b) Numerical technique (e.g. linear or multiple regression etc.) with training data for calibrating the model (i.e. determining the coefficients appearing in the equation) is required.  
(c) No physical relationship. | (a) Simply enough hence computational costs as well as the efforts of modeling can be greatly relieved.  
(b) More practicable. Indeed, this model is widely adopted by researchers or engineers for steady-state analysis. | (a) Poor selection of the deterministic input variables without any physical meaning will definitely stress on the accuracy of the model.  
(b) Too difficult to understand how the variables of the components interact with each other and whether the performance can be improved or not.  
(c) It is accurate only for operating data within the ranges of the training data covered while poor outside these ranges due to extrapolation. |
| Semi-empirical Model  | (a) It combines two different modeling approaches simultaneously.  
(b) The process among the components, based on first principle relations, can be translated to formulate the mathematical form.  
(c) The coefficients appearing in the equations are determined by the training data. | (a) Complexity of the model structure and the computational efforts can be greatly reduced.  
(b) Coefficients in the equations still have certain physical significances.  
(c) This can mutually supplement the demerits of physical and empirical. | (a) Although the mathematical form developed in this model is based on the physical meanings, the accuracy of the model is still subject to the operating range of the data covered for training or calibrating the model (i.e. coefficients). |
(a) **Empirical Model**

**Braun et al. [1989]** suggested that chiller power consumption can be correlated with the cooling load, the difference between the condensing water return temperature and the chilled water supply temperature as independent variables using bi-quadratic equation. This method has been published in ASHRAE Applications handbook for the concept of optimal chiller loading [ASHRAE 2003]. Another chiller model developed by **Yik and Lam [1998]** for air-cooled and water-cooled as well as sea water-cooled chillers based on the bi-quadratic equation (i.e. polynomial with two independent variables) with taking into account the variations of cooling load and condenser water inlet temperature (for water-cooled chiller) or outdoor air condenser inlet temperature (for air-cooled chiller) has also be adopted by many manufacturers and researchers. The output of the model is the power input of the chiller.

**Yik et al. [2012]** correlated the power consumption of the chiller to the cooling load as well as condenser air inlet or condensing water return temperature with bi-quadratic equation to develop oil-free air-cooled and water-cooled chiller model. The chiller models were established from chiller manufacturer’s performance data.

A model-based fault detection and diagnosis technique applied to the water-cooled centrifugal chiller was purposed by **Cui and Wang [2005]**. The authors used a fully empirical regression model with taking into account the major influencing variables: (i) cooling load; (ii) chilled water supply temperature; and (iii) condensing water return temperature for the evaluating of the performance of the major components of the chiller including the log mean temperature difference of condenser and evaporator, refrigerant mass flow rate, isentropic efficiency of the compressor and
motor efficiency as well as COP of chiller.

In 1980, Department of Energy (DOE) [1980] also used another approach to develop another chiller model. It consists of three representative curves with 15 regression coefficients: (i) CAPFT curve represents the availability capacity as a function of evaporator and condenser temperatures; (ii) EIRFT curves represents the full load efficiency as a function of evaporator and condenser temperatures; and (iii) EIRFPLR curve represents the efficiency as a function of the percentage unloading. The evaporator temperature (i.e. chilled water supply temperature) and condenser temperature (i.e. condenser water outlet temperature) are embodied for water-cooled chiller while outdoor air dry-bulb temperature is embodied for air-cooled chillers of in this model. Later on, Hydeman et al. [2002] claimed that the DOE electric chiller model was unresponsive to the variations in condensing water flow and had large errors in the prediction of the power of variable speed centrifugal chiller at low load conditions. Due to this reason, they attempted to modify the model, by replacing the condensing water supply temperature term to condensing water return temperature (condenser water inlet) for water-cooled chillers while the outdoor air for air-cooled chillers, for the three representative curves of the model. Their results revealed that the maximum error of the original model for centrifugal chiller with variable speed drive could be larger than 30.4% whereas 10.4% maximum error could be achieved by using the modified model. For the case of fixed speed centrifugal chiller with varied condensing water flow, there was 3.0% maximum error when adopting the modified model while 8.2% was discovered when using the original model.

Gordon et al. [1995] explored and developed an empirical model with
thermodynamic based for predicting the COP of the centrifugal chillers with validation by using experimental chiller plant data. The model can be used as a diagnosis tool to analysis the heat exchangers fouling effect on the chiller performance. Later, Ng et al. [1997] also established another thermodynamic model based on Gordon et al. [2000] for reciprocating chillers for diagnosis purpose. The method for locating the optimal operating conditions and evaluating the potential improvements was presented. In order to investigate how the condensing water flow rate influence on chiller COP and cooling load, Gordon et al. [1995] revised the original model by taking into account the control variable (i.e. condensing water flow) to seek for the impact on chiller COP and cooling load for water-cooled centrifugal chiller. While Lee [2010] developed a model for screw liquid chillers that was modified from Gordon and Ng’s simplified linear regression model for centrifugal and reciprocating chillers [Gordon and Ng 2001]. After that, Lee and Lu [2010] then conducted a research for evaluating the best prediction of water-cooled centrifugal chiller coefficient of performance (COP) using six empirically-based models. They are: (i) Simple Linear Regression Model with taking into account cooling load, chilled water return temperature and condensing water return temperature with three regression coefficients. (ii) Bi-quadratic Regression Model with taking into account cooling load and condensing water return temperature as independent variables and nine regression coefficients; (iii) Polynomial Regression Model where cooling load, chilled water return temperature and condensing water return temperature as independent variables and ten regression coefficients; (iv) Gordon-Ng Universal Model with taking account the same independent variables as Polynomial regression model with three regression coefficients; (v) Gordon-Ng Simplified (Linear Regression) Model with taking into cooling load, chilled water
supply temperature and condensing water return temperature and three regression coefficients; (vi) Lee’s Simplified Model based on the first and second laws of thermodynamics as well as the NTU-ε method of heat exchanger to predict the COP of the chillers. The independent variables included in the model are the same as Gordon-Ng Universal Model. Three groups of chiller operating data sets were used: (i) constant condensing and constant chilled water flow rate from six chillers (capacity range from 500 to 1250 RT); (ii) variable chilled water flow rate and constant condensing water flow rate from three chillers (capacity range from 500 to 1000 RT) and; (iii) variable chilled water and variable condensing water flow rate from an existing chiller with 1250 RT. The result revealed that Bi-quadratic Regression Model with root-mean-square error 2.2% and the Polynomial Regression Model with root-mean-square error 2.25% have the best accuracy for all sorts of the data sets.

Stoecker et al. [1982] developed a regression model for predicting the chiller compressor power by selecting the refrigerant condensing and evaporating temperature as the independent variables of the model. Cecchinato et al. [2010] also utilized this approach to further elaborate and developed another model for air-cooled scroll chiller for evaluating the seasonal energy performance of the chillers.

The authors used (i) multivariable linear model for power input prediction and (ii) multivariable non-linear model for COP prediction, (iii) multivariable polynomial model for power input and COP prediction. The cooling load (over rated cooling
capacity), condenser water leaving temperature and outdoor air temperature were selected as deterministic variables for the model.

Nowadays, artificial neural networks (ANN), has many merits especially for dealing with the high non-linear (HVACR) systems, has been widely applied for chiller modeling. Swider [2003] established a chiller model by using the Radial Basis Function (RBF) and the multilayer perception (MLP) artificial neural network to predict the COP of a single-circuited centrifugal and a twin-circuited twin-screw chiller. In the meantime, the author also compared the prediction performance of the model with the Simple Linear Regression Model, Bi-quadratic Regression Model, Polynomial Regression and Gordon-Ng Universal Model. The results illustrated that these two cases considered neural network models show a higher level of difficulty but also higher generalization abilities the regression models since no inclusion of further knowledge of the operation of the chiller to accurately predict the performance in case of a more complex chiller required when the neural network models are used. The accuracy of the neural models in predicting a chiller performance always found to be higher when comparing to the regression models without including further knowledge.

Yang et al. [2008] established centrifugal and screw chiller model by adopting the ANN for an institution building. In this model, it is a three layer feed-forward network (i.e. input layer, hidden layer and output layer). Among the layers, neurons are connected with different weights to form a dynamic arithmetic system with the capacity of memory, study, prediction and fault tolerance. For input layer, the number of input neurons was set as five. Chilled water supply and return temperature,
condensing water supply and return temperature as well as compressor power input were selected as the input variables. For output layer, COP and compressor power input were selected. For hidden layer, a trial and error scheme was adopted and the number of hidden neurons for the best performance of this ANN model was eight. According to the statistical result, it revealed that the coefficient of determination for COP and compressor power input for centrifugal chiller were 0.9998 and 0.9963 which were very high enough when compared with the actual measured data. For the screw chiller, it was found that the coefficient of determination for COP was 0.9987 and for compressor power input was 0.9998 which were also high enough. Apart from that, the maximum error between the simulated result and the actual measured data for centrifugal and screw chiller was within ±10%. This result depicted that using ANN for predicting the centrifugal and screw chiller quite accurately. As aforementioned, most of empirical models are based on cooling load, chilled water supply or return temperature and condensing water supply or return temperature as the major influencing variables to the chiller power consumption. Nevertheless, the operating thermo-properties of the refrigerant within the refrigeration cycle can also affect the power consumption of the chiller compressor directly.

(b) Semi-Empirical Model

Bourdouxhe et al. [1996] developed a toolkit for primary HVAC system energy calculations. The toolkit contains subroutine programs to model a water-cooled centrifugal chiller at full and part load separately. The evaporator and condenser are modeled using the NTU- method (i.e. with the assumption of constant overall heat transfer value), semi-empirical models are applied determine the power input to the centrifugal compressors when the capacity control is done by adjusting the open
position of the inlet guide vanes at constant speed.

Wang and Wang [2000] developed a model water-cooled single-stage and two-stage centrifugal chillers by simulating the compressor, condenser, evaporator and load control. For compressor part, it is simulated based on the momentum equation (i.e. Euler Turbomachine equation), the thermodynamic equations and the equations of impeller velocity component relations. The energy performance of the chiller is simulated by considering compressor polytropical work, hydrodynamic losses and mechanical and electrical losses. For evaporator and condensing part, regression model with mechanistic relations (by taking into account water flow rate and cooling load as well as heat rejection rate), for evaluating the overall heat transfer value without knowing the details of the configurations of the condenser and evaporator, was introduced in the chiller model. On the other hand, it is assumed that the refrigerant leaving from the evaporator and leaving from the condenser are in saturated states. The chiller model can also simulate the dynamics of the chiller by assuming two lumped thermal storages corresponding to the condenser and evaporator respectively.

Another type of semi-empirical model chiller model for centrifugal liquid chillers was purposed [Browne and Bansal 1998]. This model is capable for simulating both hermetic and open-drive centrifugal compressor. The model takes into account for the superheating and subcooling phenomena at the compressor suctionline and condenser outlet respectively and the capacity control formulation of the inlet guide vanes. For condenser part, the NTU-ε method (i.e. with the assumption of constant overall heat transfer value) is adopted in order to determine the variations of the
refrigerant temperatures. The heat losses and gains to or from the environment affecting the performance of the chillers under low load conditions are considered in the model. For compressor part, a polytropic compression efficiency by using the empirical regression model with the non-dimensional parameters (i.e. flow coefficient and Mach number) and the motor efficiency as well as the small pressure drop due to the degree of opening of the inlet guide vane at the suction of the compressor is applied and taken into consideration. The input parameters of the model include: (i) chilled water supply temperature; (ii) condensing water return temperature; and (iii) general parameters of the construction of the chiller. The outputs of the model are: (i) heat rejection rate at the condenser; (ii) cooling load at the evaporator; (iii) mass flow rate of the refrigerant; (iv) coefficient of performance; and (v) thermodynamic states of the refrigerant within the refrigeration cycle.

Yu and Chan [2007] developed an air-cooled centrifugal chiller model (steady-state) with a condenser-fan control. The purpose of this model is to investigate to optimize the control of condenser fans (i.e. floating refrigerant condensing temperature) with the chillers to maximize their coefficient of performance (COP). The model considers the thermodynamic behaviors among various types of the major components (i.e. evaporator, condenser, compressor and expansion device). For evaporator and condenser part, a multiple regression model based on Wang and Wang [2000] was adopted in order to determine the overall heat transfer value of the evaporator and condenser without knowing the configuration details. For compressor part, a quadratic regression model with independent variable, part load ratio, was utilized for evaluating the polytropic efficiency of the compressor as well as the ratio of the actual mass flow rate of the refrigerant to the mass flow rate at design
condition for finding the effect of the throttling pressure drop due to the inlet guide vane at the suction of the compressor. In addition, a control algorithm of the condenser fans based on the outdoor air temperature and the set point condensing temperature as well as the part load ratio of the chiller owing to determine the number of the operating condenser fans, by considering the required heat rejection air flow rate, was also formulated.

Ma and Wang [2011] developed a simplified semi-empirical chiller model for water-cooled centrifugal chiller. This model is based on fictitious refrigeration cycle by assuming all the refrigerant state points are in saturated states. That means all the thermo-properties of the saturated points of the refrigerant within the refrigeration cycle can be simply found by using the measured refrigerant condensing and evaporating temperature. The inputs of this model are: (i) condensing water return temperature; (ii) cooling load; and (iii) chilled water supply temperature set-point while the output is the chiller compressor power consumption. In this model, fictitious refrigerant mass flow rate (taking into account cooling load, saturated specific enthalpy of the refrigerant at the inlet and outlet of the evaporator) and fictitious compressor power input (taking into account calculated fictitious refrigerant mass flow rate, saturated specific enthalpy of the refrigerant at the suction and discharge of the compressor) can be evaluated based on thermodynamic principle laws. For determining the actual compressor power input, a second-order polynomial regression equation with the fictitious compressor power input, as the independent variable, was formed. Apart from that, the overall heat transfer value of condenser and evaporator can be determined by using multiple regression analysis without knowing the details of the configurations of the condenser and evaporator.
Beitelmal and Patel [2006] developed a steady-state coupling model for resolving the energy equations for a cooling tower and a centralized water-cooled centrifugal chiller simultaneously. The compressor part is simulated by using polytropic efficiency and utilizing an empirical model for the chiller power input as a function of cooling load and the temperature difference between the condensing water supply temperature and the chilled water supply temperature. This compressor part can also be used for constant and variable speed chiller. The input variables of the entire model are: (i) outdoor air conditions; (ii) heat rejection air flow rate of the cooling tower; (iii) condensing water flow rate; (iv) chilled water flow rate; (v) superheat and subcooling associated with the refrigeration cycle; and (vi) full load design condition of the chiller. The outputs can be generated include: (i) chiller coefficient of performance; (ii) compressor power input; and (iii) compressor polytropic efficiency.

A screw liquid chiller steady-state model was established with the consideration of non-economized and economized issue [Fu et al. 2002]. This model includes sub-models for key components such as non-economized compressor, economized compressor, shell-and-tube condenser, expansion value and flooded evaporator. Sequential modular method and successive substitution method are combined together for simulation. This method treats the components of the chillers as independent modules that allow it easy to maintain the simulation software. For compressor part, empirical method is applied for determining the volumetric efficiency of the compressor. For heat exchanger part, one-zone model (i.e. condensation region only) with log mean temperature difference approach is adopted for analyzing the thermo-properties of the refrigerant at the condenser while two-zone model (i.e. evaporation region and superheated region) is applied for
evaporator. The calculation of the heat transfer coefficient of the refrigeration is required. It was found that the cooling capacity of the screw liquid chiller can be increased by adding the economizer, but only when the volumetric ratio of the first stage internal compression is greater than a certain value, the economized screw liquid chiller can gain a higher COP than that of the non-economized one at the same time.

An open reciprocating compressor type chiller model for a variable speed vapour compression system consisting of sub-component models (evaporator, condenser and compressor etc.) to predict the system performance by using information on the secondary fluids input conditions and the compressor speed as well as the refrigerant thermophysical properties and the geometric characteristic of the system to predict the secondary fluids output temperatures, the operating pressures, the compressor power consumption and the system overall energy performance was purposed [Esbri et al. 2010]. For evaporator and condenser part, the details of the configuration of the heat exchangers and the calculation of the refrigerant heat transfer coefficients within the vapour compression refrigeration cycle are required. In order to correlate the speed of the compressor, refrigerant condensing and evaporating pressure to the compressor power consumption, empirical model for estimating the volumetric efficiency, isentropic efficiency and global electromechanical efficiency are included in the model.

Yu and Chan [2005] developed an air-cooled screw chiller model (steady-state) and utilized it to investigate the operating performance of the chiller under the conventional head pressure control and new condensing temperature control. The
model considers the thermodynamic behaviors among various types of the major components (i.e. evaporator, condenser, compressor and expansion device). For evaporator and condenser part, a multiple regression model based on Wang and Wang [2000] was adopted in order to determine the overall heat transfer value of the evaporator and condenser without knowing the configuration details. For compressor part, regression modeling method was applied for determining the volumetric efficiency as a function of compression ratio, the isentropic efficiency as a function of refrigerant condensing and evaporating temperature as well as chiller cooling rated capacity, the combined motor and transmission efficiency as a function of part load ratio of the chiller. In addition, a control algorithm of the condenser fans based on the outdoor air temperature and the set point condensing temperature as well as the part load ratio of the chiller owing to determine the number of the operating condenser fans, by considering the required heat rejection air flow rate, was also formulated. This model assumed the operation of the evaporators in the four refrigeration circuits are in parallel. It means the same amount of heat transfer surface in the whole evaporator will be provided for cooling the chilled water even if only one circuit is in operation. Nonetheless, in practice, the evaporator is compartmentalized with each compartment will have individual refrigeration circuit. With supplementing the deficiency of Chan and Yu’s model, Lee et al. [2010] established another model for an air-cooled screw chiller with the consideration of two separated refrigeration circuit (i.e. two screw compressors per circuit) and the evaporator shell consisting of two separated compartments (each belonging to one refrigeration circuit).
(c) **Physical Model**

A physical model for two-stage centrifugal chiller was developed [Jeong and Lee 2003]. They performed and compared the refrigeration cycle analysis of the chiller approach (i.e. based on thermodynamic equations) and non-dimensional approach (i.e. refrigerant flow rates, impeller diameters and wheel speeds) of the two-stage compressor in order to impose hydrodynamic similarity of the compression (i.e. the same values of tip Mach number, flow coefficient, polytropic head coefficient, specific speed and specific diameter except Reynolds number) as much as possible.

Lei and Zaheeruddin [2005] developed a lumped-parameter dynamic model of a water-cooled chiller based on mass and energy balance principles. The component models (evaporator, condenser, compressor and thermostatic expansion valve) were formulated and were investigated to develop an overall model of water-cooled chiller. A control-oriented approach to develop the model and the effect of the control inputs (compressor operational frequency and thermostatic expansion valve opening fraction) on the output performance of the system was studied. The result of the transient response characteristics revealed that the thermal system responses were much slower than the pressure and the mass flow rate.
1.4.3.2 Chiller Plant Optimization Approach Applications

With regard to the applications on chiller plant optimization approach, there are several guidelines, reports, articles and papers can be found elsewhere [Gidwani 1984; Johnson 1985; Spethmann 1985; Trane 2000; ESMD 2002; Webster 2003; Crowther and Furlong 2004; Ahmed 2007; ANSI/ASHRAE 2007; Hu et al. 2007; UPPC 2009; Wang 2010]. In General, the typical optimization control strategies are:

(a) Chilled Water Supply Temperature Reset, (b) Condenser Water Return Temperature Reset, (c) Chiller Sequencing Control, (d) Condensing Temperature Reset, (e) Variable Chilled & Condensing Water Flow, (f) Optimal Start/Stop, and (g) Demand Reduction during Pulldown etc.

Apart from the above listed typical optimization for chiller plant, Hartman introduced a concept of the Equal Marginal Performance Principle [Hartman 2005]. The energy performance of any system operating with multiple modulating components is optimized when the change in system output (i.e. marginal system output) per unit energy input is the same for all individual components in the system. As system output per unit input is the definition of COP, marginal system output per unit energy input is also named as marginal COP. In other words, when all components have exactly the same marginal COP, the system is at the lowest possible power input for the current output requirement. The system is optimized. The marginal component COP at a point of operation for any component is the partial derivative of the expression of total system output as a function of the energy input requirements with respect to that power input of the component.
Table 1.3 Summary of Chiller Plant Control Strategies

<table>
<thead>
<tr>
<th>Chiller Plant Control Strategies</th>
<th>Working Principle</th>
<th>Rationale</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chilled Water Supply Temperature Reset</td>
<td>Energy Trade-off between Chillers and Chilled Water Pumps</td>
<td>The COP of chiller can be increased by raising the chilled water supply temperature. Increasing the chilled water supply temperature can be supplemented by increasing the chilled water flow rate for maintaining a constant cooling load.</td>
</tr>
<tr>
<td>Condensing Water Return Temperature Reset</td>
<td>Energy Trade-off between Chillers and Cooling Tower Fans</td>
<td>The COP of chiller can be increased by decreasing the condensing water return temperature. Decrease the condensing water return temperature can be supplemented by increasing the heat rejection air flow for maintaining a constant heat rejection load.</td>
</tr>
<tr>
<td>Chiller Sequencing Control</td>
<td>Based on System Coefficient of Performance</td>
<td>The chiller or chiller plant On/Off sequence is based on the System COP at a particular cooling load to implement.</td>
</tr>
<tr>
<td>Condensing Temperature Reset</td>
<td>Energy Trade-off between Chillers and Condenser Fans</td>
<td>The COP of chiller can be increased by decreasing the refrigerant condensing temperature for air-cooled chillers. Decrease the refrigerant condensing temperature can be supplemented by increasing the heat rejection air flow for maintaining a constant heat rejection load.</td>
</tr>
<tr>
<td>Optimal Start/Stop Control</td>
<td>Taking into Account the Building Façade and Chilled Water Circuit Thermal Storage Capabilities, Outdoor Air Condition and Occupant Comfort Issue</td>
<td>-</td>
</tr>
<tr>
<td>Demand Reduction during Pulldown</td>
<td>Based on Maximum Demand during Pulldown Period</td>
<td>It is to limit the power draw of the chiller during pulldown and consequently, increase the length of time required to research the desire chilled water supply temperature.</td>
</tr>
</tbody>
</table>

**Condensing Water Return Temperature Reset:**

According to rule of thumb, for every one degree decrease in condensing water temperature, the chiller efficiency will increase 2% [Bitondo et al. 1999].
(a) **Chiller Optimal Start**

Generally, the aim of the chiller optimal start control strategy is to find the most efficiency way in order to schedule the chiller start operation (i.e. no. and size of chillers in operation), for cooling down the building indoor design temperature before occupying with the issue of energy saving purpose. Sun *et al*. [2010] purposed a chiller optimal start control strategy with the consideration of the recovery ability and the cooling load condition as the optimizing variables. The control strategy consists of two steps: (i) the first step is by adopting a simplified building model to predict the cooling load of the building and to identify a feasible set for the numbers of the chillers in operation; and (ii) the second step is evaluate the pre-cooling lead time by using the simplified building model for each number inside the feasible range identified in the first step and then calculation the corresponding energy consumption. The number and its corresponding pre-cooling lead time which produces the least energy consumption constitute the optimal start.

Zhou *et al*. [2002] established and adopted a chiller start-stop optimization program together with the usage of the thermal storage tank in order to estimate the number of chillers needed to be brought on-line and the start and stop times for each chiller daily based on the prediction of the building cooling load within the coming 24 hours.

(b) **Chiller Optimal Stop**

With considering the issue of the thermal storage capacity of the chilled water inside the chilled water pipes of the chilled water distribution system, energy wastage will be occurred when the chillers are switched-off manually right after the occupancy
period of the building, Liu [2008] conducted a study on exploring the feasibility to use optimal stop strategy to save the chiller plant energy use. This study was implemented by using mathematical HVACR component models (including: chilled water pipework model, heat exchanger model, AHU model and zone model), with the consideration of the variations of the building cooling load and outdoor air temperature as well as the thermal capacity of the zone air, in order to investigate the temperature of the chilled water as well as the indoor air temperature. By restricting the range of the indoor air temperature, the optimal stop times can then be determined.

Zhou et al. [2002] used chilled water thermal storage tank to cater the cooling load for the building during the energy peak time. They select the optimal period of time for the tank to discharge by comparing with the price in every hour during the operation time.

Liu et al. [2008] also investigated the optimal start/stop strategy on variable-air-volume air condition in intelligence building. They used a neural network to predict the cooling preparation period. In the regulating period, the air volume was decided by means of feed forward control. The previous turning off period was determined in the way of penalty function.

(c) Optimal Chiller Sequencing

Chang [2005] proposed a method for using the Branch and Bound (B&B) method to solve the optimal chiller sequencing problem as well. A second order polynomial empirical model form was adopted for modeling the COP of a chiller with
considering the independent variable part load ratio of a chiller. Later on, the author also established a chiller model by using ANN in order to determine the optimal chiller sequencing based on the chiller power prediction result. It was assumed that the chilled water temperature and the cooling water temperature of the chillers within a chiller plant are the same. By applying ANN, it is no need to measure the chilled water flow rates for chiller power prediction.

Sun et al. [2009] developed a control algorithm to improve the reliability and the energy efficiency of chiller sequencing control based on the total cooling load measurement of centralized multiple centrifugal plants. The algorithm consists of three steps: (i) a fused measurement of building cooling load is used instead of using conventional method (i.e. the direct/indirect measurement); (ii) the maximum cooling capacity of individual chillers is computed online using a simplified centrifugal chiller model; and (iii) the online computed maximum cooling capacity is calibrated according to the quality of the fused measurement in order to deal with the direct measurement suffering from systematic errors. A case study was also performed in order to compare the performance of the improved chiller sequencing control strategy and the conventional chiller sequencing control strategy.

**Optimal Chiller and Thermal Energy Storage Sequencing**

Behl et al. [2012] proposed a green schedule to the chiller plants to reduce their peak aggregate power demand while ensuring safe operation of the thermal energy storage. This schedule can optimize the overall COP by utilizing the thermal energy storage and switching its operation between COP-optimal charging and discharging modes. The scheduling algorithm is based on backward reach set computation of the thermal
energy storage dynamics. The energy performance characteristics of the chiller and the thermal energy storage were modelled by using empirical modeling method.

(e) Optimal Chiller Loading Sharing

Chang [2004] and [2006] adopted Lagrangian method to solve the optimal chiller loading problem with compared the deficiencies of the conventional method (i.e. equal load rated method. The coefficient of performance (COP) of a centrifugal chiller is correlated with the part load ratio of a chiller to generate a chiller model. The COP function of each chiller unit within the multiple chiller plant is represented as a second order polynomial of part load ratio (PLR) of each chiller. The objective function is to find a set of chiller outputs which can meet the operating limits while maximizing the COP of chillers. The operating limit of the PLR of each chiller was set from 0.5 to 1.0. The results showed that the Lagrangain method reduces large energy consumption (i.e. increase the COP of the chillers) and is superior to the conventional method.

After that, Chang [2006] discovered that using Lagrangian method cannot solve the optimal chiller loading problem properly since the chiller performance (i.e. Power Input of a Chiller – PLR) curves simultaneously include convex functions and non-convex function. Therefore, the author adopted another optimization method called Simulated Annealing. This time, the power input function of each chiller unit within the multiple chiller plant is represented as a second order polynomial of part load ratio (PLR) of each chiller. The objective function is to find a set of chiller outputs which can meet the operating limits while minimizing the power input of chillers. The operating limit of the PLR of each chiller was set from 0.3 to 1.0. The
results showed that utilizing the Simulated Annealing method can solve the inherent chiller performance characteristics problem while Lagrangian method cannot. Apart from that, Simulated Annealing method can produce a highly accurate optimal chiller loading result.

Furthermore, Chang et al. [2009] employed another optimization method (i.e. Evolution Strategy) to solve the optimal chiller loading problem. Genetic Algorithm was also applied in order to compare the result with using Evolution Strategy. The results also revealed that using Evolution Strategy can solve the optimal chiller loading as the power consumption models curves include convex functions and concave functions simultaneously. In addition, this method can also provide more accurate optimal loading result for a multiple chiller plant.

Another study was conducted by Jahanbani et al. [2008] using two optimization methods for solving optimal chiller loading for a multiple chiller plant. These two methods are Continuous Genetic Algorithm and Particle Swarm Optimization and they can also solve the power consumption models. Geem [2011] explored that using Generalized Reduced Gradient in solving the optimal chiller loading can have better result when compared with Genetic Algorithm, Simulated Annealing and Particle Swarm Optimization. Coelho and Mariani [2013] also utilized the approach of Firefly Algorithm (IFA) based on Gaussian Distribution function to determine the optimal partial loading ratio of each chiller in order to minimize the energy consumption of multi-chiller systems.
(f) **Optimal Chilled Water Supply Temperature Set Point**

Evolution Strategy was also utilized in solving the optimal chilled water supply temperature of each chiller within a multiple chiller plant [Chang 2007]. The general idea is the same as chiller loading sharing. A second order polynomial modeling method with the chilled water supply temperature in chillers as input function and the power input as output function. Later, Chang and Chen [2009] used a Hopfield Neural Network to determine the optimal chilled water temperature for a multiple chiller plant. The author used a linear input-output model as a substitute for the sigmoid function which can eliminate the shortcoming of the conventional Hopfield Neural Network method. In addition, this method can also solve the chiller performance characteristics problem.

(g) **Optimized Condenser Fans Operation**

Chan and Yu [2002] developed an air-cooled reciprocating water chiller model with taking into account the operation of the condenser fans (constant speed) based on semi-empirical model to describe the energy performance of the chillers and explored the energy performance of the chillers under the head pressure control and the proposed condensing-temperature control in a subtropical climate. The chiller model were calibrated and validated using the operating data of an existing chiller plant. The set point of the condensing-temperature was reset in response to the temperature of air entering the condensers (i.e. outdoor air temperature) and was used to determine the number of the condenser fans staged in different operating conditions. It was discovered that using the proposed condensing-temperature control operating at variable lower condensing-temperatures based on the established operating mode of the condenser fans and compressors, the chiller power
consumption can be kept below 2.0 kW per TR throughout the entire range of outdoor temperature and part-load conditions, giving an average efficiency of 1.08 kW per TR. The energy can be traded-off due to cycling on more condenser fans and reducing compressor consumption.

Later on, Yu and Chan [2006] extended the study to air-cooled screw chillers with variable speed condenser fans. The set points of condensing temperature were used to determine the number and speed of condenser fans staged in various operating conditions. It was observed that adjusting the set point in response to the outdoor air temperature alone cannot achieve maximum COP when using variable speed condenser fans. Apart from that, comparing to head pressure control with constant speed condenser fans, the adoption of variable speed condenser fans with the condensing temperature control can have the chiller COP to increase by 4.0% to 127.5%, which corresponds to a decrease of 1.8kW to 154.6kW in the chiller power consumption.

Furthermore, the authors also evaluated the potential electricity cost savings of the air-cooled screw chillers serving the hotel with condensing temperature control, evaporative pre-coolers and variable speed condenser fans [Yu and Chan 2006]. Head pressure control was set as base case. It was ascertained that with regard to a chiller plant with a 15 year lifespan, the life cycle electricity cost savings were estimated at between HK$ 2,291,247 and HK$ 8,320,732 depending on the features used. The simple payback of the features could range from 1.07 to 3.32 years.

Yu and Chan [2008a] also established an air-cooled centrifugal chiller model to
investigate the optimization of the control of the condenser fans within the chillers in order to maximize their COPs. In order to investigate the percentage of energy saving when using different condensing temperature control methods, head pressure control with variable chilled water flow, condensing temperature control with constant variable chilled water flow, and condensing temperature control with variable chilled water flow were studied. It was found that the condensing temperature control with variable chilled water flow enable the COP to increase by 0.8 – 191.7%, depending on chiller load and outdoor air conditions. A cooling load profile of an office building was utilized to assess the potential energy savings resulting from the increased chiller COP and optimum staging of chiller and pumps. There was 16.3% to 21.1% reduction in the annual electricity consumption of the chiller plant within building.

The authors modified the DOE-2.1E chiller model (i.e. empirical model) to predict the change of chiller COP due to various set points of condensing temperature and pre-cooling of air stream entering the condenser for air-cooled centrifugal chillers serving an office building [Yu and Chan 2010]. Four operating schemes, head pressure control, head pressure control with a fixed mist generation rate, condensing temperature control with a fixed mist generation rate and condensing temperature control with an optimal mist generation rate, were studied. It was predicted that using optimal water mist control with condensing temperature control could have a 19.84% reduction in the annual electricity consumption of the system.

**(h) Optimization of Cooling Tower Fans**

optimal control strategies by utilizing the quadratic relationships for the chiller and water systems. Two basic methodologies for determining optimal or near optimal independent control variable/set point in the system to minimize the instantaneous operating cost of the overall chiller plant were introduced [Wang and Ma 2008]. One was the component model-based nonlinear optimization algorithm that the power consumption of chillers, cooling towers, condenser and chilled water pumps as well as supply and return air fans were expressed as quadratic relationships. This methodology was used as a simulation tool for exploring the entire system performance [Wang and Ma 2008]. The other method was a system-based for near optimal control that an overall empirical cost function of the total power consumption of the chiller plant was developed using a quadratic function. Later on, the method was further developed by Braun [2007] to establish a general control algorithm for cooling tower in cooling plants with electric and/or gas-driven chillers. This control algorithm took into account optimal fan sequencing, criteria for optimal tower airflow, chiller cost sensitivity to tower airflow and cooling tower fan cost sensitivity to tower airflow.

Braun et al. [1989b] also demarcated several guidelines for near optimal control of chilled water system without significant energy storage. Apart from that, they also identified the optimal supervisory control of the chilled water systems was primarily a function of the total cooling load and ambient wet-bulb temperature. These results formed the basis to develop a near optimal control algorithm for cooling towers [Braun and Diderrick 1990]
(i) **Chiller Plant Global Optimization**

Ma and Wang [2011] formulated an optimal strategy using simplified models of major components and the genetic algorithm for determining a set of optimal set point for a central plant consisting of chillers, cooling towers, water pumps and heat exchangers as well as air-handling units. The simplified models, with linear in parameters characteristics, are used to predict the energy performance of each of the components (i.e. chillers, cooling towers etc.) of the central chiller plant under different operating conditions and response to the changes of the control settings. The control settings are condenser water return temperature set point, chilled water supply temperature set point and heat exchanger water supply temperature set point. In order to enhance the accuracy of the models, the recursive least square (RLS) estimation technique with exponential forgetting is used to identify and update the model parameters online. The results demonstrated that using the proposed strategy can save about 0.73% – 2.55% daily energy of the system as compared to a reference strategy using conventional settings.

Wang et al. [2007] developed a model-based supervisory control strategy for online control of the chiller plant (including chillers and cooling towers). The semi-empirical models were established to simulate the energy performance of the chillers and cooling towers in response to different working conditions as well as control settings. The Performance Map and Exhaustive Search method (PMES) was adopted to determine optimal set point of the condenser water return temperature under different working operation with least total power consumption (i.e. sum of chiller power input and cooling tower fan power input). Two other control methods were selected to compare the result of the proposed method. The performance of the
Fix Approach Temperature (i.e. difference between the condenser water return temperature and the outdoor wet-bulb temperature) at 5.0°C was used as benchmark. The Near Optimal method was adopted to compare the PMES Method as well as Fixed Approach Temperature at 5.0°C. The results showed that using PMES method in a typical summary day could have 1.221% energy saving comparing Fixed Approach method while using Near Optimal method could have 0.239% energy saving.

A model-based optimization strategy for the condenser water loop of centralized chiller plant with variable speed control was developed [Lu et al. 2004]. The plant includes chillers, cooling towers and condensing water pumps. The objective is to minimize the overall power consumption of the plant. Empirical model was adopted to model the energy consumption of chiller, cooling tower fan and condensing water pump. A Modified Genetic Algorithm was used to determine the optimal set points of the centralized chiller plant under different operating conditions. The optimal set points are condensing water flow rate (i.e. condensing water pump speed) and heat rejection air flow rate (i.e. cooling tower fan speed). Two conventional methods (i.e. the fixed mass flow rates of condensing water and heat rejection air flow rate and the fixed mass flow rates of condensing water with varied heat rejection flow rate) were selected to compare with proposed optimal method (i.e. variable condensing water flow rate and heat rejection air flow rate).

Later, Lu et al. [2005] also carried out same model-based optimization strategy for centralized chiller plant with taking into account the chilled water loop and the condenser water loop. The heat exchangers, cooling coils and cooling towers are
given in the constraint section. An Adaptive Neural Fuzzy Inference System (ANFIS) was used to find the optimal pressure set points for chilled water pumps and air-handling unit cooling coil fans and to simplify the chilled water networks and supply air duct networks.

Yu and Chan [2008b] had a further study on investigating the energy performance of chiller and cooling tower systems integrated with condenser water flow and optimal speed control for tower fans and condenser water pumps. The semi-empirical mathematical model was adopted for simulating the energy performance of water-cooled centrifugal chillers and cross-flow cooling tower in order to assess how different control methods of the cooling towers and condenser water pumps influence the trade-off between the chiller power, chilled water and condensing water pump power, cooling tower fan power and water consumption under various operating conditions. The constant speed and variable speed configuration (i.e. condenser water pump and cooling tower fan) with fixed condenser water entering temperature, fixed approach temperature and optimal condensing water entering temperature for maximum system COP were explored. The chilled water flow rate and chilled water supply temperature were set at design values. Load-Based Speed Control (i.e. based on part load operating condition of the chiller) was introduced to control the speed of the cooling tower fans and condenser water pumps in order to achieve optimum system performance (i.e. system coefficient of performance) by regression analysis.

The authors further extended the study to all variable speed chiller system with primary loop only which can operate much more efficiently at part load in response
to changes in building cooling load \[\text{Yu and Chan 2009}\]. With applying the Load-Based Speed Control for all variable speed chiller system serving an office building, the annual total electricity use by 19.7% and annual water use by 15.9% relative to the corresponding constant speed plant.

\textbf{Lau et al. [1985]} conducted a study on using various control strategies including chilled water set point reset, chiller control sequencing, cooling tower fan speeds and condenser pump flow rates with adopting TRNSYS simulation tool in order to determine how much energy saving could be achieved when adopting different optimal control strategies for chilled water plant. Computer models of the chilled water system based on empirical curve fitting (except cooling tower) were developed in this study while an effectiveness model based on manufacturer’s data was used for the cooling tower.

\textbf{Yao et al. [2004]} developed an empirical model for optimal operation for the large cooling system serving the residential building. System COP was set as optimization problem in order to analyze the effect of energy saving of the cooling system.

\textbf{Chow et al. [2002]} integrated an Artificial Neural Network (ANN) and Genetic Algorithm (GA) to determine the optimal control of the absorption chillers system. ANN was used to model the system characteristics of the system while GA was utilized to determine the total energy cost with taking into account the fuel cost, the electricity cost, chiller and chilled water pump electric power, cooling water pump electric power as well as auxiliary electric power. Three optimal control cases were studied: (a) Constant Chilled and Cooling Water Flow Rates, (b) Variable Cooling...
Water Flow Rate (30% - 120%) with Constant Chilled Water Flow Rate and (c) Variable Cooling Water Flow Rate (30% - 120%) and Variable Chilled Water Flow Rate (50% - 120%).

Fong et al. [2006] developed a simulation optimization approach for effective energy management of building HVAC systems by adopting an evolutionary programming to handle the discrete, nonlinear and highly constrained optimization problems and an empirical chiller model and a simplified cooling coil empirical model. These models were then utilized to predict the system energy and environment performance. The simulation result showed that with using the optimized set points of the chilled water supply temperature and supply air temperature, 7% energy savings (monthly basis) in a subway station could be achieved.

Kusiak et al. [2010] used eight data-mining algorithms to model the nonlinear relationship among energy consumption, control settings (supply air temperature and supply air static pressure) and a set of uncontrollable parameters. The multiple-linear perception (MLP) ensemble outperformed other models tested in the study. The model was used to solve with a particle swarm optimization algorithm.

Taylor [2012] made use of empirical models for developing optimized control sequences for primary-only variable chilled water plant. All the near optimal control set points are correlated to approach temperature, chilled water flow ratio, chilled water supply temperature, condenser water flow ratio, condenser water return temperature, cooling degree-days based on 65°F, Differential pressure, chiller efficiency, cooling tower efficiency, Integrated Part Load Value (IPLV),
Non-Integrated Part Load Value (NPLV), range temperature, part load ratio of the chiller plant and outdoor web-bulb temperature.

1.4.4 Deficiency of Model-Based Supervisory Control Method with Optimization Techniques in Real Chiller Plants

In general, the local/global optimization is a branch of applied mathematics which based on a simulation model to evaluate the outcome of a certain process to find a candidate solution set of input parameters such that the outcome is the vertex of a concave or convex function. The candidate solution set normally is accepted as a set of coincident parameters input to the simulation model which produces the most effective outcome with the least input of resources. There exists numerous algorithms for candidate solutions in applied mathematics. The usual objective is to seek the candidate solution set for the optimum point. If the model has a comprehensive or intensive set of input parameters, the optimization process may end up millions of simulations in order to exhaust the possible values within the applicable range of each of the parameters to form an input set. Therefore, various procedures are used to reduce the number of simulations possible to generate the candidate solution set. Monte Carlo algorithm is one of the common optimization approaches [Fishman 1997]. Other algorithms are possible but often requires massive computation resources and time.

The reasons that many of successful research optimization algorithms have not been able to be applied in real chiller plants because:

- Optimization can be successful in laboratories or limited experiment in chiller
plants if the parameters in consideration are less or ignored and that the analysis are broadly based on simulation. In real practice, there are too many parameters involved which render these developed protocols not realistic.

- All these algorithms aim to identify candidate sets for the optimum point. If the numbers of parameters are limited to a few, the algorithms are possible for development. When the performance of the system or equipment such as the chiller is considered, the coincidental occurrence of the large number of parameters is impossible.

- In the optimization of a system or an equipment performance, a chiller, for example, it is best to be tested under the sequential conditions as in doing simulations. It is difficult to set up such comprehensive conditions in laboratories, not to mention about testing the equipment on site.
1.5  TYPICAL PROBLEMS OF CHILLER OPERATING DATA FROM BUILDING MANAGEMENT SYSTEM (BMS)

For existing buildings, Building Management System (BMS) associated with various types of sensors is very crucial and helpful in supervising and controlling the operating performance of the chiller system and components by measuring the operating parameters or variables such as: temperature, flow rate and pressure difference of working fluid as well as power consumption of HVAC systems or equipment etc. As a result, different operating ranges (i.e. from partial load to full load conditions) of the chiller system or equipment can be fully monitored and logged in BMS via the associated sensors (i.e. sensing points). This also facilitates a room for the building services engineers or researchers to understand how to achieve or determine the optimal performance of the chiller system based on the logged data. Nevertheless, many studies indicated the fact that no BMS with associated sensors are well designed or maintained. This section summarizes the typical problems of chiller operating data from BMS.

1.5.1  Inadequate Sensing Points Provision

It is no doubt that adequate sensors provision for monitoring the performance of the chiller system, by logged the major operating variables of the chiller system, should be required so that sufficient data can allow the building services engineers or researchers to conduct analysis for maintaining or achieving the chiller in optimal performance condition in terms of energy saving. However, many studies reported that instrumentation provision was in general inadequate. Yik and Chiu [1995] did a survey of 30 central chiller plants regarding the provision of instrumentation in
chiller plants in Hong Kong. The survey results indicated that instrumentation provision was inadequate. Another similar study carried out for an academic institution and a deluxe hotel [Deng et al. 1995] also reported that the instrumentation in both buildings were generally insufficient. This limitation also constraints the engineers or researchers to develop a chiller model (physical or empirical model base) using the logged operating data of the chiller system for calibration.

Moreover, many BMS only logged the operating data of the chiller system focusing on water-side only (chilled water and condensing water) and the power consumption of the system while the refrigeration side (e.g. refrigerant saturated evaporating or condensing temperature etc.) of the system is general neglected. This phenomenon can be found quite often in Hong Kong. In fact, the operating data of the refrigerant side can be monitored and displayed in the local control panel of the chiller. Due to the capital cost investment issue, most of the building owners in Hong Kong are not willing to pay for integrating the refrigeration side of a chiller to BMS. Despite the COP (for chiller assembly) or (System COP) for entire chiller plant can be determined, it is difficult to monitor the performance of individual components of the chiller. Comstock and Braun [1999] did an experimental study to investigate the impacts on the power and COP of the chiller corresponding to the four level of severity of the fault. They categorized the reduction of the heat transfer coefficient of the condenser in to four levels (i.e. by reducing 12%, 20%, 30% and 45% of the condenser tubes). The results indicated that the COP of a chiller could be decreased by 0.8%, 0.9%, 1.9% and 4.1% respectively. It proves that monitoring the refrigerant saturated condensing temperature is vitally important. Cleaning action can be early
taken when the degradation of the performance of the condensers is serious since it will affect the COP of the chiller. As a result, it is very difficult to assess the actual performance of the chiller system.

Furthermore, this situation also incurs in future problem when the building owners want to install additional measurement sensing points for the system or equipment after the operation and maintenance (O&M) period. No doubt, an unavoidable high additional cost charged by the sub-contracting firms will be the result (e.g. site constraint etc.). Although this phenomenon has been discovered since 1990’s, the problem still exits nowadays. Apart from inadequacy, the common design practice of the building services engineers in Hong Kong will normally over-design (i.e. over-provide) the measurement sensing points for monitoring the system or equipment without rationale. Consequently, a lot of useful and un-useful data will then be collected that makes trouble to the building facility managements as well as O&M staffs to manage and deal with them.

1.5.2 Sensor Faults

BMS is always suffered from a lot of faults. For example, sensor faults due to soft fault (i.e. drifting, biases, and precision degradation) and hard failure (i.e. complete failure), data transmission lost and capacity constraint of the computer sever or database etc. [Bitondo 1999; Jia 2002; Wang and Chen 2004]. Apart from the problems of the BMS data (namely the term “raw data”), the quality of the as-installed measurement instruments found in chiller plants in Hong Kong were generally inaccurate [Deng et al. 1995; Yik and Chiu 1995]. Consequently, these cause a big discrepancy between the estimated results using the simulation models
and sub-metering actual measurements. For chiller sequencing control, measuring the cooling load by the product of the differences between chilled water return and supply temperature and chilled water flow rate is widely adopted. The differences between the chilled water return and supply temperature is usually small with a design value of around 5°C only. Nonetheless, a research study reported that a bias error of 1°C in both supply temperature and return water temperature sensors might result in up to a 40% error in the total cooling load measurement [Wang and Cui 2005]. This incurs poor chiller sequencing control in terms of energy efficient may be the result. In the meantime, this phenomenon is quite often reported from the facility management (FM) team in many buildings in Hong Kong. Therefore, it implies that it is impossible to determine the optimal performance of the chiller plant before without conditioning the raw data.

1.5.3 Data Synchronization Problem

For the sake of data analysis, data synchronization to merge the logged data from BMS in order to generate a data set in a same time manner (i.e. same instant time) is essential. A typical chiller system generally consists of a lot of associated equipment (e.g. chillers, chilled water pumps, condensing water pumps as well as cooling towers etc.). Different types of sensors may be installed for monitoring each of the equipment. In practice, different sensors may have different vendors or manufacturers. At the same time, the setting for time interval of data logging of different sensors (i.e. operating variables of each of the equipment) may also be different. Furthermore, data lost is also often occurred in BMS. It is therefore, this situation causes a problem of data synchronization.
1.6 OBJECTIVES OF THIS RESEARCH

In reviewing of the Hong Kong government policy and attitude regarding the energy use in centralized chiller plants within buildings, it was discovered that their enforcements are not enough and still vague. Understanding the characteristics of the energy consumption of chiller plant within building enables the building operators or engineers to address appropriate optimization schemes for chiller plant in order to suit for different situations. Nevertheless, there is no any well-organized protocol or method for analyzing the annual energy consumption of chiller plant within building. In recent decades, a lot of chiller modeling techniques and chiller plant optimization methods have been well established. Those chiller plant optimization works mainly rely on engineering models in which the works are more academicals rather than practical since a lot of assumptions are made behind and the phenomena of the performance degradation of each component of a chiller are always neglected. In addition, insufficient building information and sensing points/data as well as invalid operating data of chillers for monitoring the operating conditions of chillers for optimization works are always found in many buildings in Hong Kong. As a result, those optimization works for chiller plant become impractical.

The specific objectives of this study are summarized as follows:

**Objective 1**

- To further revise a mathematical model for simulating the thermo-properties of R134a under different operating conditions in order to facilitate for chiller optimization analysis.
Objective 2

- To study the energy signature of different types of air-conditioned buildings in Hong Kong for building energy performance analysis based on statistical model approach. The difficulties in conducting the analysis are also examined.

Objective 3

- To introduce a method for dissection of the building energy consumption using engineering modeling technique for building energy performance analysis as well as chiller plant performance analysis.

Objective 4

- To develop a data logic management protocol for individual chiller performance analysis. The typical problems of the BMS for chiller performance analysis are also studied.

Objective 5

- To optimize the operating performance of individual chiller with introducing a new concept of Heuristic-Engineering-Statistical (HES) approach for data mining protocol.

Objective 6

- To establish an electronic web-based integrated analysis scheme for chiller performance optimization.
1.7 STRUCTURE OF THESIS

This thesis is structured in eight chapters in such a way that each chapter will bring out the outcome of the objectives stated in Section 1.6 supported with research results. Each chapter is briefed as follows:

Chapter 1 is an introduction chapter that reviews the geographic, climate and demographics as well as energy consumption patterns over the past decade of Hong Kong. These provide clues that the geographical, climatic and demographical environments are different from other countries that may influence in energy consumption patterns in Hong Kong. In addition, as the energy consumption due to the air-conditioning system in commercial sector, an active action of the Hong Kong government is therefore vitally important. A review on the Hong Kong government policy and attitude on centralized chiller plants within buildings is studied. An overview on Hong Kong and International Standards for standard rating conditions and minimum requirement of chiller efficiency are also discussed. Apart from that, a literature review on chiller plant optimization approaches is also presented. As chiller plant optimization mainly relies on adequate sensing points and valid operating data of the chiller plant stored in the BMS, the typical problems of chiller operating data from BMS is highlighted. Finally, the objectives of this study are also presented.

Chapter 2 discusses a philosophy of HES approach and differences between HES approach and traditional research ways for optimization. In addition, a general overview on the fundamental principle of refrigeration cycle and the functions of the major components of a typical centrifugal chiller are also presented. Furthermore, a modification of R134a model, based on Wilson et al. [1988], for calculating the
thermal-properties under different working conditions is established and the accuracy of the model based on lab data is validated.

Chapter 3 emphasizes on the utilization of the chiller thermal performance lines based on building thermal performance lines for the assessment of energy consumption due to chiller plant. An energy signature for building energy performance analysis based on statistical model approach is reviewed. A deterministic climatic variable in developing the thermal performance line model based on thermo-dynamic model is studied and identified. The application of using statistical model for analyzing the energy performance of air-conditioned buildings in Hong Kong is demonstrated. Moreover, the typical problems normally encountered during the conduction of the building energy performance analysis are detailed discussed. Furthermore, a survey on the availability of the essential building information for building electricity consumption analysis is also presented. A method for dissection of the building energy consumption using engineering modeling technique for building energy performance analysis as well as the chiller plant performance analysis based on building energy signature are also suggested and studied.

Chapter 4 highlights the significance of the data logic management protocol. The establishment of the data logic management protocol for individual chiller performance analysis is explained. Based on the established data logic management protocol, an application of this protocol for two existing chiller plants in Hong Kong is discussed.
With combing the functions of the data logic management protocol discussed in Chapter 4, a new concept of HES approach for individual chiller optimization is introduced in Chapter 5. The concept of this approach is to optimize the performance of a chiller, by adjusting the operating condition at the preferable efficacy zone under low electricity consumption with high COP. An application of this approach for two existing chiller plants in Hong Kong is also presented. In addition, a parametric analysis on the controllable variables and KPIs of a chiller regarding on time series and part load condition as well as chiller efficiency signature are also studied.

Chapter 6 presents the development of an electronic web-based integrated analysis scheme for chiller performance optimization, named as Chiller Server, with its applications.

A conclusion of the research is drawn in Chapter 7 which also discusses the limitations of the research and needs for further research study.
Figure 1.10 Structure of Thesis
PHILOSOPHY OF HEURISTIC-ENGINEERING-STATISTICAL APPROACH

AND

OVERVIEW ON TYPICAL CENTRIFUGAL CHILLER

The first section of this chapter discusses the HES approach for chiller plant optimization in detail. The differences between traditional research ways and HES approach for optimization are also highlighted. The second section presents a general overview on the typical centrifugal chiller. The key performance indices for describing the performance of each major component of a centrifugal chiller are reviewed. A mathematical refrigerant model for predicting the thermo-properties of R134a at different working conditions with its accuracy, based on lab data, is presented in the last section of this chapter.


2.1 HEURISTIC-ENGINEERING-STATISTICAL APPROACH FOR OPTIMIZATION

2.1.1 Philosophy of Heuristic-Engineering-Statistical Approach for Optimization

In real building operation, these research successful algorithms (see Section 1.4.2) have not yet been developed into commonly applied control algorithms for building energy performance. The computation resources requirement deters a real time optimization development in building services system operation. In this study, a HES protocol is developed. This protocol makes the best use of the data collected from the building data acquisition system often named as BMS because these systems not only monitor but also control. The data acquired from the BMS will go through a statistical analysis to find out the numeric relationship of any combination of data, usually presented in two dimensional graphs with a third dimension superimposed onto the graph as family curves. These family curves are generated from engineering equations or simple models which are often key performance indices for the optimized parameter or system. These curves can show the trend of the change of the optimized solution. Heuristic, according to Dictionary.com, means “to learn, discover, understand, or solve problems based on experimentation, evaluation or
trial-and-error methods.

In this HES approach, the statistical relationship of the chosen parameters form the ‘optimization map’. This map can be divided into optimization zones. The distribution of any controlling parameter superimposed on this ‘map’ is then plotted as frequency distribution charts. These charts for the highly optimized zones will be examined in order to determine the most prevalent value which is interpreted as an optimized setting of that controlling parameter. Although the most optimized setting of each controlling parameter can be set, it does not imply that the candidate solution set is the collection of these parametric values because these values are not coincidental.

The key points of the working principle of the Heuristic-Engineering-Statistical Approach are highlighted below:

- Each set of data acquired on-site is considered as an experimental set up. This argument is true because under the slow change of conditions due to environmental climatic conditions, load profiles and etc., each set of data can be considered as a steady-state condition of an experiment.

- If a building and its system have been operated for a while after hand-over, the operations of the building and the system are at annual cyclic state. If data is collected within a year continuously and at reasonable sampling rate, for example, hourly data, the building and its systems are considered to have gone through thousands of experiments at steady-state conditions. This is equivalent to have gone through a Monte Carlo process of experiments.

- Any adjustment of an input set to achieve a more optimized condition, the outcome is likely found by matching data of the acquired data in order to test
for improvement.

- Instead of striking for the optimum point which is almost impossible in real life, the regions of low optimum cluster of operation sets are identified. Within these regions, the controllable parameters are found from the engineering family curves on the "optimization map". The controllable parameter can then be adjusted to the "prevailing values" in order to move these points from the less optimized zone to the more optimized zone in the "optimization map". Thus, the building and the system can be "controlled" to operate under more optimized conditions.

The discussion in the last point above is based on the point by point trial by real life experiments. Knowledge is gained from the data mining process through the statistical and engineering analysis. Hence, a heuristic procedure is developed to make the less optimum operating conditions be controllable to more optimum operating conditions. The HES process is elaboratively used in chiller optimization in Chapter 5.
2.2 OVERVIEW ON TYPICAL CENTRIFUGAL CHILLER

2.2.1 Fundamental Principle of Refrigeration Cycle and Major Components

Basically, a centrifugal chiller is a machine that removes the heat in the colder (lower potential) to warmer (higher potential) direction. In other words, a chiller is to remove the heat from a liquid (e.g. chilled water) to other medium (e.g. atmospheric air or condensing water) via a vapour compression refrigeration cycle. The liquid can be circulated through a heat exchanger to cool air or equipment as required.

In the refrigeration cycle, it consists of five processes. The first process is evaporation (from Point 6 to Point 1) that the refrigerant evaporates at a constant temperature with changing from a liquid state to a saturated vapour state by absorbing the heat from the chilled water via the evaporator.

After the evaporation, the saturated vapour of the refrigerant from the outlet of the evaporator will then be further heated up to superheated state due to the heat generated from the compressor motor (from Point 1 to Point 1’).

Before entering into the compressor, the superheated refrigerant will be throttled by
the inlet guide vane at part load condition isentropically i.e. without changing the specific enthalpy content (from Point 1’ to Point 2).

The next process is polytropic compression process that the superheated refrigerant will be compressed and discharged at a high temperature and pressure. The process is due to the power consumed by the compressor is converted into heat in the refrigerant (from Point 2 to Point 3).

In the condenser, the superheated refrigerant will be desuperheated to the saturated vapour state and condensation will occur that allows the saturated vapour refrigerant changing to saturated liquid state (from Point 3 to Point 4).

The refrigerant will also be further sub-cooled (from Point 4 to Point 5). Lastly, the sub-cooled refrigerant expands isentropically before entering into the evaporator (from Point 5 to Point 6).

Note: In Figure 2.1, the red line denotes a saturated line. The left hand side of the red line represents the sub-cooled region while the right hand side represents the superheated regions. In the middle of the dome (i.e. saturated line), it means mixture
region (i.e. liquid and vapour constituent).

![Typical Refrigeration Cycle for Single-Stage Centrifugal Chillers](image)

**Figure 2.1** Typical Refrigeration Cycle for Single-Stage Centrifugal Chillers

2.2.2 Configuration of Typical Centrifugal Chiller

In general, the major components of a centrifugal chiller consist of five parts: (a) Evaporator, (b) inlet guide vane, (c) Compressor, (d) Condenser and (e) Expansion Device. Their major function is to facilitate a refrigeration cycle, taking place in the refrigerant circuit of a chiller, for removing the heat from building to atmosphere. In the refrigeration cycle, it mainly embraces five processes and these processes are corresponding to each major component of the chiller. They include:
2.2.2.1 Evaporator

The heat is removed from the building via the chilled water to the liquid refrigerant with low temperature and pressure taking place in the evaporator. The process is called evaporation (latent heat transfer only – i.e. no temperature of the refrigerant is changed). After the evaporation, the liquid refrigerant will change to vapour state. No liquid state of refrigerant entering into the compressor is allowed since it will damage to the components of the compressor.

The evaporator is normally a shell and tube configuration. A lot of copper/aluminum tubes installed inside the evaporator that allow the chilled water entering into evaporator with transferring the heat to the refrigerant in the shell. Flooded-Type Evaporator is widely adopted in centrifugal chiller.

2.2.2.2 Inlet Guide Vane

Before the refrigerant (with vapour state) from the outlet of the evaporator entering into the compressor, throttling process will be taken place due to the inlet guide vane installed at the inlet of the compressor. No work input is required (i.e. adiabatic process). The effect of the throttling process depends on the degree of opening of the inlet guide vane. The function of the inlet guide vane is to modulate the required
amount of refrigerant flow by throttling before entering into the compressor.

2.2.2.3 Compressor

Basically, the major function of the centrifugal compressor is to rise up the temperature and pressure of the vapour refrigerant to the condenser by converting the velocity pressure to static pressure of the refrigerant due to compression. Centrifugal compressor consists of impeller(s) and diffuser. The vapour refrigerant enters into the compressor and the impeller is used to increase the velocity pressure of the refrigerant by centrifugal force. The outlet velocity from the impeller can be divided into two parts: (a) Tangential Velocity which is directly proportion to the rotational speed of the impeller and the diameter of the impeller, (b) Radial Velocity which is directly proportional to the mass volume of the vapour refrigerant [Trane 2009]. After that, the vapour refrigerant will be discharged, with high static pressure due to the diffuser, to the condenser. The vapour refrigerant will change to liquid state before entering to the expansion device.

A centrifugal compressor can be classified into two different types: (a) Direct-Drive Type and (b) Gear-Drive Type. For the Direct-Drive Type, the rotational speed of the impeller will be equal to that of motor. For the Gear-Drive Type, the rotational speed
of the impeller is faster than that of the motor that is reflected by the gear ratio of the
gearbox. Another method to categorize the types of compressors is based on the
connection method between the motor and the compressor. They are: (a) Open Type
and (b) Semi-Hermetic Type. For Open Type, the compressor is completely
separated from the motor while they are connected via a shaft with coupler. The heat
generated from the motor is removed either by plant room air or water. The
Semi-Hermetic is that the motor is semi-enclosed inside the compressor in which the
refrigerant can be used to cool down the heat generated from the motor. Nowadays,
in order to enhance the compression efficiency and the minimize the size of the
compressor, centrifugal compressor may also consist of multi-impeller for
multi-stage compression.

2.2.2.4 Condenser

The heat is removed from the refrigerant to the condensing water (for cooling tower
and then discharged to atmosphere air) or the atmosphere air directly (for air-cooled
condenser) taking place in the condenser. The process is called condensation (latent
heat transfer only – i.e. no temperature of the refrigerant). After the condensation, the
vapour refrigerant will change to liquid state before entering the expansion device.
2.2.2.5 **Expansion Device**

Expansion device is used to throttling the refrigerant from high temperature and high pressure liquid state to low temperature and low pressure liquid state. No work input is required. Another function of the expansion device is to throttle the refrigerant according to the required amount of the refrigerant flow via the evaporator. Expansion device can be simply an orifice plant or an electronic expansion valve.

![Figure 2.2 Configuration of Typical Centrifugal Chiller](image_url)
2.2.3 Key Performance Indices of Typical Centrifugal Chiller

In this study, seven KPIs are introduced in order to monitor the condition of each component of a chiller during their routine operation. Each KPI can represent how the health condition of each component of a chiller is since each KPI have strong physical meaning and sensitive to a particular fault but insensitive to other faults. Moreover, each KPI is also derived from the fundamental of thermodynamic equations and they are summarized in Table 2.1.
<table>
<thead>
<tr>
<th>Component</th>
<th>Key Performance Index</th>
<th>Fault Diagnosis Classifier</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporator</td>
<td>Overall Heat Transfer Value ($UA_{ev}$) [kW/°C]</td>
<td>Evaporator Fouling</td>
</tr>
<tr>
<td>Condenser</td>
<td>Overall Heat Transfer Value ($UA_{cd}$) [kW/°C]</td>
<td>Condenser Fouling or Non-Condensable in Refrigerant</td>
</tr>
<tr>
<td>Compressor</td>
<td>Polytropic Efficiency ($\eta_{poly}$) [%]</td>
<td>Compressor Impeller Problems</td>
</tr>
<tr>
<td></td>
<td>Electro-mechanical Loss Efficiency ($\eta_m$) [%]</td>
<td>Bearing or Shaft problems or Excessive Lubricating Oil</td>
</tr>
<tr>
<td>Expansion Device</td>
<td>Discharge Coefficient ($C_o A_o$) [kPa]</td>
<td>Distortion of Expansion Device</td>
</tr>
<tr>
<td>Refrigerant</td>
<td>Refrigerant Mass Flow Rate ($m_r$) [kg/s]</td>
<td>Refrigerant Leakage Problems</td>
</tr>
<tr>
<td>Overall Chiller</td>
<td>Chiller Coefficient of Performance ($COP$)</td>
<td>Energy Consumption Impacts</td>
</tr>
</tbody>
</table>
2.2.3.1 Overall Heat Transfer Value of Evaporator

Evaporator is a heat exchanger that is used for transferring the heat from chilled water to refrigerant. It can be suffered from rusting, debris from water etc. causing fouling when it is operated for a certain period of time. The evaporator fouling can affect the heat transfer efficiency of the evaporator.

Generally, there are two methods to model the performance of the heat exchangers:

(a) the Effectiveness-NTU (Number of Transfer Units) Method and the Overall Heat Transfer Value Method [Bourdouxhe et al. 1994; Comstock et al. 1999; McQuiston et al. 2000]. The former method has a limited application since it does not take into consideration of the superheat temperature and the subcooling temperature. Thus, this method is mostly applied to isothermal heat exchangers only [Yu 2004]. Apart from that, the overall heat transfer value of the heat exchanger is assumed to be a constant. In reality, the variations of the temperatures of the operating fluids (i.e. chilled water and refrigerant) are influenced by the operating condition of the evaporator and the condenser. Therefore, using Effectiveness-NTU may not be capable to model the performance of the heat exchangers under full load and part load condition. The Overall Heat Transfer Value Method works well for a single-pass counter-flow heat exchanger without phase change and other types of heat
exchangers under steady-state flow conditions. Overall Heat Transfer Value of the heat exchanger, for shell and tube type, is the overall conductance product contains thermal resistance components including the local heat transfer coefficients of chilled water at the tube side and of refrigerant at the shell side as well as of the fouling factor of the tube materials. Poor performance of the heat exchangers due to the fouling can be directly reflected by Overall Heat Transfer Value of the heat exchangers (i.e. heat transfer efficiency of the heat exchangers). Therefore, the Overall Heat Transfer Value of the Evaporator is adopted as the key performance index representing the heat transfer efficiency of the evaporator.

The heat exchanger process taking place in evaporator can be described by the following energy balance equations:

**Chilled Water Side**  

\[ q_{rl} = m_w \ C_{pw} \ (T_{chwr} - T_{chws}) \]  

(2.1)

**Refrigerant Side**  

\[ q_{rl} = m_r \ (h_1' - h_6) \]  

(2.2)
Evaporator Assembly

\[ q_{rl} = UA_{ev} \cdot LMTD_{ev} \]  
\hspace{10cm} (2.3)

\[ LMTD_{ev} = \frac{(T_{chwr} - T_{chws})}{\ln (\left(\frac{T_{chwr} - T_{ev}}{T_{chws} - T_{ev}}\right))} \]  
\hspace{10cm} (2.4)

\[ UA_{ev} = \frac{q_{rl}}{LMTD_{ev}} \]  
\hspace{10cm} (2.5)

where: \( q_{rl} \) is the cooling load of the evaporator; \( m_w \) is the chilled water flow rate; \( T_{chws} \) is the chilled water supply temperature; \( T_{chwr} \) is the chilled water return temperature; \( C_{pw} \) is the specific heat capacity of water [i.e. 4.185kJ/kg/\degree C]; \( m_r \) is the refrigerant mass flow rate; \( h_1 \) is the specific enthalpy of refrigerant at the inlet of the compressor; \( h_6 \) is the specific enthalpy of the refrigerant at the inlet of evaporator; and \( LMTD_{ev} \) is the logarithmic mean temperature difference of evaporator.

2.2.3.2 Overall Heat Transfer Value of Condenser

Basically, condenser is also a heat exchanger that is similar to evaporator. The only difference is that it transports the heat from the refrigerant to the atmospheric air (for air-cooled type) or the condensing water (for water-cooled type). Unlike the evaporator, condenser for water-cooled type is connected to an open-loop circuit that
the condensing water will directly contact with atmospheric air. As a result, dust from air, debris and rusting affect the heat transfer efficiency of the condenser seriously when compared with evaporator (i.e. closed-loop circuit). Obviously, it can be imagined that how the significant of the key performance index representing the heat transfer efficiency of the condenser is. Therefore, Overall Heat Transfer Value of the Condenser is adopted as the key performance index of condenser.

The heat exchange process taking place in condenser can be described by the following energy balance equations:

**Condensing Water Side**

\[ q_{cd} = m_{cdw} \cdot C_{pw} \cdot (T_{cdws} - T_{cdwr}) \]  

(2.6)

**Refrigerant Side**

\[ q_{cd} = m_r \cdot (h_3 - h_5) \]  

(2.7)

**Condenser Assembly**

\[ q_{cd} = U A_{cd} \cdot L M T D_{cd} \]  

(2.8)
\[ LMTD_{cd} = \frac{(T_{cdws} - T_{cdwr})}{\ln \left( \frac{(T_{cdws} - T_{cd})}{(T_{cdwr} - T_{cd})} \right)} \]  
(2.9)

\[ UA_{cd} = \frac{q_{cd}}{LMTD_{cd}} \]  
(2.10)

where \( q_{cd} \) is the heat rejection rate of the condenser; \( m_{cdw} \) is the condensing water flow rate; \( T_{cdws} \) is the condensing water leaving temperature; \( T_{cdwr} \) is the condensing water entering temperature; \( h_3 \) is the specific enthalpy of refrigerant at the discharge outlet of the compressor; \( h_5 \) is the specific enthalpy of the refrigerant at the outlet of condenser; and \( LMTD_{cd} \) is the logarithmic mean temperature difference of condenser.

### 2.2.3.3 Polytropic Efficiency of Centrifugal Compressor

The compression efficiency of the compressor can be affected by the distortion of the impeller’s blade and base as well as the shape of vane’s blade and base and any change in surface smoothness during operation. Therefore, modeling the compressor efficiency of the compressor in order to reflect the performance of the compressor is vital.

To model the irreversibilities in the compressor, adiabatic efficiency defined as the ratio of isentropic work required to actual work can be considered [ASHRAE 1996].
Nevertheless, the drawback of using the adiabatic efficiency is that the same compressor may produce different adiabatic results with different refrigerants and also with the same refrigerant at different suction conditions. Therefore, using adiabatic efficiency of a compressor is not capable for representing the actual performance of the compressor. Cui and Wang [2005]; Jia and Reddy [2003] also discussed this issue and recommended that using a polytropic efficiency of the compressor to model the centrifugal compressor is more reasonable since the compression process in a centrifugal compressor is near a polytropic process (i.e. entropy is not constant during the compression process which is different from adiabatic process). Therefore, polytropic efficiency is used as a key performance index of the centrifugal compressor.

\[ \eta_{poly} = \frac{W_{poly}}{W_{com}} \]  \hspace{1cm} (2.11)

\[ W_{poly} = \ln \left( \frac{P_{cd}}{P_{ev}} \right) \left( P_{cd} v_3 - P_{ev} v_2 \right) / \left( \ln \left( \frac{P_{cd} v_3}{P_{ev} v_2} \right) / \ln \left( \frac{P_{cd} v_3}{P_{ev} v_2} \right) \right) \]  \hspace{1cm} (2.12)

\[ W_{com} = m_c (h_3 - h_2) \]  \hspace{1cm} (2.13)

where \( \eta_{poly} \) is the polytropic efficiency of compressor; \( W_{com} \) is the polytropic work.
consumed by the compressor; $P_{cd}$ is the condensing pressure of refrigerant; $P_{ev}$ is the evaporating pressure of refrigerant; $v_2$ is the specific volume of refrigerant after inlet guide vane throttling; and $v_3$ is the specific volume of refrigerant at the discharge outlet of compressor.

### 2.2.3.4 Electro-Mechanical Loss Efficiency

For typical centrifugal chiller, the motor is connected to the compressor externally via a shaft with coupler. The frictional losses in the transmission due to bearing friction of the shaft etc. is one of the major effects causing inefficiencies to the compressor. Excessive or inadequate lubricating oil may affect the transmission efficiency of the motor. Due to this reason, not all the electricity drawn by the motor can be totally transferred to the chiller impeller. In order to reflect the motor transmission efficiency, the Electro-Mechanical Loss Efficiency, defined as the ratio of power that the actual work input consumed by the compressor to the electricity power input to the compressor, is adopted as a key performance index for motor efficiency.

$$\eta_m = \frac{W_{com}}{P_{cc}}$$

(2.14)
where $\eta_m$ is the electro-mechanical loss efficiency [%]; $W_{com}$ is the actual work consumed by the compressor [kW]; and $P_{cc}$ is the chiller electricity power input to the compressor [kW].

### 2.2.3.5 Expansion Device

For typical centrifugal chillers, an orifice plate instead of using expansion valve is used as throttling device in order to maintain a certain level of pressure difference between condenser and evaporator as well as to control the refrigerant flow rate via the evaporator. According to Jia and Reddy [2003], the usual fault of the fixed orifice plate is partial blockage which can affect proper refrigerant flow within the refrigerant circuit. The performance of the orifice plant can be modelled by Discharge Coefficient. The Discharge Coefficient is defined as the product of the fluid friction coefficient (also is a function of refrigerant velocity/flow rate) and the total cross-sectional area of the orifice. Therefore, the Discharge Coefficient is adopted as a key performance index of expansion device.

The isentropical process taking place in expansion device can be described by the following energy balance equation:
\[ h_5 = h_6 \]  \hspace{1cm} (2.15)

2.2.3.6 Refrigerant Mass Flow Rate

The leakage of refrigerant may be occurred if the refrigerant circuit of a chiller is not connected properly. On the other hand, abnormal pressure difference between the condenser and evaporator may affect the refrigerant flow rate. Therefore, using the refrigerant mass flow rate as a key performance index to reflect the health condition of the refrigerant circuit of a chiller and to detect another abnormal condition of the pressure difference between the condenser and evaporator is preferable.

2.2.3.7 Coefficient of Performance

Coefficient of Performance \((COP)\) of a chiller can be used as an overall health indicator to reflect the faults made by overall components of a chiller during its full load and part load conditions.

\[ COP = \frac{q_{rl}}{P_{cc}} \]  \hspace{1cm} (2.16)
2.3 DEVELOPMENT OF REFRIGERANT MODEL WITH VALIDATION

In order to investigate the energy performance of the chiller with its associated components, understanding the thermal properties of the refrigerant at different states within the refrigeration cycle is a must. Nowadays, the refrigerant of R134a has been widely adopted in centrifugal chillers since this type of refrigerant is an environmental friendliness chemical (i.e. zero ozone depletion potential). Therefore, it is essential to model the thermal properties of this refrigerant for chiller performance analysis. The methodology for modeling R134a is presented with validation in this section.

2.3.1 Methodology

Wilson et al. [1988] presented a set of algorithms for modeling the thermal properties of refrigerant R134a with experimental data validation. Nevertheless, the model for calculating the specific enthalpy and specific entropy at saturated liquid state were not presented in the paper. Therefore, it is very difficult to use of those algorithms for further analysis especially for chiller performance analysis. Due to this reason, a computing programming “GNU Octave” is adopted in order to
determine the thermo-properties of R134a at saturated liquid state, saturated vapour state and superheated state conditions. The laboratory data of R134a presented by Wilson et al. [1988] are utilized for model validation. With minimizing the errors occurred in the results, some modifications on the algorithms proposed by Wilson et al. [1988] are applied.

### 2.3.1.1 Liquid State

**(a) Specific Volume at Liquid State**

\[
v_L = \left( \rho_c + \sum D_n (1 - T_r)^{n/3} \right)^{-1} \tag{2.17}
\]

where \(v_L\) is Specific Volume at Liquid State \([m^3/kg]\); \(\rho_c\) is Critical Liquid Density \([i.e. 512.2 kg/m^3]\); \(T_r = T/T_c\); \(T\) is Temperature \([^\circ K]\); \(T_c\) is Critical Temperature \([i.e. 374.25^\circ K]\); \(D_1 = 819.6183; D_2 = 1023.582; D_3 = -1156.757; and D_4 = 789.7191.\)

**(b) Specific Enthalpy at Liquid State**

\[
h_L = C_0 + C_1 T + C_2 T^2 + C_3 T^3 + C_4 \ln (P \nu_L) + C_5 (\ln (P \nu_L))^2 + C_6 (\ln (P \nu_L))^3 \tag{2.18}
\]

where \(h_L\) is Specific Enthalpy at Liquid State \([kJ/kg]\); \(T\) is Temperature \([^\circ K]\); \(P\) is
Pressure [kPa]; \( \nu_L \) is Specific Volume at Liquid State [m\(^3\)/kg]; \( C_n \) are Coefficients; \( C_0 = -414.26; \) \( C_1 = 4.0416; \) \( C_2 = -0.011301; \) \( C_3 = 0.000011955; \) \( C_4 = 34.165; \) \( C_5 = 5.6043; \) and \( C_6 = 0.33294. \)

(c) **Specific Enthalpy at Liquid State**

\[
s_L = C_7 + C_8 T + C_9 T^2 + C_{10} T^3 + C_{11} \ln (P \nu_L) + C_{12} (\ln (P \nu_L))^2 + C_{13} (\ln (P \nu_L))^3
\]

(2.19)

where \( s_L \) is Specific Entropy at Liquid State [kJ/kg]; \( T \) is Temperature [°K]; \( P \) is Pressure [kPa]; \( \nu_L \) is Specific Volume at Liquid State [m\(^3\)/kg]; and \( C_n \) are Coefficients;

\( C_7 = -0.88572; \) \( C_8 = 0.0085496; \) \( C_9 = -0.000019994; \) \( C_{10} = 0.00000018519; \) \( C_{11} = 0.10847; \) \( C_{12} = 0.011201; \) and \( C_{13} = 0.00052451. \)

2.3.1.2 **Vapour State**

(a) **Pressure**

\[
\ln P = A + (B / T) + (C T) + (D T^2) + E ((F - T) / T) \ln (F - T)
\]

(2.20)

where \( T \) is Temperature [°K]; \( P \) is Pressure [kPa]; \( A = 24.8033988; \) \( B = -0.3980408 \times 10^4; \) \( C = -0.2405332 \times 10^{-1}; \) \( D = 0.2245211 \times 10^{-4}; \) \( E = 0.1995548; \) and \( F = \)
0.3748473 \times 10^3.

\[(b) \quad \text{Specific Enthalpy at Vapour State}\]

\[h = h_0 + j_1 (P v - R T) + (C_{p1} T + C_{p2} T^2 / 2 + C_{p3} T^3 / 3 + C_{p4} T^4 / 4 + C_{p5} \ln T) + j_1 ((A_2 / (v - b)) + (B_3 / (v - b)^2 + (C_4 / 2(v - b)^2 + C_5 / 3(v - b)^3 + C_6 / 4(v - b)^4))) + j_1 e^{KT_r} (1 + KT_r) (C_2 / (v - b) + C_3 / (2(v - b)^2 + C_4 / (3(v - b)^3 + C_5 / 4(v - b)^4)) \]  \hspace{1cm} (2.21)

\[(c) \quad \text{Specific Entropy at Vapour State}\]

\[s = s_0 + (C_{p1} \ln T + C_{p2} T + C_{p3} T^2 / 2 + C_{p4} T^3 / 3) + j_1 R \ln ((v - b) P_1 / R T) - j_1 ((B_2 / (v - b)) + (B_3 / (v - b)^2 + B_4 / 3(v - b)^3 + B_5 / 4(v - b)^4) + j_1 K e^{KT_r} / T_c (C_2 / (v - b) + C_3 / 2(v - b)^2 + C_4 / 3(v - b)^3 + C_5 / 4(v - b)^4)) \]  \hspace{1cm} (2.22)

where \(v\) is Specific Volume Obtained by Iterative Solution of the Martin-Hou Equation at Given Temperature \(T^\circ\)K and Pressure \(P\) [kPa]; \(h_0\) is Specific Enthalpy at Reference Temperature [i.e. 59.11831 kJ/kg]; \(s_0\) is Specific Entropy at Reference Temperature [i.e. 0.291438kJ/kg\(^\circ\)K]; \(T_r = 374.25^\circ\)K; \(P\) is Pressure [kPa]; \(P_1 = 101.325\)kPa; \(j_1 = 1.0;\) \(R = 81.4881629 \times 10^{-3} \)kJ/kg\(^\circ\)K; \(b = 0.3455467 \times 10^{-3};\) \(K = 5.475;\) \(C_4 = 0;\) \(C_{p1} = -0.5257455 \times 10^{-2};\) \(C_{p2} = 0.3296570 \times 10^{-2};\) \(C_{p3} = -2.017321 \times 10^{-6};\) \(C_{p4} = 0.0;\) \(C_{p5} = 15.82170;\) \(A_2 = -0.1195051;\) \(A_3 = 0.1447797 \times 10^{-3};\) \(A_4 =
-1.049005 \times 10^{-7}; \text{ and } A_5 = -6.953904 \times 10^{-12}.

### 2.3.2 Validation

The validation of the accuracy for the refrigerant model for calculating the thermal properties of R134a based on the maximum percentage error is summarized in Table 2.2 and the graphical presentations for liquid state and vapour state are shown in Figure 2.3 (a) to (f) respectively. It is noted that the circle symbol and the triangle symbol shown on the graphs represent modeled and lab data respectively. As shown in Table 2.2, the percentage of the maximum error of the advanced refrigerant model for predicting the thermo-properties of R134a at different working conditions were fell within 0.01% to 0.05% for liquid state and 0.01% for vapour state. These results proved that the accuracy of this advanced refrigerant model for predicting the thermal-properties of R134a is significantly high.
Table 2.2  Summary of Accuracy (i.e. Percentage of Maximum Error) of Advanced Refrigerant Model (R134a)

<table>
<thead>
<tr>
<th>State</th>
<th>Thermo-Property Type</th>
<th>Maximum Error Percentage (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid</td>
<td>Specific Volume</td>
<td>0.01</td>
</tr>
<tr>
<td></td>
<td>Specific Enthalpy</td>
<td>0.05</td>
</tr>
<tr>
<td></td>
<td>Specific Entropy</td>
<td>0.05</td>
</tr>
<tr>
<td>Vapour</td>
<td>Pressure</td>
<td>0.01</td>
</tr>
<tr>
<td></td>
<td>Specific Enthalpy</td>
<td>0.01</td>
</tr>
<tr>
<td></td>
<td>Specific Entropy</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Figure 2.3 (a)  Comparison of Predicted Specific Volume at Liquid State of R134a between Lab Data and Modeled Data
Figure 2.3 (b) Comparison of Predicted Specific Enthalpy at Liquid State of R134a between Lab Data and Modeled Data

Figure 2.3 (c) Comparison of Predicted Specific Entropy at Liquid State of R134a between Lab Data and Modeled Data
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Figure 2.3 (d)  Comparison of Predicted Pressure at Vapour State of R134a between Lab Data and Modeled Data

Figure 2.3 (e)  Comparison of Predicted Specific Enthalpy at Vapour State of R134a between Lab Data and Modeled Data
Figure 2.3 (f) Comparison of Predicted Specific Entropy at Vapour State of R134a between Lab Data and Modeled Data
2.4 SUMMARY

A philosophy of HES approach and differences between HES approach and traditional research methods for optimization are firstly discussed and presented in this chapter. Before analyzing the potential for optimizing the performance of individual chiller or chiller plant, it is vital to understand the basic theory, working principle and major components of a chiller. This chapter also provides a general overview on the fundamental principle of refrigeration cycle and the functions of the major components of a typical centrifugal chiller. In addition, a mathematical refrigerant model for calculating the thermo-properties of R134a is established and validated (based on lab data) in this chapter. The accuracy of this refrigerant model is significantly high.
With the Hong Kong Government concerns recently, Building Energy Efficiency Ordinance (BEEO) Chapter 610 has been addressed and is in effect on 21 September 2012 [EMSD 2012]. In order to ensure that existing buildings are energy efficient, the Government also launched a Guideline on Energy Audit in 2007 as well as the Energy Audit Code in 2012 [EMSD 2007; EMSD 2012]. This document provides the general basic methodology for conducting energy audit as well as producing energy audit report. In general, one of the influential parameters affecting the energy consumption of a building is the outdoor air condition (i.e. contributing the building cooling load and influencing the operating efficiency of the chiller plant). Nevertheless, no such factor is taken into consideration, in regard to the building electricity consumption, in the document. It was commonly observed that most of the energy audit reports produced by the building services engineers or consulting companies in Hong Kong are rather superficial especially analyzing the energy consumption of the building over the past years. Most of them simply concluded that
the increase of building energy consumption over the past years were mainly due to the effect of global warming since the outdoor air temperature has been raising up over the years. No detailed investigation on what was/were the reason(s) incurring the increase of building energy consumption over the past years could be found and this situation was very often. Therefore, it is worthwhile to explore the between the outdoor air condition and the building energy consumption.

Apart from that, the electricity consumption of the chiller plant within building normally occupied for about 30-40% total of the building electricity consumption. Inefficient operation of the chiller plant will, no doubt, incur additional electricity cost. Notwithstanding the document of the Guidelines on Energy Audit in 2007 suggested what types of operating parameters of the chiller plant are monitored and recorded, no detailed methodology on investigating the performance of the chiller plant influencing the building electricity consumption with the changes of the outdoor air condition is suggested. Consequently, it is difficult to realize whether the performance of the chiller plant was working in energy efficient manner over the past years.

The first section of this chapter discusses the energy signature for building energy
performance analysis based on statistical model. The second section is the application part for demonstrating how to make use of the statistical model approach for energy performance analysis of air-conditioned buildings in Hong Kong. The third section highlights the typical problems normally encountered during the conduction of the building energy performance analysis. A survey on the availability of the essential building information for building electricity consumption analysis is also presented. The fourth section suggests a method for dissection of the building energy consumption using engineering modeling technique for building energy performance analysis. The final section delivers the method for analyzing the chiller plant performance based on building energy signature.
3.1 ENERGY SIGNATURES FOR BUILDING ENERGY PERFORMANCE BASED ON STATISTICAL MODEL

As discussed in Chapter One, the air-conditioning system accounts around 30% to 40% of the total building energy consumption while chiller operation consumes about 60% of the total of the air-conditioning system in typical commercial building. Therefore, understanding the energy use in air conditioning system is vital important for building energy saving. Indeed, the energy use in air-conditioning system is dominant weather dependent load in the building while the energy consumption of the lighting system, lifts and escalators and other electrical installations in buildings are depending on their usage pattern rather than the weather dependent. Thus, these installations are normally stable over a year and they can be regarded as base load.

In order to realize how the outdoor weather conditions influence the building energy consumption and the chiller plant energy consumption, using a simple method “thermal performance lines” (or energy signatures) is suggested since it can avoid numerous computational efforts with reasonable accurate results. In general, thermal performance lines method can be applied to predict energy consumption of buildings or chiller plants, to evaluate the effectiveness of the energy improvement measure in
the building or chiller plant, to compare how energy efficient is used between buildings by setting minimum requirement of the buildings or chiller plants and to detect malfunction in chiller plant [Yik et al. 1995; Yik and Sat 2001; Yu and Chan 2005]. Besides, using thermal performance lines can also be used to identify the thermal characteristics of different types of buildings.

### 3.1.1 Applications of Thermal Performance Lines Method

The concept of the thermal performance line is the best-fit straight line drawn through a series of points (i.e. curve-fit model) with using a statistical technique of least squares regression [Yik et al. 1995]. Each of the point represents a measurement of energy consumption over a stated period and the mean outdoor air temperature for that period. Instead of correlating the building energy consumption and mean outdoor air temperature, the thermal performance lines method was originally derived on the basis of monthly building energy consumption and degree-day data for heated buildings in cold climates. The base temperature upon which the degree-day data is developed 15.5°C [Levermore 1989; CIBSE 1991; Yik and Sat 2001].

In cold climate regions, two joint line segments will appear when plotting the mean
building energy consumption against mean outdoor air temperature for heated buildings [Levermore 1995]. The first segment is a downward sloping line illustrating the reduction in heating energy consumption with the increase in outdoor air temperature [Yik et al. 1995]. The second segment is a horizontal line which reflects the magnitude of the fixed, non-weather dependent load. The outdoor air temperature corresponding to the turning point between the two segments is regarded as balance point temperature. This is the outdoor air temperature under which the internal heat gains balance the heat losses and no heating is required.

While in sub-tropic climate regions, two joint line segments will appear when plotting the mean building energy consumption against mean outdoor air temperature for air-conditioned buildings. The first segment is the horizontal line that represents the base load (i.e. non weather dependent load – i.e. constant building energy consumption over a certain range of outdoor air temperature). The second segment an upward sloping interprets the increasing in cooling energy consumption with increase in outdoor air temperature. The outdoor air temperature corresponding to the turning point between two segments is called the balance point temperature. This is the outdoor air temperature under internal heat gain balance the heat loss, no heating or cooling is required in such buildings.
In earlier study, Levermore [1995] applied the thermal performance line method together with a balanced temperature and degree-days method with the application of a boiler model, a control model as well as an intermittent heating factor (based on CIBSE Building Energy Code) to study variations of the degree-days base temperature of 15.5°C to the heating system energy consumption under the influences of the control performance, the intermittent heating as well as the lighting, people and solar heat gains on two storey open plan office building.

_____________________

**Degree-Days:**

*It measures the deviation of the mean (average) temperature from a base temperature which is a temperature where neither heating nor cooling is used.*

**Heating or Cooling Degree-Days:**

*It means the deviation below or above the base temperature for heating or cooling respectively.*

**Base (Balance) Temperature:**

*The base temperature depends on building construction methods and climatic variables such as humidity, wind regime etc. for the particular region. Traditionally, the base temperature is normally assumed at 15.5°C in the UK for 18 regions while 18.3°C and 10.0°C are published by ASHRAE for*
heating and cooling degree-day data. Alternative threshold values are considered: In Bangladesh, 

Mourshed [2011] proposed 10.0°C to 20.0°C for heating degree-days and 10.0°C to 28.0°C for cooling degree-days. In Thailand, 24.0°C for the base temperature was considered [Parkpoom and Harrison 2008]. Ruth and Lin [2006] used 11.7°C for commercial sector and 15.6°C for residential sector for the state of Maryland in USA. Howden and Crimp [2001] selected 17.5°C as the base temperature for Sydney.
Yik et al. [1995] studied the influences of the climatic variables on the energy performance of the sea water-cooled chiller plants in two institutional building in Hong Kong with utilizing the thermal performance line techniques. The authors took into account the outdoor air dry-bulb temperature and moisture content as well as the global solar radiation as the independent variables in order to study the significance on the electricity consumption of the chiller plant. To select the optimal mathematical model form with high accuracy to represent the energy signature of the chiller plants, multiple linear regression model form was firstly selected to study. Nine predictor variables, including three independent variables and their square as well as their product terms, were included into the model. With using stepping multiple linear regression and variance inflation factor analysis methods, any insignificant predictor variables were eliminated. Nonetheless, the authors claimed that using the multiple regression model involves different weather parameters for two chiller plants (outdoor air dry-bulb temperature and moisture content for building A while outdoor air moisture content and global solar radiation for building B) which is difficult to inter-plant comparison whereas the best single independent variable models of both plants have outdoor air dry-bulb temperature and moisture content as the independent variables. Therefore, single independent variable model (i.e. simple linear regression model) was suggested and used for the chiller plant
performance comparison between two buildings.

Similar approach was applied for constructing the thermal performance lines for air-conditioned commercial buildings in Hong Kong in order to study how the climatic variables affecting the overall building electricity consumption [Yik and Sat 2001]. Twenty-six exiting building including office and commercial buildings were selected. In this case, the regression results indicated that on a comparative basis, the square of outdoor air temperature dry-bulb temperature emerged as the best climate variable for constructing the linear thermal performance line modes for the twenty-six building.

Lam [1998] selected three climatic variables, including monthly cooling degree-day total, monthly enthalpy-day total and monthly cooling radiation–day total to correlate the monthly electricity consumption of residential sector in Hong Kong for each year from 1979 to 1993 in order formulate a multiple regression model.

Lam and Li [2003] conducted a building energy consumption survey for 4 shopping malls in Hong Kong. Based on the collected annual electricity consumption data in GWh and kWh per floor are m² as well as the mean monthly temperature data, linear
regression models for the shopping malls were established.

Day [2003] presented a detailed analysis of measured cooling energy data from a large air-conditioned site in Perth, Western Australia with using thermal performance lines method and comparing the theoretical results for improving the use of cooling degree-days analysis for chiller energy consumption in building.

Lam et al. [2004] used five years weather data to produce average monthly cooling degree days for Hong Kong. With correlating between the electricity consumption, of 20 selected fully air-conditioned office buildings with centralized HVAC system, and the average monthly cooling degree days, a linear regression model for predicting the monthly electricity consumption of the selected buildings was developed. Furthermore, similar approach with taking into account the building envelope heat gain per unit floor area, internal load density to generate another regression model to evaluate the monthly electricity use per unit gross floor area per day.

Later on, Yu and Chan [2005] further to elaborate the thermal performance line method to assess the energy performance of the air-cooled chiller plant under
different operating strategies in hypothetical hotel. Four operating strategies are: (a) head pressure control, (b) head pressure control with evaporative cooler, (c) condensing temperature control and (d) condensing temperature control with evaporative cooler. In order to prove the product of outdoor air dry-bulb temperature and moisture content has high correlation with the electricity consumption of chiller plant, the authors studied the relationship between the building cooling load and the product of outdoor air dry-bulb temperature and moisture content, high coefficient of determination \(R^2\) of 0.81 was explored. This result made consistent with the study did by Yik et al. [1995]. The result also showed that using the condensing temperature control with evaporative cooler is the most energy efficient comparing with other operating strategies.

Ghiaus [2006] proposed a method by using the range between the 1\textsuperscript{st} and the 3\textsuperscript{rd} quartile of the quantile-quantile (q-q) plot to check the heat losses and the outdoor temperature distribution and to performance the linear regression in this range of the q-q plot for the office buildings having different geometrical locations (including Moscow, Rome, Vienna, Munich and Paris).

Later on, Lam et al. [2008] also did the same approach for establishing the
regression models for predicting the monthly average daily electricity consumption and the monthly electricity use per unit gross floor area. Unlike the previous study using average monthly cooling degree days, the authors adopted the principle component analysis of outdoor air dry-bulb temperature, outdoor air wet-bulb temperature, global solar radiation, clearness index and wind speed with clustering analysis to produce two principle components for describing 80% of the variance in the weather data over 28 years in Hong Kong. After that, regression methods were established based on the principle components.

Lam et al. [2009] adopted the principle component analysis of outdoor air dry-bulb temperature, outdoor air wet-bulb temperature, global solar radiation, clearness index and wind speed with clustering analysis to identify 18 typical day types for Hong Kong. With correlating the daily chiller plant electricity consumption with the corresponding day types using third order polynomial model form, three regression models for daily chiller plant electricity consumption, day-type-average daily chiller plant electricity consumption and monthly-averaged daily chiller plant electricity consumption were established.

A linear regression method was applied to estimate the potential impact of the
climate change, between the present period (1961 – 1990) and the future period (2040 – 2069), on the heating energy of 11 existing residential houses located in Montreal [Zmeureanu and Renaud 2008]. The heating energy use of the existing residential houses was based on the utility bills from the owner while climatic data were based on the prediction from the Canadian Centre for Climate Modeling and Analysis [CCCMA 2007]. Two energy signatures with using a linear regression method were proposed. For the energy signature developed from hourly or daily total of the collected dataset, the frequency of occurrence of each temperature bin throughout the heating season was taken into account for the annual energy use for heating. For the energy signature developed from daily average values of the collected dataset (i.e. monthly or bimonthly values of energy use divided by the number of days), the annual energy use for heating takes into account of the number of days and the monthly average outdoor temperature.

Belussi and Danza [2012] proposed a Multidisciplinary Energy Signature (MESH) approach for accurately characterizing the energy signature of the existing building with using heating system. The MESH approach combines mathematics, geometry and statistics. The steps of using MESH approach consists of four steps: (a) determination of the correlation of the energy consumption with the outside
temperature by using Pearson Correlation, (b) Identification of Outliers by adopting Grubbs Test, (c) Definition of the real energy signature by regression and (d) assessment of the concavity or convexity of the energy signature curve by using second degree function. Apart from that, the authors also introduced three terms for energy signature (i.e. real energy signature, design energy signature and energy signature by law) for the comparison of the actual energy performance of the building to the design case and minimum statutory requirements regarding building energy efficiency in Italy.

Matri-Herrero et al. [2013] used the energy signature by regressing the daily energy consumption and the meteorological data to determine the total heat loss efficient, the effective heat capacity and the net solar gain for nine public buildings in Spain. In the meantime, due to the thermal inertia of the building causing error in predicting the heat capacity of the building and in turns of the total heat loss efficient, data filtering process was applied in order to remove the dynamic data during the thermal inertia of the building regarding the non-working days and non-working days after holidays.

Nordstrom et al. [2012] studied the effect of U-Values on the building energy
performance for six single-family houses built between 1962 and 2006. A linear regression method was applied to reflect the energy signature of the houses based on the measurements of the total power used for heating and the indoor and outdoor temperature for each house during three winter months in northern Swedish. It was discovered that for two houses built in the same year, a house had smaller U-value, the slope of its energy signature was lower than that of the house having larger U-value.

Although the variations of the outdoor weather conditions influencing the electricity consumption of the building or the chiller plant can be simply reflected by using the thermal performance lines method, this method still has some limitations especially when identifying the energy management opportunities (EMOs) on various types of building services systems or installations during energy audit exercise. For instance, the chiller plant is used to deal with the building cooling load. Building cooling load consists of different components (including heat gains due to building façade, heat gains due to solar radiation via the glazing, heat gains due to ventilation and infiltration, heat gains due to internal occupancy, lighting as well as equipment). Each component of the cooling load needs energy to be removed. Therefore, identify the energy consumption corresponding to each component of the cooling load and
the non-weather base load within building can address different energy saving measures for the building. However, previous studies highlighted above only focused on the overall energy use in chiller plants and buildings. No dissection of the energy consumption of a building (e.g. base load and weather dependent load) was studied or discussed. The major functions of using statistical model – thermal performance lines for building energy signature are illustrated in Figure 3.1

**Figure 3.1**  Major Functions of Using Statistical Model – Thermal Performance Line for Building Energy Signature
3.1.2 Theoretical Background on Using Thermal Performance Lines

In general, the chiller plant is used to deal with the building cooling load (i.e. remove heat) which includes sensible and latent load. It is assumed that all air cooling system is related to the energy balance across the cooling coil. Therefore, the calculation of the cooling coil load can be expressed as below:

\[ Q_{\text{coil}} = m_a C_{pa} (\theta_{ao} - (\theta_c - \Delta \theta_L')) \]  (3.1)

where \( m_a \) is the mass flow rate of air [kg/s]; \( C_{pa} \) is the specific heat capacity of air [1.02 kJ/kg/°C]; \( \theta_{ao} \) is the outdoor air dry-bulb temperature [°C]; \( \theta_c \) is the off-coil dry-bulb temperature and \( \Delta \theta_L' \) is the fictitious sensible temperature rise due to latent gains. This sensible temperature rise would give the same size of latent load across the latent load across the coil. The mathematical equation can be written as:

\[ \Delta \theta_L' = 2450 (g_{ao} - g_c) \]  (3.2)

where \( g_{ao} \) and \( g_c \) are the outdoor and off-coil air moisture contents respectively. The factor of 2450 is the latent heat of vapourization of air [kJ/kg]. This approach allows the latent load on the coil that can be treated as a sensible load and hence be incorporated within a cooling degree-day integral.
The cooling base temperature ($\theta_b$) can be defined as:

$$\theta_b = \theta_c - \Delta \theta_L$$  \hspace{1cm} (3.3)

If the differences between hourly outdoor temperature and the base temperature are summed over period of time (i.e. degree-hours), this value can be multiplied by $m_a$ and $C_{pa}$ to give the cooling load from the air treated by the cooling coil in kWh. Then, the equation can be expressed as:

$$E_{cooling} = m_a C_{pa} \Sigma (\theta_{ao} - \theta_b)$$  \hspace{1cm} (3.4)

Since the chiller plant is to produce chilled water to the cooling coil in order to remove the cooling load from the air, the electricity consumption of the chiller [kWh] can be calculated as:

$$E_{chiller} = E_{cooling} / COP$$  \hspace{1cm} (3.5)

where COP is the coefficient of performance of a chiller.
With rearranging Equation (3.4) and (3.5)

\[ \frac{E_{\text{chiller}}}{\sum (\theta_{ao} - \theta_b)} = 24 \frac{m_a C_{pa}}{COP} \]

It is noted that Equation (3.4) divided by 24 yields degree-days.

With the assumptions made by [Day 2005], a plot of chiller electricity consumption against the cooling degree-days can produce a straight line where the slope is equivalent to:

\[ \text{Slope} = 24 \frac{m_a C_{pa}}{COP} \quad (3.6) \]

Same concept can also be applied to the electricity consumption of the chiller plant and the mean outdoor air temperature, the slope of the thermal performance line can be interpreted as:

\[ \frac{E_{\text{chiller}}}{\Delta T_o} = T_{op} m_w C_{pw} \Delta T_{chw} / COP \quad (3.7) \]

\[ \text{Slope} = T_{op} m_w C_{pw} \Delta T_{chw} / COP \quad (3.8) \]
where $\Delta T_o$ is the difference of the mean outdoor air temperature; $T_{op}$ is the annual operating hours of the chiller plant [hr]; $m_w$ is the chilled water flow rate [kg/s]; $C_{pw}$ is the specific heat capacity of water [4.185 kJ/kg/$^\circ$C]; $\Delta T_{chw}$ is the chilled water temperature difference [$^\circ$C]; and $E_{cooling} = T_{op} m_w C_{pw} \Delta T_{chw}$.

From Equation (3.8), it can be easily proved that the deterioration of the efficiency of the chiller plant (i.e. $COP$) at a constant particular cooling energy profile will increase the slope of the thermal performance line or vice versa.
3.2 ADOPTION OF STATISTICAL MODEL APPROACH FOR ENERGY PERFORMANCE ANALYSIS OF AIR-CONDITIONED BUILDINGS IN HONG KONG

3.2.1 Building and Energy Consumption Information

In order to study the changes of the climatic variable influencing the building energy consumption, 20 buildings, including: office buildings, commercial buildings, building complex, community halls, theatres, exhibition centres, institutional building, laboratory building, wholesales markets and farm office were studied. Based on the available building information collected from the facility management of each building, the details of each building are summarized in Table 3.1.
### Table 3.1  Building Information Schedule

<table>
<thead>
<tr>
<th>Building Code</th>
<th>Bldg Type</th>
<th>GFA (m²)</th>
<th>WWR</th>
<th>U-Value of Building Façade</th>
<th>No. of Floors</th>
<th>Chiller Plant Type</th>
<th>AC System Type</th>
<th>Operating Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bldg A</td>
<td>Gov. Office</td>
<td>4,350</td>
<td>0.40</td>
<td>N.A</td>
<td>LG1/F (E&amp;M Plant Rms); G/F (Lobby; Exhibition Hall); 1/F - 7/F (Offices, Computer Rooms).</td>
<td>ACC Screw (2 x 300 kW; 2 x 120 kW).</td>
<td>PAUs + FCUs (Open Plan Offices); Window Type AC Unit (Computer Rooms); CRAC Units (Server Room).</td>
<td>08:00 - 19:00 (Mon - Sat); Closed (Sun or Public Holiday); 24 Hours (Server Room).</td>
</tr>
<tr>
<td>Bldg B</td>
<td>Gov. Office</td>
<td>34,252</td>
<td>0.54</td>
<td>N.A</td>
<td>Block A G/F - 5/F (Offices); Block B G/F - 5/F (Offices).</td>
<td>WCC Centrifugal (2 x 200 TR) with 2 x CT; ACC Centrifugal Oil-Free (1 x 100 TR); ACC Screw (1 x 100 TR).</td>
<td>AHUs VAV or CAV (Open Plan Offices); PAUs + FCUs (Open Plan Offices); FAP + VRVs (Open Plan Offices); CRAC Units (Server Room).</td>
<td>08:00 - 18:00 (Mon - Sun); Closed (Public Holiday); 24 Hours (Server Room and Management Office).</td>
</tr>
<tr>
<td>Bldg C</td>
<td>Commercial</td>
<td>30,880</td>
<td>0.53</td>
<td>N.A</td>
<td>G/F - 2/F (Shops); 3/F - 22/F (Offices)</td>
<td>ACC Screw (5 x 1037 kW).</td>
<td>PAUs + FCUs (Shops and Management Office); PAUs + AHUs (Offices and Server Room)</td>
<td>08:00 - 20:00 (Mon - Sat); 09:00 - 18:00 (Sun and Public Holiday).</td>
</tr>
<tr>
<td>Bldg D</td>
<td>Commercial</td>
<td>22,450</td>
<td>0.50</td>
<td>N.A</td>
<td>G/F (Car Park and Management Office); 1/F (Lift Lobby); 2/F - 3/F (Car Park); 5/F (Lobby); 15/F (E&amp;M Plant Rm); 16/F - 25/F (Offices).</td>
<td>ACC Reciprocating (5 x 700 kW).</td>
<td>PAUs + FCUs (Office); Split-Type AC Units (Management Office).</td>
<td>08:00 - 20:00 (Mon - Sat); Closed (Sun and Public Holiday).</td>
</tr>
</tbody>
</table>
### Building Code, Bldg Type, GFA (m²), WWR, U-Value of Building Façade, No. of Floors, Chiller Plant Type, AC System Type, Operating Hours

<table>
<thead>
<tr>
<th>Building Code</th>
<th>Bldg Type</th>
<th>GFA (m²)</th>
<th>WWR</th>
<th>U-Value of Building Façade</th>
<th>No. of Floors</th>
<th>Chiller Plant Type</th>
<th>AC System Type</th>
<th>Operating Hours</th>
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</thead>
<tbody>
<tr>
<td>Bldg E</td>
<td>Commercial</td>
<td>38,160</td>
<td>0.40</td>
<td>N.A</td>
<td>G/F - 1/F (Car Park and Management Office); 1/F - 2/F (Car Park); 3/F - 4/F (Offices).</td>
<td>ACR (8 x 1,060 kW).</td>
<td>Individual AC Packaged Units (Offices).</td>
<td>08:00 - 20:00 (Mon - Sat); Closed (Sun and Public Holiday).</td>
</tr>
<tr>
<td>Bldg F</td>
<td>Government Complex</td>
<td>49,489</td>
<td>0.65</td>
<td>N.A</td>
<td>G/F and 1/F (Market); 2/F (Cooked Food Centre); 3/F and 4/F (Offices); 5/F (Library); 6/F and 7/F (Sports Centre).</td>
<td>ACC Screw (3 x 1047 kW for Market); ACC Screw (1 x 1047 kW for Office); ACC Screw (3 x 1134 kW for Office and Library); ACC Screw (1 x 318 kW) for Poultry at Night Time.</td>
<td>AHUs (Market on 1/F and 2/F); PAUs + FCUs (Offices on 3/F and 4/F); AHUs and PAUs + FCUs (Library on 5/F); AHUs (Sports Centre on 6/F and 7/F).</td>
<td>06:00 - 20:00 (Mon - Sun) - 1/F and 2/F Market; 06:00 - 02:00 (Mon - Sun) - 2/F Cooked Food Centre; 08:00 - 18:00 (Mon - Fri) - 3/F and 4/F Offices; 10:00 - 19:00 (Mon - Wed) &amp; 10:00 - 17:00 (Sat - Sun) &amp; 10:00 - 13:00 (Public Holiday) - 3/F Library; 07:00 - 23:00 (Mon - Sun) - 6/F and 7/F Sports Centre.</td>
</tr>
<tr>
<td>Bldg G</td>
<td>Complex</td>
<td>18,190</td>
<td>0.60</td>
<td>N.A</td>
<td>G/F and 1/F (Market); 2/F (Cooked Food Centre); 3/F (Library); 4/F (Swimming Pool); 5/F (Offices); 6/F - 9/F (Sports Centre).</td>
<td>ACC Screw (1 x 1225 kW)</td>
<td>AHUs (Market on G/F and 1/F); AHUs (Cooked Food Centre on 2/F); AHUs and PAUs + FCUs (Library on 3/F); AHUs and PAUs + FCUs (Swimming Pool on 4/F); AHUs and PAUs + FCUs (Swimming Pool on 4/F); PAUs + FCUs (offices on 5/F); AHUs &amp; PAUs + FCUs (Sports Centre on 6/F - 9/F).</td>
<td>06:00 - 20:00 (Mon - Sun) - G/F and 1/F Market; 06:00 - 20:00 (Mon - Sun) - 2/F Cooked Food Centre; 10:00 - 19:00 (Mon - Wed) &amp; 10:00 - 17:00 (Sat - Sun) &amp; 10:00 - 13:00 (Public Holiday) - 3/F Library; 06:30 - 22:00 (Tue - Sun) &amp; 18:00 - 22:00 (Mon) - 4/F Swimming Pool; 09:00 - 18:00 (Mon - Fri) - 5/F Offices; 07:00 - 23:00 (Tue - Sun) &amp; 15:00 - 23:00 (Mon) - 6/F to 9/F 6/F Sports Centre.</td>
</tr>
<tr>
<td>Building Code</td>
<td>Bldg Type</td>
<td>GFA (m²)</td>
<td>WWR</td>
<td>U-Value of Building Façade</td>
<td>No. of Floors</td>
<td>Chiller Plant Type</td>
<td>AC System Type</td>
<td>Operating Hours</td>
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<tr>
<td>Bldg H</td>
<td>Complex</td>
<td>16,210</td>
<td>0.50</td>
<td>N.A</td>
<td>G/F and 1/F (Market); 2/F (Centre for Food Safety); 3/F (Cooked Food Centre); 4/F - 5/F (Library); 6/F - 9/F (Offices); 10/F - 13/F (Sports Centre).</td>
<td>ACC Screw (3 x 752 kW)</td>
<td>AHUs (Market on G/F and 1/F);</td>
<td>06:00 - 20:00 (Mon - Sun) - G/F and 1/F Market; AHUs (Centre for Food Safety on 2/F &amp; Cooked Food Centre on 3/F); PAUs + FCUs (Offices on 6/F - 9/F); AHUs &amp; PAUs + FCUs (Sports Centre on 10/F - 13/F)</td>
</tr>
<tr>
<td>Bldg I</td>
<td>Community</td>
<td>1,600</td>
<td>0.45</td>
<td>N.A</td>
<td>1/F (Hall).</td>
<td>ACC Screw (2 x 258 kW).</td>
<td>AHUs (Hall).</td>
<td>07:00 - 22:00 (All).</td>
</tr>
<tr>
<td>Bldg J</td>
<td>Community</td>
<td>1,550</td>
<td>0.40</td>
<td>N.A</td>
<td>1/F (Hall).</td>
<td>AC DX Package Units (2 x 120 kW; 2 x 73.4 kW).</td>
<td>AC DX Package Units.</td>
<td>Depends on Booking Schedule.</td>
</tr>
<tr>
<td>Bldg K</td>
<td>Community</td>
<td>900</td>
<td>0.45</td>
<td>N.A</td>
<td>G/F (Office and Hall); 1/F (Multi-Function Rm).</td>
<td>AC DX Packaged Units (2 x 108 kW; 2 x 200 kW).</td>
<td>DX PAUs + DX AHUs (Hall);</td>
<td>09:00 - 22:00 (Office);</td>
</tr>
<tr>
<td>Bldg L</td>
<td>Community</td>
<td>600</td>
<td>0.45</td>
<td>N.A</td>
<td>G/F (Office and Hall).</td>
<td>ACC Reciprocating (2 x 100 kW).</td>
<td>AHUs (Hall);</td>
<td>09:00 - 22:00 (Office);</td>
</tr>
</tbody>
</table>

CHILLER PERFORMANCE ASSESSMENT BASED ON BUILDING THEMRAL PERFORMANCE LINE CONCEPT
## CHILLER PERFORMANCE ASSESSMENT BASED ON BUILDING THEMRAL PERFORMANCE LINE CONCEPT

<table>
<thead>
<tr>
<th>Building Code</th>
<th>Bldg Type</th>
<th>GFA (m²)</th>
<th>WWR</th>
<th>U-Value of Building Façade</th>
<th>No. of Floors</th>
<th>Chiller Plant Type</th>
<th>AC System Type</th>
<th>Operating Hours</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bldg M</td>
<td>Theatre</td>
<td>4,440</td>
<td>0.50</td>
<td>N.A</td>
<td>Wing A G/F (Backstage Area; Stage Area; and Store Rm); UG/F (Dressing Rms); and 1/F (Dressing Rms; Auditorium; and Office); Wing B G/F (Box Office; Lobby; and Entrance); and 1/F (Restaurant; Rehearsal Rms; Committee Rms; Auditorium; and Offices).</td>
<td>ACC Screw (2 x 530 kW)</td>
<td>PAU + AHU (Whole Building except small office rooms etc.); PAU + FCU (Small Office Rooms; and Back of Houses).</td>
<td>09:00 - 23:30 (All).</td>
</tr>
<tr>
<td>Bldg M</td>
<td>Theatre</td>
<td>5,896</td>
<td>0.55</td>
<td>N.A</td>
<td>2/F (Booking Offices; Box Office; and Auditorium); 3/F (Auditorium; Exhibition Hall; and Classrooms).</td>
<td>ACC Screw (3 x 530 TR)</td>
<td>PAU + AHU (Auditorium); PAU + FCU (Small Office Rooms; and Back of Houses); VRV (Exhibition Halls).</td>
<td>09:00 - 23:00 (All except Booking Offices and Box Offices); 09:00 - 18:00 (Booking Offices and Box Office); Closed (Sat, Sun and Public Holidays) - Booking Offices.</td>
</tr>
<tr>
<td>Bldg O</td>
<td>Theatre</td>
<td>25,774</td>
<td>0.50</td>
<td>N.A</td>
<td>Gi/F (Conference Rm; Lecture Rms; Dance Studio; and Music Studio); P/F (Box Office); U/P/F (Auditorium); M/F (Dressing Rms); 1/F Exhibition Gallery; Reception Lounge; and Office; and 2/F (Activities Hall, Dressing Rms; Lobby; Main Entrance; and Auditorium Balcony).</td>
<td>SWC Centrifugal (2 x 1800 kW).</td>
<td>AHU + FCU (G/F - All Rooms; 1/F Exhibition Gallery; and 2/F - Auditorium; Main Entrance; and Activities Hall); PAU + FCU (2/F and M/F - Dressing Rooms); Split-Type AC Units (Common Areas).</td>
<td>09:00 - 22:00 (all)</td>
</tr>
</tbody>
</table>
| Bldg P        | Exhibition Centre | 12,500   | 0.56| N.A                       | Gi/F (Café, Atrium; Shops; Exhibition Areas; Offices); 1/F (Exhibition Areas; Offices). | DX PAU + DX AHU. | Closed (Tue except Public Holidays); 24 Hours (Offices). | 10:00 - 17:00 (except Tue);
### Building Code | Bldg Type | GFA (m²) | WWR | U-Value of Building Façade | No. of Floors | Chiller Plant Type | AC System Type | Operating Hours |
<table>
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</thead>
<tbody>
<tr>
<td>Bldg Q</td>
<td>Exhibition Centre</td>
<td>13,759</td>
<td>0.50</td>
<td>N.A</td>
<td>G/F (Entrance Lobby); 1/F - 2/F (Art Studio; Music Practice Rm; Lecture Hall; Theatre; Offices and Music Store); 3/F (Offices; Lecture Hall and Theatre); 4/F (Exhibition Hall and Office)</td>
<td>ACC Screw (3 x 935 kW); ACC Reciprocating (2 x 352 kW).</td>
<td>PAUs + FCUs (Offices); AHUs (other than offices areas).</td>
<td>10:00 - 17:00 (except Tue); Closed (Tue except Public Holidays); 24 Hours (Offices).</td>
</tr>
<tr>
<td>Bldg R</td>
<td>Exhibition Centre</td>
<td>57,726</td>
<td>0.50</td>
<td>N.A</td>
<td>UB/F to P/F (Offices; Concert Hall; Theatre); 3/F (Offices); 4/F (Exhibition Gallery; Function Rms); 5/F (Offices); 6/F (Ticketing Office); 7/F - 9/F (Offices)</td>
<td>SWC Centrifugal (4 x 620 TR).</td>
<td>PAUs + FCUs (Office); AHUs (other than office areas).</td>
<td>08:45 - 16:45 (Mon - Fri) &amp; 09:00 - 11:30 (Sat) - UB/F to P/F Offices; 07:30 - 21:30 (Mon - Sun) - 4/F to 9/F Offices; Depends on Programme - UB/F to P/F Concert Hall and Theatre.</td>
</tr>
<tr>
<td>Bldg S</td>
<td>Institutional</td>
<td>25,339</td>
<td>0.60</td>
<td>2.32 W/m²°C (External Wall); 2.85 W/m²°C (Glazing); 0.4 W/m²°C (Roof).</td>
<td>G/F- 7/F (Classrooms; Offices).</td>
<td>WWC Centrifugal (4 x 1513 kW) with 4 x CT; WWC Screw (1 x 704 kW) with 1 x CT.</td>
<td>PAUs + FCUs (Lecture Rooms); PAUs + AHUs (Lobby).</td>
<td>07:00 - 23:00 (Mon - Sat); 07:00 - 17:00 (Sun); Close (Holiday).</td>
</tr>
<tr>
<td>Bldg T</td>
<td>Laboratory</td>
<td>2,160</td>
<td>0.70</td>
<td>N.A</td>
<td>Phase 1 LG/F - UG/F (Offices and Labs); Phase 2 G/F (Labs).</td>
<td>ACC Reciprocating (4 x 52.7 kW) for Phase 1; ACC Reciprocating (2 x 157 kW) for Phase 2.</td>
<td>PAUs + AHUs (Labs); VRV Units (Offices).</td>
<td>08:00 - 18:00 (Mon); 08:00 - 17:45 (Tue - Fri); Closed (Sat - Sun; Holiday); 24 Hours (All Labs).</td>
</tr>
</tbody>
</table>
Note:

ACC denotes Air-Cooled Chiller
ACR denotes Air-Cooled Radiator
AHU denotes Air-Handling Unit
CAV denotes Constant Air Volume
CRAC Unit denotes Computer Room Air Conditioning Unit
CT denotes Cooling Tower
DX denotes Direct Expansion
FCU denotes Fan Coil Unit
Gov denotes Government
N.A denotes Not Available
PAU denotes Primary Air-Handling Unit
SWC denotes Sea Water-Cooled Chiller
VAV denotes Variable Air Volume
VRV Unit denotes Variable Refrigerant Volume Unit
WCC denotes Water-Cooled Chiller
WWR denotes Window to Wall Ratio
3.2.2 Formulation of Thermal Performance Line Model

Before formulating the thermal performance line model for particular building, collection of the electricity consumption data of each building as well as the climatic data over a year are necessary. In this study, the electricity data collection for the building was based on the power utility electricity bills while the collection of the climatic data was based on the records from the Hong Kong Observatory. This reason is that not all the buildings have building management system (BMS) with logging or recording the overall electricity consumption of the building. In addition, the electricity bills for commercial buildings were for public area only due to the constraint to obtain the electricity bills for each tenant. In order to assess how energy efficient of the building is, three past consecutive year records (monthly basis) were collected for comparison.

With regard to the selection of the climatic variables correlating to the building electricity consumption, it was proved that the effect of the global solar radiation on the building electricity consumption is significantly small when compared with outdoor air dry-bulb temperature as well as outdoor air relative humidity [Abdel-Nai et al. 1990; Yik and Sat 2001]. In addition, it is also discovered that the variations of the monthly outdoor relative humidity over the past 10 year are relative constant when compared with the monthly outdoor air dry-bulb temperature and the monthly outdoor air moisture content. Although the outdoor air relative humidity can be directly obtained by measurement, it seems that it cannot truly reflect how the outdoor weather influencing the building electricity consumption. Conversely, the profiles of the monthly outdoor air moisture content over the past 10 years are most likely similar to the monthly outdoor air dry-bulb temperature. Furthermore, the
outdoor air moisture content is calculated from the outdoor air dry-bulb temperature as well as coincident outdoor air relative humidity. Apart from that, moisture content is direct measure of water vapour content of air and this is the basis for determining the latent load of the building [Yik et al. 1995].

Figure 3.2 Monthly Mean Outdoor Air Dry-Bulb Temperature Profile for Hong Kong (from Year 2002 to Year 2011)
CHAPTER 3        CHILLER PERFORMANCE ASSESSMENT BASED ON BUILDING THERMAL PERFORMANCE LINE CONCEPT

Figure 3.3  Monthly Mean Outdoor Air Relative Humidity Profile for Hong Kong (from Year 2002 to Year 2011)

Figure 3.4  Monthly Mean Outdoor Air Moisture Content Profile for Hong Kong (from Year 2002 to Year 2011)
CHAPTER 3  
CHILLER PERFORMANCE ASSESSMENT BASED ON BUILDING  
THEMRAIL PERFORMANCE LINE CONCEPT

Figure 3.5  Mean Outdoor Air Dry-Bulb Temperature Against Mean Outdoor Air Relative Humidity (from Year 2002 to Year 2011)

Figure 3.6  Mean Outdoor Air Dry-Bulb Temperature Against Mean Outdoor Air Moisture Content (from Year 2002 to Year 2011)
With gained the past experience from Yik et al. [1995]; Yik and Sat [2001]; Yu and Chan [2005], using linear regression mathematical model form to represent the influence of the climatic variables on the building energy consumption is better than adopting the multiple linear regression mathematical model form while the accuracy of the model with using linear regression mathematical model form is still high.

Furthermore, one of the arguments on selecting and forming the thermal performance line model is that the building electricity consumption was correlated with $T_oW_o$. Previous study validated that high accuracy of the model could be maintained [Yik and Sat 2001]. From energy audit point of view, it is difficult to identify EMOs for the building (i.e. dissecting the building energy consuming components) since $T_o$ affects sensible load of the building while $W_o$ has effect on the latent load of the building. Due to this reason, it is considered to investigate how each climatic variable influencing the building energy consumption individually. As a result, the thermal performance line model can be written as:

$$E_{Building} = f(T_o, T_o^2, W_o, W_o^2, T_oW_o)$$  \hspace{1cm} (3.9) 

Hence, the mathematical form of the thermal performance lines models can be represented follows:

The temperature dependent model can be interpreted as:

$$E_{Building\ (T_o)} = a_0T_o + a_1$$  \hspace{1cm} (3.10) 

$$E_{Building\ (T_o^2)} = b_0T_o^2 + b_1$$  \hspace{1cm} (3.11) 

where $a_0 - a_1$ and $b_0 - b_1$ are the coefficients for thermal performance line model (temperature dependent).
The moisture content dependent model can be interpreted as:

\[ E_{\text{Building}}(W_o) = c_0 W_o + c_1 \]  \hspace{1cm} (3.12)

\[ E_{\text{Building}}(W_o^2) = d_0 W_o^2 + d_1 \]  \hspace{1cm} (3.13)

where \( c_0 - c_1 \) and \( d_0 - d_1 \) are the coefficients for thermal performance line model (moisture content dependent).

The energy dependent model can be interpreted as:

\[ E_{\text{Building}}(T_oW_o) = e_0 T_o W_o + e_1 \]  \hspace{1cm} (3.14)

where \( e_0 \) and \( e_1 \) are the coefficients for thermal performance line model (energy dependent).

Since different sizes of buildings will have different electricity consumption utilization patterns (i.e. characteristics), using the building electricity consumption with a unit kWh as a base to compare building to building is unfair especially for the different types of buildings (e.g. commercial buildings) [Bannister and Hinge 2006].

Due to this reason, the building electricity consumption data were normalized by the gross floor area of the respective building [i.e. kWh/m\(^2\)] before formulating the thermal performance line model.

It is noted that each building has its own energy performance characteristics and they are unique. In the thermal performance line model, the embedded coefficients of the model reflect the identity (i.e. energy performance characteristics) of each building.
In order to test the goodness of fit of the single independent climatic variable thermal performance line models, the coefficient of determination \( R^2 \) is adopted. The range of this value is between 0 to 1. The higher the \( R^2 \), the high the correlation between the building electricity consumption and selected climatic variable is:

\[
R^2 = 1 - \frac{\sum_{i=1}^{N} (y_i - \hat{y}_i)^2}{\sum_{i=1}^{N} (y_i - \bar{y})^2}
\]  

(3.15)

where \( N \) is the number of samples; \( y_i \) is sample value; \( \hat{y}_i \) is the predicted value; and \( \bar{y} \) is a mean of the sample value.

### 3.2.3 Selection of Deterministic Climatic Variable for Building Energy Performance Analysis

#### 3.2.3.1 Based on Statistical Model Results

As discussed in Section 3.2.1, each of the five selected climatic variables (i.e. \( T_o \), \( T_o^2 \), \( W_o \), \( W_o^2 \) and \( T_o W_o \)) are used to correlate with the building electricity consumption, by using linear regression model technique, to generate thermal performance line models for 20 surveyed buildings.

As the results tabulated in Table 3.2, the \( R^2 \) of \( T_o^2 \) is always higher than \( T_o \). This phenomenon is consistent with the results carried out by Yik and Sat [2001]. Among the surveyed buildings, the average \( R^2 \) value of \( T_o^2 \) is 0.8092 while 0.7900 for \( T_o \). Moving on the \( W_o^2 \) and \( W_o \), this case is very similar to the comparison between \( T_o^2 \) and \( T_o \). The \( R^2 \) value of \( W_o^2 \) is higher than \( W_o \). The average \( R^2 \) value of \( W_o^2 \) among the surveyed buildings is 0.0466 while 0.0452 for \( W_o \). For the product of \( T_o W_o \), the average \( R^2 \) value is nearly the same as \( T_o^2 \) which is 0.8023. The reason of high correlation between \( T_o W_o \) and building electricity consumption is that the
weather-dependent load (i.e. HVAC system or equipment) is used to deal with the building cooling load including sensible and latent load that requires a lot of energy or electricity consumption of the building. As mentioned in previous section, the product of $T_0 W_0$ can reflect the coincident effect on building electricity consumption due to the temperature as well as moisture content dependent loads.
## Table 3.2 Coefficient of Determination ($R^2$) for Individual Selected Independent Climatic Variable Correlated to Building Electricity Consumption

<table>
<thead>
<tr>
<th>Building Type</th>
<th>Mean Monthly Outdoor Air Dry-Bulb Temperature ($T_w$)</th>
<th>Product of Mean Monthly Outdoor Air Dry-Bulb Temperature and Moisture Content ($T_w^2$)</th>
<th>Mean Monthly Outdoor Air Moisture Content ($W_w$)</th>
<th>Product of Mean Monthly Outdoor Air Moisture Content and Moisture Content ($W_w^2$)</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature and Moisture Content ($T_w W_w$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Government Office</td>
<td>$R^2$ = 0.9466, $\text{Slope} = 0.0455$, $Y \text{Intercept} = -0.3773$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Commercial</td>
<td>$R^2 = 0.8830$, $\text{Slope} = 0.0223$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Complex</td>
<td>$R^2 = 0.8320$, $\text{Slope} = 0.0356$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Community Hall</td>
<td>$R^2 = 0.7815$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Community Hall</td>
<td>$R^2 = 0.7456$, $\text{Slope} = 0.0356$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Community Hall</td>
<td>$R^2 = 0.7285$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Theatre</td>
<td>$R^2 = 0.7041$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Theatre</td>
<td>$R^2 = 0.6802$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Theatre</td>
<td>$R^2 = 0.6566$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
<tr>
<td>Theatre</td>
<td>$R^2 = 0.6345$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
<td>$R^2 = 0.0455$, $\text{Slope} = 0.0008$, $Y \text{Intercept} = -0.2333$</td>
</tr>
</tbody>
</table>

Average: $R^2 = 0.7041$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$

Maximum: $R^2 = 0.8830$, $\text{Slope} = 0.0455$, $Y \text{Intercept} = -0.2333$

Minimum: $R^2 = 0.7041$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$

Standard Deviation: $R^2 = 0.7001$, $\text{Slope} = 0.0233$, $Y \text{Intercept} = -0.2333$
3.2.3.2 Based on Thermal Dynamic Model

In determining the most deterministic climatic variable on the building electricity consumption, a fundamental model of “ASHRAE’s CLTD/SCL/CLF Method” for calculating the annual cooling load of a building is adopted [ASHRAE 1993]. With the building cooling load due to transmission heat gain via external facades and glazing, ventilation and infiltration load, internal lighting load and small power load, the design cooling load of a building for a given time \( t \) is expressed as [Lee et al. 2001]:

\[
q(t) = \sum q_{f,i}(t) + \sum q_{g,i}(t) + q_{ven,s}(t) + q_{ven,l}(t) + q_{inf,s}(t) + q_{inf,l}(t) + q_{occ,s}(t) + q_{occ,l}(t) + q_{lgt}(t) + q_{sp}(t)
\]  

(3.16)

where \( q \) is the total design building cooling load [W/m\(^2\)]; \( q_{f,i} \) is the cooling load due to conduction heat gain via the \( i \)th building façade element including wall and glazing [W/m\(^2\)]; \( q_{g,i} \) is the cooling load due to solar gain via glazing [W/m\(^2\)]; \( q_{ven,s} \) and \( q_{ven,l} \) are the ventilation load including sensible and latent load respectively [W/m\(^2\)]; \( q_{inf,s} \) and \( q_{inf,l} \) are the infiltration load including sensible and latent load respectively [W/m\(^2\)]; \( q_{occ,s} \) and \( q_{occ,l} \) are the occupancy load including sensible and latent load respectively [W/m\(^2\)]; \( q_{lgt} \) is the internal lighting load [W/m\(^2\)]; \( q_{sp} \) is the internal small power or equipment load [W/m\(^2\)]; and \( t \) is the condition at a particular time [s].

Then, the design cooling load of a building can be further depicted as below [Lee et al. 2001]:

\[
q(t) = \sum A_{f,i} U_{f,i} (T_o(t) - T_r) + \sum A_{g,i} SC_{g,i} MSHGF_i CLF_i (t) + \rho_a V_o C_{pa} (T_o(t) - T_r) + \rho_a V_o h_{fg,0} (W_o(t) - W_r) + \rho_a V_{inf} C_{pa} (T_o(t) - T_r) + \rho_a V_{inf} h_{fg,0} (W_o(t) - W_r) + q_{occ,s}(t)
\]
CHAPTER 3  
CHILLER PERFORMANCE ASSESSMENT BASED ON BUILDING THERMAL PERFORMANCE LINE CONCEPT

\[ q(t) = q_{occ,s}(t) + q_{occ,l}(t) + q_{lg}(t) + q_{sp}(t) \]  \hspace{1cm} (3.17)

where \( A_{f,i} \) is the area of the building façade element including wall and glazing [m\(^2\)]; \( U_{f,i} \) is the overall heat transfer value of the building façade element including wall and glazing [W/m\(^2\)/°C]; \( T_o \) is the outdoor air dry-bulb temperature [°C]; \( T_r \) is the design indoor air dry-bulb temperature [°C]; \( A_{g,i} \) is the area of the glazing which is the \( i \)th building façade element [m\(^2\)]; \( SC_{g,i} \) is the shading coefficient of the glazing which is \( i \)th building façade element [-]; \( MSHGF_i \) is the maximum solar heat gain factor of the glazing which is the \( i \)th building façade element [W/m\(^2\)]; \( CLF_i \) is the cooling load factor for cooling load due to solar heat gain from a glazing which is the \( i \)th building envelope [-]; \( \rho_a \) is the air density [kg/m\(^3\)]; \( V_o \) is the fresh air flow rate [m\(^3\)/s]; \( C_{pa} \) is the specific heat capacity of air [kJ/kg/°C]; \( h_{fg,0} \) is the latent heat of evaporation of water [kJ/kg]; \( W_o \) is the outdoor air moisture content [kg/kg]; \( W_i \) is the indoor air moisture content [kg/kg]; and \( V_{inf} \) is the infiltration flow rate [m\(^3\)/s].

Equation (3.17) can be expanded as follows:

\[ q(t) = \sum A_{f,i} U_{f,i} T_o(t) - \sum A_{f,i} U_{f,i} T_r + \sum A_{g,i} SC_{g,i} MSHGF_i CLF_i(t) + \rho_a V_o C_{pa} T_o(t) - \rho_a V_o C_{pa} T_r + \rho_a V_o h_{fg,0} W_o(t) - \rho_a V_o h_{fg,0} W_r + \rho_a V_{inf} C_{pa} T_o(t) - \rho_a V_{inf} C_{pa} T_r + \rho_a V_{inf} h_{fg,0} W_o(t) - \rho_a V_{inf} h_{fg,0} W_r + q_{occ,s}(t) + q_{occ,l}(t) + q_{lg}(t) + q_{sp}(t) \]  \hspace{1cm} (3.18)

With considering the influence of climatic variables on the cooling load of the building, non-weather dependent variables can be ignored. Therefore, equation (3.18) can be simplified to become:

\[ q'(t) = \sum A_{f,i} U_{f,i} T_o(t) + \rho_a V_o C_{pa} T_o(t) + \rho_a V_o h_{fg,0} W_o(t) + \rho_a V_{inf} C_{pa} T_o(t) + \rho_a V_{inf} h_{fg,0} W_o(t) \]  \hspace{1cm} (3.19)
In fact, the constant items (i.e. $\rho_a$, $C_{pa}$ and $h_{fg,o}$) in equation (3.19) can also be ignored. As a result, equation (3.19) can be written as:

$$q'(t) = \Sigma A_{f,i} U_{f,i} T_o(t) + V_o' T_o(t) + V_o'' W_o(t) + V_{inf}' T_o(t) + V_{inf}'' W_o(t) \quad (3.20)$$

where $V_o'$ is the sensible load factor due to fresh air flow rate [kW/$\degree$C]; $V_{inf}'$ is the sensible load factor due to infiltration air flow rate [kW/$\degree$C]; $V_o''$ is the latent load factor due to fresh air flow rate [kW]; and $V_{inf}''$ is the latent load factor due to infiltration air flow rate [kW].

By converting the building cooling load to the building electricity consumption of a chiller, equation (3.20) becomes:

$$E_{cc}(t) = (\Sigma A_{f,i} U_{f,i} T_o(t) + V_o' T_o(t) + V_o'' W_o(t) + V_{inf}' T_o(t) + V_{inf}'' W_o(t)) / \text{COP}(t) \quad (3.21)$$

where $E_{cc}$ is the building electricity consumption [kW]; and COP is the coefficient of performance of a chiller.

With considering the Carnot Cycle of a refrigeration compression of a chiller, i.e. $\text{COP} = T_R / (T_o - T_R)$, equation (3.21) can be expressed as:

$$E_{cc}(t) = (\Sigma A_{f,i} U_{f,i} T_o(t) + V_o' T_o(t) + V_o'' W_o(t) + V_{inf}' T_o(t) + V_{inf}'' W_o(t)) \left( (T_o - T_R) / T_R \right) \quad (3.22)$$

Since the COP is a function of $1 / T_o$, the building electricity consumption becomes a function of $T_o^2$ and $T_o W_o$.
Thus, the thermal dynamic model and statistical model (shown in previous section) give the same major climatic variables affecting a building cooling load and hence the building electricity consumption. Through the statistical and thermodynamic model, it can be proved that $T_o^2$ and $T_o W_o$ are the deterministic climatic variables affecting on the building energy consumption.
In addition, by correlating the climatic variables between $T_o^2$ and $T_oW_o$ using the outdoor weather data provided by the Hong Kong Observatory (HKO) with covering from Year 2002 to Year 2011, the correlation between them ($R^2 = 0.9767$) is extremely high enough (see Figure 3.7). This provides a room to consider the priority of selecting either $T_o^2$ or $T_oW_o$ for reflecting the weather-dependent load of a building. For the result with higher accuracy, adopting $T_oW_o$ is the first priority to predict the variable of the weather-dependent load. For convenience purpose, we can select $T_o^2$ instead of using $T_oW_o$.

To conclude, using $T_oW_o$ is the best climatic variable that can be used to reflect how the weather-dependent load affecting the building energy consumption when applying thermal performance line model. With the limitation of the availability of the weather data, using $T_o^2$ is the second choice for reflecting the changes of the weather-dependent load influencing on the building energy consumption while it will not violate the accuracy of the result.
Figure 3.7  Product of Mean Outdoor Air Dry-Bulb Temperature Against Product of Mean Outdoor Air Dry-Bulb Temperature and Outdoor Air Moisture Content (from Year 2002 to Year 2011)
3.2.4 Energy Performance Signatures with Different Types of Buildings for Surveyed Buildings

In general, thermal performance line model or energy signature can reflect the actual performance of the buildings regarding the electricity consumption. It is realized that the electricity consumption characteristics of different types of buildings (e.g. office, commercial or residential) are different. For the same type of buildings, the electricity consumption patterns may be similar. In this section, with using the coefficient term of the thermal performance line model (i.e. for each unit change of climatic variable, how much electricity consumption of the building is influenced) and the base load (i.e. non-weather dependent load), identification of electricity consumption characteristics of different types of buildings are studied.
In this study, 20 surveyed buildings are categorized into eight building types/groups (including office building, commercial building, building complex, community hall, theatre, exhibition centre, institutional building and laboratory building). As shown in Table 3.2, it is clearly showed that the slopes of the thermal performance line model of $T_o^2$ and $T_o W_o$ for the buildings under same group are nearly the same and are distinct to other groups of buildings.

For commercial building, the slopes of the thermal performance line model of both climatic variables for Building C are nearly the same as Building D (i.e. from 1.3 to 1.4 for $T_o^2$ and from 1.6 to 1.8 for $T_o W_o$). The reason is that these two buildings have similar building configuration and building services system as well as operating characteristic (see Table 3.1). Nevertheless, it was discovered that the slope of the thermal line model of both variables for Building E is totally different from others. It is because no centralized chiller plant is adopted in Building E while air-cooled radiators are installed to serve the tenant areas of the building. On the other hand, the $WWR$ of Building E is 0.40 which is the lowest among Building C and D. Due to this reason, the slope of the thermal line model for Building E is very low (i.e. 0.4 for $T_o^2$ and 0.5 for $T_o W_o$) when compared with Building C and D. Thus, Building E is not most sensitivity to the ambient condition comparing to Building C and D.

For building complex, the slopes of the thermal performance line model of both climatic variables for Building F, G and H are nearly the same (i.e. ranging from 2.8 to 2.9 for $T_o^2$ and from 3.4 to 3.7 for $T_o W_o$). This phenomenon may be due to their building operating schedule, functional use and building configurations are nearly the same.
For community hall, the slopes of the thermal performance line model of both climatic variables for Building I, J, K and L (i.e. ranging from 3.3 to 4.6 for $T_o^2$ and from 4.2 to 6.0 for $T_oW_o$) are very similar. It can be explained that their building configurations and operating schedules are more or less the same. Therefore, the effects of the changes of the outdoor thermal environment on the building energy consumption for Building I, J, K and L are the same.

Similar phenomenon was also observed in theatre group, the slopes of the thermal performance line model of $T_o^2$ and $T_oW_o$ for Building M, N and O are fall within 2.1 to 2.7 and 2.6 to 3.7 and respectively. For institutional building, the slopes of the thermal performance lines of $T_o^2$ and $T_oW_o$ are 2.7 and 3.6 respectively. This result is quite similar to that of theatre.

For exhibition centre, the slopes of the thermal performance line model of $T_o^2$ and $T_oW_o$ for Building P and Q are similar (i.e. 1.7 & 1.8 for $T_o^2$ and 2.2& 2.3 for $T_oW_o$ respectively). This phenomenon can be proved that their WWRs (i.e. 0.56 for Building P and 0.50 for Building Q) are quite similar. However, since the WWR of Building R is the lowest when compared with others under the same building type, it was also discovered that the slopes of the thermal performance line model of $T_o^2$ and $T_oW_o$ for Building R are quite low (i.e. 0.9 for $T_o^2$ and 1.1 for $T_oW_o$).

For office building, the slopes of the thermal performance line model of $T_o^2$ and $T_oW_o$ for Building A is significant larger than Building B. Since the WWR of Building B is 0.54 which is higher than that of Building A (i.e. 0.40), a large difference of the slope of the thermal performance line model between these two
buildings will be the result. The result also revealed that Building B is not sensitivity to the changes of outdoor thermal environment since the slope of the thermal performance line model is too low (i.e. 0.5 for $T_o^2$ and 0.7 for $T_o W_o$). It is suggested to select more office buildings to validate the result for office building group.

Building T is a laboratory building which has very high $WWR$. This building design can be reflected by an extremely high slope of the thermal line model of both variables for Building T (i.e. 11.7 for $T_o^2$ and 15.3 for $T_o W_o$) and the slopes of the model for both climatic variables are distinct from other types of buildings. It incurs the electricity consumption of Building T can be affected by the outdoor air condition significantly. Remedial action on adding shading devices or UV coating on the glazing is suggested.

In Hong Kong, the air-conditioning system serving for the building is normally a year round operation especially for the building design with perimeter and interior zone (i.e. 3 – 4m away from the perimeter zone). Since the interior zone of the building may have a lot of electrical equipment, a lot of internal heat will be generated even during winter season. Therefore, air-conditioning supply to remove the heat generated in interior zone of the building during winter time is also required. For energy saving issue, some of the buildings in Hong Kong also adopt free-cooling operation when the outdoor air dry-bulb temperature is equal or below 15.0°C. This figure depends on the judgment of the building operators and therefore, the set point of the free-cooling will be different from building to building. On the other hand, it is hard to determine the base (balance) temperature since the minimum outdoor air dry-bulb temperature provided by HKOH is of 15.0°C while no further record or
information available when the outdoor air dry-bulb temperature is below 15.0°C. As a result, it is very difficult to identify the base load (i.e. non-weather dependent) from the thermal performance line model for the Hong Kong’s situation. According to EMSD [2012] and EMSD [2007] as well as ASHRAE [2009], the building design criteria of lighting power density, small (equipment) power density, occupancy density and occupancy load (sensible and latent load – depending on their activity) are available for us to approximate the base load of building. Unfortunately, the estimation of the base load of the existing building may be incorrect (i.e. over-estimated or under-estimated). One of the reasons is that energy saving programme have been implemented in 20 surveyed buildings (e.g. switching conventional compact fluorescent tube from T8 to T5 or LED, de-lamping or using of energy efficient electrical equipment etc.) and retrofitting work in order to maximize the utilization of the working place (in turns changing of the occupancy density) from time to time.
3.2.5 Energy Performance Signatures for Building Energy Audit

In this section, five representable surveyed buildings are selected for highlighting the usefulness of the thermal performance line model for building energy performance assessment. These five surveyed buildings generally cover common situations encountered in most of the air-conditioned buildings in Hong Kong. Through the building energy performance assessment with using the thermal performance line model, the building energy consumption over the past years can be identified. For convenience purpose without affecting the accuracy of the prediction of the building energy performance assessment, a climatic variable \( T_o \) is adopted in this study.

Regarding the assessment of the building energy performance with the use of thermal performance line model, four typical cases together with graphical explanations are discussed as below:
For Case 1, the entire thermal performance line for current case/year is shifted downward comparing to base case/previous year. This phenomenon may be due to the energy reduction of base load (i.e. non-weather dependent) occurred in the current case or year. The base load may include lighting system, electrical installations or electrical appliances etc.
For Case 2, the entire thermal performance line for current case/year is shifted upward comparing to base case/previous year. This phenomenon may be due to the increase in energy of base load (i.e. non-weather dependent) occurred in the current case or year. Therefore, investigation on the base load electrical equipment is required to validate whether the electrical equipment was working inefficient or additional electrical equipment was installed in the current case/year.

For Case 3, the slope of the entire thermal performance line for current case/year is increased comparing to base case/previous year. This phenomenon may be due to the increase in energy of weather dependent load (i.e. HVAC system) occurred in the current case or year. Therefore, investigation on the HVAC system (including chiller plant or air-side system) are required to validate whether the HVAC system was working inefficient or additional HVAC equipment was installed in the current case/year.

For Case 4, the slope of the entire thermal performance line for current case/year is decreased comparing to base case/previous year. This phenomenon may be due to the reduction in energy of HVAC system or the enhancement of the energy efficiency of the HVAC system or energy saving measured for the HVAC system occurred in the current case or year.
Building A

Figure 3.9 (b) illustrated that the slopes of the thermal performance lines over the past three consecutive years (from Year 2009 to Year 2012) were similar. Moreover, it was also observed that the slopes of the thermal performance lines models for temperature dependent for the second year was lower comparing to the first year (see Figure 3.9 (a)). It may be due to the energy saving measure on HVAC system was carried out in the second year. While it is also observed that the slope of the thermal performance line mode for the third year was higher comparing to the second year. It implies that the energy efficiency of the HVAC system was reduced in the third year. Therefore, it is recommended to inspect the entire HVAC system in order to improve the energy performance of the HVAC system.

Besides, it was also observed that energy reduction could be achieved over the past three years since the thermal performance line model for the first, second and third year were shifted downward gradually. It proves that energy saving measures on base load (i.e. non-weather dependent load) was implemented every year. Indeed, the lighting system upgrading retrofit project (from T8 compact fluorescent tube to T5 compact fluorescent tube) was carried out over the past three years.

With using the thermal performance line model (temperature dependent) for Building A over the past three years (see Figure 3.9 (a)), the percentage changes of building energy consumption for increasing 0.5°C against the baseline outdoor air dry-bulb temperature (i.e. the average mean monthly outdoor air temperature over the past three years – 23.3°C) over the past three year are summarized in Table 3.3 and Table 3.5. Since $T_o^2$ can directly reflect how the outdoor thermal environment
influences the building energy consumption, the thermal performance line model with $T_o^2$ is selected in this study.

![Figure 3.9 (a) Product of Thermal Performance Line Model for Temperature Dependent](image)

![Figure 3.9 (b) Thermal Performance Line Model for Energy Dependent](image)

**Figure 3.9 (a) – (b) Thermal Performance Line Models for Past Three Consecutive Years (from Year 2009 to Year 2012) for Building A**

**Table 3.3 Prediction of Percentage Changes of Building Electricity Consumption for Past Three Years (Building A)**

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_o^2$</th>
<th>Year 2009 to Year 2010</th>
<th>Year 2010 to Year 2011</th>
<th>Year 2011 to Year 2012</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>kWh per m$^2$</td>
<td>% Change</td>
<td>kWh per m$^2$</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>70.2</td>
<td>-</td>
<td>66.5</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>71.1</td>
<td>+1.277</td>
<td>67.4</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>72.1</td>
<td>+2.549</td>
<td>68.3</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>73.0</td>
<td>+3.815</td>
<td>69.2</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>74.0</td>
<td>+5.073</td>
<td>70.2</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>74.9</td>
<td>+6.324</td>
<td>71.1</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>75.9</td>
<td>+7.567</td>
<td>72.1</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>76.9</td>
<td>+8.800</td>
<td>73.1</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>78.0</td>
<td>+10.024</td>
<td>74.1</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>79.0</td>
<td>+11.238</td>
<td>75.1</td>
</tr>
</tbody>
</table>
### Table 3.4  Prediction of Percentage Changes of Building Electricity Consumption over Past Three Years (Building A)

<table>
<thead>
<tr>
<th>$T_s$</th>
<th>$T_e$</th>
<th>Year 2010 to Year 2011</th>
<th>Year 2011 to Year 2012</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Baseline Year</td>
<td>Previous Year</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>-5.269%</td>
<td>-5.269%</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>-5.229%</td>
<td>-5.229%</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>-5.189%</td>
<td>-5.189%</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>-5.150%</td>
<td>-5.150%</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>-5.110%</td>
<td>-5.110%</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>-5.071%</td>
<td>-5.071%</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>-5.032%</td>
<td>-5.032%</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>-4.994%</td>
<td>-4.994%</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>-4.955%</td>
<td>-4.955%</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>-4.918%</td>
<td>-4.918%</td>
</tr>
</tbody>
</table>
Building B

Figure 3.10 (b) illustrated that the slopes of the thermal performance line for the second year and the third year were progressively increased when comparing with the first year. It may be due to the additional ventilation load (including sensible and latent load) of Building B was increased as the retrofitting work for the changing the building layout for increasing the occupancy density of some areas was carried out in Year 2011. The action obviously affected the ventilation requirement of the building.

It was also observed that there was a big shift upward of the thermal performance line for the second year when comparing to the first year. It may be due to an additional air-cooled centrifugal oil-free chiller with 100 TR of cooling capacity was installed for the building in June 2011. Hence, the building electricity consumption of the building was no doubt increased in second year.

Apart from that, there was a shift downward of the thermal performance line for the third year comparing to that of the second year. It implies that energy saving on base load was implemented in the third year. It could be proved that the energy efficient lighting system retrofitting work (switching from T8 compact fluorescent tube to T5 compact fluorescent tube) and the de-lamping programme as well as the installation of automatic timer control for electrical small power equipment were conducted in the third year.

Same phenomena were also explored when temperature dependent thermal performance line model ($T_o^2$) was adopted (see Figure 3.10 (a)). Nevertheless, the effects of the retrofitting work (additional air-cooled oil-free chiller) on the building
electricity consumption for the weather-dependent load could not be reflected clearly since the slopes of the thermal performance line for second and third year were the same.

With using the thermal performance line model (temperature dependent) for Building B over the past three years (see Figure 3.10 (a)), the percentage changes of building energy consumption for increasing $0.5^\circ$C against the baseline outdoor air dry-bulb temperature (i.e. the average mean monthly outdoor air temperature over the past three years $-23.3^\circ$C) over the past three year are summarized in Table 3.5 and Table 3.6. Since $T_o^2$ can directly reflect how the outdoor thermal environment influences the building energy consumption, the thermal performance line model with $T_o^2$ is selected in this study.

![Figure 3.10 (a) Thermal Performance Line Model for Temperature Dependent](image)

![Figure 3.10 (b) Thermal Performance Line Model for Energy Dependent](image)

**Figure 3.10 (a) – (b) Thermal Performance Line Models for Past Three Consecutive Years (from Year 2009 to Year 2012) for Building B**
### Table 3.5 Prediction of Percentage Changes of Building Electricity Consumption for Past Three Years (Building B)

<table>
<thead>
<tr>
<th>$T_a$</th>
<th>$T_i$</th>
<th>Year 2009 to 2010 kWh per m²</th>
<th>% Change</th>
<th>Year 2010 to 2011 kWh per m²</th>
<th>% Change</th>
<th>Year 2011 to 2012 kWh per m²</th>
<th>% Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>6.0</td>
<td>-</td>
<td>6.5</td>
<td>-</td>
<td>6.2</td>
<td>-</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>6.2</td>
<td>+1.908%</td>
<td>6.6</td>
<td>+1.932%</td>
<td>6.3</td>
<td>+2.012%</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>6.3</td>
<td>+3.785%</td>
<td>6.7</td>
<td>+3.831%</td>
<td>6.5</td>
<td>+3.986%</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>6.4</td>
<td>+5.629%</td>
<td>6.8</td>
<td>+5.696%</td>
<td>6.6</td>
<td>+5.922%</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>6.5</td>
<td>+7.441%</td>
<td>7.0</td>
<td>+7.528%</td>
<td>6.7</td>
<td>+7.821%</td>
</tr>
<tr>
<td>25.8</td>
<td>666.4</td>
<td>6.6</td>
<td>+9.220%</td>
<td>7.1</td>
<td>+9.326%</td>
<td>6.9</td>
<td>+9.681%</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>6.8</td>
<td>+10.966%</td>
<td>7.2</td>
<td>+11.090%</td>
<td>7.0</td>
<td>+11.504%</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>6.9</td>
<td>+12.680%</td>
<td>7.4</td>
<td>+12.819%</td>
<td>7.1</td>
<td>+13.289%</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>7.0</td>
<td>+14.360%</td>
<td>7.5</td>
<td>+14.515%</td>
<td>7.3</td>
<td>+15.037%</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>7.2</td>
<td>+16.007%</td>
<td>7.7</td>
<td>+16.177%</td>
<td>7.4</td>
<td>+16.747%</td>
</tr>
</tbody>
</table>

### Table 3.6  Prediction of Percentage Changes of Building Electricity Consumption over Past Three Years (Building B)

<table>
<thead>
<tr>
<th>$T_a$</th>
<th>$T_i$</th>
<th>Year 2010 to 2011 Baseline Year % Change</th>
<th>Previous Year % Change</th>
<th>Year 2011 to 2012 Baseline Year % Change</th>
<th>Previous Year % Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>+6.844%</td>
<td>+6.844%</td>
<td>+2.605%</td>
<td>-3.967%</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>+6.869%</td>
<td>+6.869%</td>
<td>+2.710%</td>
<td>-3.892%</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>+6.893%</td>
<td>+6.893%</td>
<td>+2.812%</td>
<td>-3.818%</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>+6.917%</td>
<td>+6.917%</td>
<td>+2.912%</td>
<td>-3.745%</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>+6.940%</td>
<td>+6.940%</td>
<td>+3.011%</td>
<td>-3.674%</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>+6.963%</td>
<td>+6.963%</td>
<td>+3.108%</td>
<td>-3.604%</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>+6.986%</td>
<td>+6.986%</td>
<td>+3.203%</td>
<td>-3.535%</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>+7.008%</td>
<td>+7.008%</td>
<td>+3.297%</td>
<td>-3.468%</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>+7.030%</td>
<td>+7.030%</td>
<td>+3.389%</td>
<td>-3.402%</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>+7.051%</td>
<td>+7.051%</td>
<td>+3.478%</td>
<td>-3.337%</td>
</tr>
</tbody>
</table>
**Building D**

As shown in Figure 3.11 (a) and (b), the slope of the thermal performance line for the second year was decreased when compared with the first year while it was an increase for the third year when compared with the second year. Therefore, investigation on the energy efficiency of the HVAC system for Building D is suggested.

Besides, the thermal performance line for the second year was shifted upward compared with the thermal performance line for first year while the thermal performance line model for third year was moved downward when comparing with the models for second year. With the changes of the slope of the thermal performance line for the past three years, it is the reason why intersection point among these three lines is occurred. Regarding the base load over the past three years for Building D, it could be explained that the base load of the building was increased in the second year. Moreover, some energy saving measures were implemented (e.g. replacement of T8 fluorescent tubes to T5 fluorescent tubes, switching-off or de-lamp some lamps etc.) in third year.

Similarly, with using the thermal performance line model (temperature dependent) for Building C over the past three years (see Figure 3.10 (a)), the percentage changes of building energy consumption for increasing 0.5°C against the baseline outdoor air dry-bulb temperature (i.e. the average mean monthly outdoor air temperature over the past three years – 23.3°C) over the past three year are summarized in Table 3.7 and Table 3.8. Since \( T_o^2 \) can directly reflect how the outdoor thermal environment influences the building energy consumption, the thermal performance line model
with \( T_o^2 \) is selected in this study.

From Table 3.8, it was clearly indicated that the building electricity consumption in the second and third year were smaller than that of in the first year. Based on the Figure 3.11 (a) and (b), it was proved that the base load in the second year was increased when compared with the first year. As a result, with using Figure 3.11 (a) and (b) as well as Table 3.8, it could be concluded that the weather-dependent load due to the HVAC system influences a lot of portion of building electricity consumption. Therefore, energy saving measures on HVAC system can save a lot of energy use in building.

Figure 3.11 (a) Thermal Performance Line Model for Temperature Dependent

Figure 3.11 (b) Thermal Performance Line Model for Energy Dependent

Figure 3.11 (a) – (b) Thermal Performance Line Models for Past Three Consecutive Years (from Year 2009 to Year 2012) for Building D
Table 3.7  Prediction of Percentage Changes of Building Electricity Consumption for Past Three Years (Building D)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_i$</th>
<th>Year 2009 to 2010</th>
<th>Year 2010 to 2011</th>
<th>Year 2011 to 2012</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>kWh per m²</td>
<td>% Change</td>
<td>kWh per m²</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>7.2</td>
<td>-</td>
<td>7.1</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>7.4</td>
<td>+3.877%</td>
<td>7.4</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>7.7</td>
<td>+7.544%</td>
<td>7.6</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>8.0</td>
<td>+11.015%</td>
<td>7.9</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>8.3</td>
<td>+14.303%</td>
<td>8.2</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>8.6</td>
<td>+17.421%</td>
<td>8.4</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>8.9</td>
<td>+20.379%</td>
<td>8.7</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>9.2</td>
<td>+23.188%</td>
<td>9.0</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>9.6</td>
<td>+25.857%</td>
<td>9.3</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>9.9</td>
<td>+28.395%</td>
<td>9.6</td>
</tr>
</tbody>
</table>

Table 3.8  Prediction of Percentage Changes of Building Electricity Consumption over Past Three Years (Building D)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_i$</th>
<th>Year 2010 to 2011</th>
<th>Year 2011 to 2012</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Baseline Year</td>
<td>Previous Year</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>-0.629%</td>
<td>-0.629%</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>-0.922%</td>
<td>-0.922%</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>-1.199%</td>
<td>-1.199%</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>-1.462%</td>
<td>-1.462%</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>-1.711%</td>
<td>-1.711%</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>-1.948%</td>
<td>-1.948%</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>-2.173%</td>
<td>-2.173%</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>-2.387%</td>
<td>-2.387%</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>-2.590%</td>
<td>-2.590%</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>-2.784%</td>
<td>-2.784%</td>
</tr>
</tbody>
</table>
Building F

Refer to Figure 12 (a) and (b), the slope of the thermal performance line for the second year was smaller comparing to the first year while an increase was observed in the third year when compared with the first year. The reason is that a water mist system was installed in 2011 in order to maximum the system COP of the chiller plant by evaporative cooling on the condensers of the air-cooled chiller. In the meantime, Building F is a building complex that is operated by EMSD. An energy saving action by increasing the indoor air temperature set point temperature to 25.5°C has been taken since 2011. Therefore, the slope of the thermal performance line for second year was smaller than that of the first year.

Unfortunately, a lot of complaints regarding the poor indoor environment were reported in the second year (i.e. feeling hot and stuffy). Thus, the remedial action was taken by further reducing the supply air temperature of the HVAC system in order to maintain the acceptable indoor air temperature and relative humidity. As a result, the slope of the thermal performance line for the third year was still higher than that of first and second year. Thermal comfort measurement has also been conducted during the spring season and the results showed that the measured indoor air temperatures of all areas within Building F were around 22.5°C to 23.8°C while the measured indoor air relative humidity were around 54.3°C to 61.8°C.

In addition, it was also discovered that the electricity consumption of the base load (non-weather dependent load) of Building F was decreased continuously over the past three years since the thermal performance line for the second and third year were shifted downward progressively. This may be due to the energy saving
measures were implemented (e.g. de-lamping for some lamps in corridor areas, installing motion sensor in corridor in order to control the on/off operation of the lighting system and switching T5 compact fluorescent lamp to LED Lamp for exit sign etc.).

Similarly, the percentage changes of building energy consumption for increasing 0.5°C against the baseline outdoor air dry-bulb temperature (i.e. the average mean monthly outdoor air temperature over the past three years – 23.3°C) over the past three year, based on the thermal performance line models for three years, are summarized in Table 3.9 and Table 3.10. Since $T_o^2$ can directly reflect how the outdoor thermal environment influences the building energy consumption, the thermal performance line model with $T_o^2$ is selected in this study.

![Figure 3.12 (a) - (b) Thermal Performance Line Models for Past Three Consecutive Years (from Year 2009 to Year 2012) for Building F](image-url)
### Table 3.9 Prediction of Percentage Changes of Building Electricity Consumption for Past Three Years (Building F)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_i$</th>
<th>Year 2010 to Year 2011</th>
<th>Year 2011 to Year 2012</th>
<th>Year 2012 to Year 2013</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>kWh per m$^2$</td>
<td>% Change</td>
<td>kWh per m$^2$</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>22.3</td>
<td>-</td>
<td>20.8</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>22.9</td>
<td>+2.797%</td>
<td>21.4</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>23.6</td>
<td>+5.499%</td>
<td>22.0</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>24.2</td>
<td>+8.110%</td>
<td>22.7</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>24.9</td>
<td>+10.633%</td>
<td>23.3</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>25.6</td>
<td>+13.070%</td>
<td>24.0</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>26.3</td>
<td>+15.424%</td>
<td>24.7</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>27.0</td>
<td>+17.698%</td>
<td>25.3</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>27.7</td>
<td>+19.895%</td>
<td>26.0</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>28.4</td>
<td>+22.017%</td>
<td>26.8</td>
</tr>
</tbody>
</table>

### Table 3.10 Prediction of Percentage Changes of Building Electricity Consumption over Past Three Years (Building F)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_i$</th>
<th>Year 2011 to Year 2012</th>
<th>Year 2012 to Year 2013</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Baseline Year Previous Year Baseline Year Previous Year</td>
<td></td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>-6.842% -6.842%</td>
<td>-1.232% +6.023%</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>-6.708% -6.708%</td>
<td>-0.983% +6.137%</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>-6.577% -6.577%</td>
<td>-0.742% +6.246%</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>-6.451% -6.451%</td>
<td>-0.509% +6.352%</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>-6.330% -6.330%</td>
<td>-0.284% +6.454%</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>-6.212% -6.212%</td>
<td>-0.067% +6.552%</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>-6.098% -6.098%</td>
<td>+0.143% +6.647%</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>-5.988% -5.988%</td>
<td>+0.346% +6.738%</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>-5.882% -5.882%</td>
<td>+0.543% +6.826%</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>-5.779% -5.779%</td>
<td>+0.732% +6.911%</td>
</tr>
</tbody>
</table>
**Building I**

Figure 3.13 (a) to (b) present the thermal performance lines models for temperature, and energy dependent over the past three consecutive years for Building I. It is discovered that the $R^2$ was very low. It could be depicted that the influence of the outdoor thermal environment on the building electricity consumption was not significant. This phenomenon may be due to the operation of the HVAC system serving for Building I. Indeed, Building I is a community hall and it consists of activity area and stage area. Two DX packaged AC units are used to serve the activity area while one DX packaged AC unit is to serve the hall. The operation of these AC units depends on the numbers of occupants. If the stage area is not occupied, the AC unit will be switched-off. Therefore, the number of occupants inside the hall dictates the outdoor thermal environment for the on/off operation of these three DX packaged AC units.

Nevertheless, it could still easily observe that the high building electricity consumption was occurred at the high side of the outdoor air dry-bulb temperature and the product of the outdoor air dry-bulb temperature and outdoor air moisture content. As discussed in Section 3.2.3, $COP$ of the AC unit or chiller is mainly affected by the heat rejection load while the heat rejection load is by the heat rejection medium (i.e. condensing water, sea water or ambient air). Therefore, the higher the outdoor dry-bulb temperature for air-cooled packaged AC unit, the lower of the $COP$ would be since the heat from the refrigerant cannot be discharged freely from the condenser to the ambient air.

Apart from investigating on the weather-dependent load, it was also explored that the
base load (including lighting and electrical equipment etc.) of Building K was increased continuously over the past three years. One of the reasons is that the lighting system and the wall-mounted fans serving for entire community hall were always switched-on during occupied and unoccupied period. Therefore, it is recommended to switch-off all the electrical equipment or appliances when the community hall is not occupied. Apart from that, a programme for replacing the conventional T8 compact fluorescent tubes by T5 energy efficient compact fluorescent tube is also suggested.

Despite the operation of the HVAC system serving for Building I is not weather-dependent, the energy efficiency (i.e. $COP$) of the HVAC system is still influenced by the outdoor thermal environment. Therefore, it is still valid to use the thermal performance line model (temperature dependent) for Building I over the past three years (see Figure 3.13 (a)), the percentage changes of building energy consumption for increasing 0.5°C against the baseline outdoor air dry-bulb temperature (i.e. the average mean monthly outdoor air temperature over the past three years – 23.3°C) over the past three year are summarized in Table 3.11 and Table 3.12. Since $T_o^2$ can directly reflect how the outdoor thermal environment influences the building energy consumption, the thermal performance line model with $T_o^2$ is selected in this study.
CHAPTER 3
CHILLER PERFORMANCE ASSESSMENT BASED ON BUILDING THERMAL PERFORMANCE LINE CONCEPT

Figure 3.13 (a) Thermal Performance Line Model for Temperature Dependent

Figure 3.13 (b) Thermal Performance Line Model for Energy Dependent

Figure 3.13 (a) – (b) Thermal Performance Line Models for Past Three Consecutive Years (from Year 2009 to Year 2012) for Building I

Table 3.11 Prediction of Percentage Changes of Building Electricity Consumption for Past Three Years (Building I)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_i$</th>
<th>Year 2009 to Year 2010</th>
<th>Year 2010 to Year 2011</th>
<th>Year 2011 to Year 2012</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>kWh per m²</td>
<td>% Change</td>
<td>kWh per m²</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>29.3</td>
<td>-</td>
<td>30.0</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>30.8</td>
<td>+5.119%</td>
<td>31.4</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>32.3</td>
<td>+9.843%</td>
<td>32.7</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>33.9</td>
<td>+14.213%</td>
<td>34.2</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>35.5</td>
<td>+18.264%</td>
<td>35.6</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>37.1</td>
<td>+22.028%</td>
<td>37.1</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>38.8</td>
<td>+25.532%</td>
<td>38.6</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>40.5</td>
<td>+28.800%</td>
<td>40.2</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>42.2</td>
<td>+31.854%</td>
<td>41.7</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>44.0</td>
<td>+34.712%</td>
<td>43.3</td>
</tr>
</tbody>
</table>
### Table 3.12  Prediction of Percentage Changes of Building Electricity Consumption over Past Three Years (Building I)

<table>
<thead>
<tr>
<th>$T_s$</th>
<th>$T_s^1$</th>
<th>Year 2010 to Year 2011</th>
<th>Year 2011 to Year 2012</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Baseline Year</td>
<td>Previous Year</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>+2.330%</td>
<td>+2.330%</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>+1.781%</td>
<td>+1.781%</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>+1.273%</td>
<td>+1.273%</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>+0.801%</td>
<td>+0.801%</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>+0.363%</td>
<td>+0.363%</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>-0.045%</td>
<td>-0.045%</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>-0.426%</td>
<td>-0.426%</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>-0.782%</td>
<td>-0.782%</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>-1.116%</td>
<td>-1.116%</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>-1.428%</td>
<td>-1.428%</td>
</tr>
</tbody>
</table>
3.3 SURVEY ON AVAILABILITY OF ESSENTIAL BUILDING INFORMATION FOR BUILDING ELECTRICITY CONSUMPTION ANALYSIS

When dealing with the analysis of the building electricity consumption (including weather-dependent and non-weather dependent load) in terms of chiller plant optimization, the collection of the essential building information is vitally important. Unfortunately, it was discovered that the attitude different building owners or property managements, regarding the collection of essential building information, will be different. As a result, it makes the task of the chiller plant optimization in real buildings becoming difficult. Therefore, it is worthwhile to conduct a survey on the availability of essential information for existing chiller plant optimization study in Hong Kong with different types of building owners (including Hong Kong government, institutional facility management office and private property management company).

There are 16 buildings owned by the government buildings, 3 buildings owned by private company and 1 institutional building selected for the study. For essential building information, they are classified into five categories: (a) general building information data, (b) power consumption data, (c) occupancy data, (d) chiller plant operation data and (e) operating hours data. Table 3.13 summarizes the results of the survey on essential information of the building for existing chiller plant optimization with different types of building owners.
### Table 3.13  Survey on Building Energy Performance Information for Energy Audit

<table>
<thead>
<tr>
<th>General Building Information Data</th>
<th>Power Consumption Data</th>
<th>Occupancy Data</th>
<th>Chiller Plant Operating Data</th>
<th>Operating Hours Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Building Number</td>
<td>Building Owner Type</td>
<td>Gross Floor Area (m²)</td>
<td>Monthly Chiller Plant Power Consumption (kWh)</td>
<td>Lighting Power Consumption (kW)</td>
</tr>
<tr>
<td>-----------------</td>
<td>---------------------</td>
<td>---------------------</td>
<td>--------------------------</td>
<td>---------------------</td>
</tr>
<tr>
<td>Bldg A</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg C</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg D</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg E</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg F</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg G</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg H</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg I</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg J</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg K</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg L</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
<tr>
<td>Bldg M</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Bldg N</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Bldg O</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Bldg P</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Bldg Q</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Bldg R</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Bldg T</td>
<td>Government Owned</td>
<td>✓</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>Total Building Numbers</td>
<td>16</td>
<td>16</td>
<td>16</td>
<td>16</td>
</tr>
<tr>
<td>Information Provided</td>
<td>16</td>
<td>0</td>
<td>7</td>
<td>0</td>
</tr>
<tr>
<td>% of Information Provided</td>
<td>100.0%</td>
<td>0.0%</td>
<td>43.8%</td>
<td>0.0%</td>
</tr>
</tbody>
</table>
### General Building Information Data

<table>
<thead>
<tr>
<th>Building Number</th>
<th>Building Owner Type</th>
<th>Gross Floor Area (m²)</th>
<th>Monthly Chiller Plant Power Consumption (kW)</th>
<th>Lighting Power Consumption (kW)</th>
<th>Small Power Consumption (kW)</th>
<th>Occupant Density (occ per m²)</th>
<th>People Sensible Load (W per occ)</th>
<th>People Latent Load (W per occ)</th>
<th>System Average COP</th>
<th>Chiller Plant Operating Data</th>
<th>Lighting Annual Operating Hours (hrs)</th>
<th>Small Power Annual Operating Hours (hrs)</th>
<th>Daily Occupancy Profile</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bldg C</td>
<td>Private Company Owned</td>
<td>✓ ✓ X</td>
<td>✓ X X X X</td>
<td>✓ ✓ ✓ X X X X</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ X X X X X X X ✓ X ✓ X X ✓ X X X X X X X X</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>X</td>
</tr>
<tr>
<td>Bldg D</td>
<td>Private Company Owned</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>X</td>
</tr>
<tr>
<td>Bldg E</td>
<td>Private Company Owned</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓ ✓</td>
<td>X</td>
</tr>
<tr>
<td>Total Building Numbers</td>
<td></td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
<td>3 3 3 3 3 3 2 3 3 3 3</td>
</tr>
<tr>
<td>% of Information Provided</td>
<td></td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
<td>100.0% 100.0% 100.0% 0.0% 0.0% 0.0% 0.0% 100.0% 100.0% 100.0% 0.0% 0.0%</td>
</tr>
</tbody>
</table>

**Note:**
1. ✓ denotes information or data were available.
2. X denotes information or data were not available.
3.3.1 **Collection of Power Consumption Data**

For the collection of the power consumption data, the private company owned buildings and the institutional building owned could provide the monthly chiller plant power consumption no matter via their log sheet, sub-meter for chiller plant or BMS. Therefore, this enables the engineers or building operator to understand the energy consumption of the chiller plant performance monthly or annually. However, there was no any chiller power consumption data available by the government owned buildings. In fact, the Hong Kong government is now carrying out a programme for installing sub-meters in order to monitor the centralized building services equipment including chiller plant. It was also investigated that all the surveyed government owned buildings have installed with sub-meters. Unfortunately, the building operators did not record the power consumption data for the chiller plant regularly. As a result, no chiller plant power consumption data were available.

Besides, similar result was also observed in the availability of lighting power consumption. In order to monitor the power consumption of the lighting power consumption, the sub-meters were all installed in private company owned buildings as well as institutional building owned. Some of the government owned buildings (including Building A, B, F, I, J, K and L) could provide their lighting schedules (including quantity of the light fitting, light fitting rated power and daily operating hours) so that it was possible to estimate the monthly or annually lighting power consumption rather than by monitoring the sub-meters. In fact, this is another way round method to collect the lighting power consumption data.

It was so surprising that only the institutional building had a sub-meter to monitor
the small power consumption data and the building operator also took records regularly. Due to the multi-tenant in nature, the private property management company will normally not responsible for monitoring and recording the small power consumption in tenants’ areas. Also, the total power consumption of the tenants’ areas is usually monitored by the meters located in electrical meter rooms that are normally provided by the power utility company. No energy breakdown (i.e. HVAC load, lighting load and small power load) could be achieved. Therefore, no small power consumption data were available from private company owned buildings. Similar situation was also existed in government owned buildings. Despite sub-meters for monitoring the small power circuit were installed, no record was conducted.

Notwithstanding the design criteria for lighting power density and small power density for different types of buildings are available in building performance based code and building energy code [EMSD 2007; EMSD 2012], those design criteria may be different from the actual installation. As a result, over-estimation or under-estimation may be occurred. Therefore, it is suggested to the building operator or owner to keep on monitoring and recording the power consumption data of chiller plant power consumption, lighting power consumption and small power consumption regularly (monthly record is preferable).

3.3.2 Collection of Occupancy Data

In general, the occupancy data embraces: (a) occupancy density, (b) people sensible load and (c) people latent load for each functional use area. Those types of data can be achieved by referring to the building design report or O&M manual. Nonetheless,
due to the historical problem, some buildings may be changed a lot of building owners or property management company etc. Therefore, no building design document available is often found in most of existing buildings in Hong Kong especially for private owned buildings. Hopefully, some government buildings (e.g. Building F) built not more than 10 years still have the building design documents as well as O&M manual. For the government buildings that were built no more 10 years, no such information or data available will be the case. Apart from that, it was also discovered that the institutional building owned could still provide the design document so that the occupancy data could be collected. It is noted that this institutional building was built in 2005.

In case of there is no any occupancy data available, adopting the building performance based code and building energy code launched by EMSD [EMSD 2007; EMSD 2012] for occupancy density as well as ASHRAE [2009] for people load is also acceptable. However, such design criteria may not match with the existing situation.

### 3.3.3 Collection of Chiller Plant Operating Data

The average system COP of the chiller plant can reflect the overall chiller plant efficiency regarding the energy consumption. Therefore, this record is also vital for the engineers or building operator to analyze the energy efficiency of the chiller plant and in terms of total building electricity consumption. In order to monitor and record the system COP of the chiller plant, it relies on the provision of the sub-meters for monitoring the electricity consumption of every piece of major equipment as well as chilled water temperature sensors and chilled water flow meter for monitoring the
cooling load of the chiller plant. As shown in Table 3.14, the private owned buildings and institutional building owned could have the chiller operating data provided regardless of the accuracy of the chiller water temperature sensors and chilled water flow meters. For government owned buildings, no such operating data were available. Due to this reason, it is very difficult to carry out the energy performance analysis on chiller plant for government owned building especially for the buildings that were built more than 10 years.

### 3.3.4 Collection of Operating Hours Data

Basically, the building services system operation of the most of the surveyed buildings, no matter government owned buildings, private company owned buildings or institutional buildings, are automatic. The operating hours depend on the time schedule settings in the BMS or CCMS (central control and monitoring system) programme. Therefore, this type of data could be easily achieved from most of the surveyed buildings. Nonetheless, it was found that Building I, J, K and L are the community hall. Their on/off operations of the building services system (including chiller plant, lighting system and small power) are controlled manually and depend on the booking schedule of the hall. Therefore, it was very difficult to obtain the actual daily or monthly operating hours. As a result, no such operating hours data were provided from community halls.

Therefore, it is recommended the building operator of the community hall should record the operating schedule of the building services system day by day.
Furthermore, the estimation of the occupancy load (i.e. sensible and latent load) at different hours during a day is vital. One of the essential information is the occupancy profile. Refer to Table 3.14, it could be found that no any building could provide this type of information. This makes very difficult to determine the occupancy load of the HVAC system. Despite the occupancy profile for particular types of premises are available from the building performance based code [EMSD 2007], it is still suggested the building operator to conduct a survey on occupancy profile for different functional use areas within building hourly during typical weekdays and weekend.
3.4 DISSECTION OF BUILDING ENERGY CONSUMPTION USING ENGINEERING MODELING OF BUILDING ENERGY PERFORMANCE

With supplementing the previous section concerning about the problems on analyzing the electricity consumption of various types of the major building services systems in existing buildings in Hong Kong, it is possible to have a detailed building electricity consumption analysis with using the thermal performance line model technique. This section provides a general idea with a case study on understanding the building energy signature with regard to weather and non-weather dependent loads (i.e. building services systems) owing to conduct the detailed analysis on the dissection of the building electricity consumption by using an engineering modeling approach.

3.4.1 Methodology

In order to analyze the detailed electricity consumption with different energy consumer components (i.e. building services systems) within building, engineering modeling approach with using the available simulation software packages have been widely adopted in the recent 30 decades (for instance: BLAST, DOE-2, eQUEST, EnergyPlus, HAP v4.61, TRACE 700, SPARK, HVACSIM+, HTB2 with BECON or TRNSYS etc.). The detailed information about the comparison of common types of building energy simulation tools can be found from Hui [1996], Yik and Burnett [1996], Yu [2001] and California Energy Commission [2002]. The main concept of using these available simulation tools is to firstly generate the cooling/heating load requirement of a building based on different sorts of input parameters of the building.
characteristics. After that, the sizing of the building services systems (i.e. HVAC system etc.) can then be implemented with inputting the design parameters regarding the building services system. Finally, the annual building electricity consumption (e.g. hourly or monthly base) can be simulated.

Among different simulation tools, eQUEST was selected for the study. eQUEST stands for Quick Energy Simulation Tool which is used for building energy performance design tool [Energy Efficiency & Renewable Energy 2011]. This simulation tool is a freeware and open to public. The latest version is Version 3.64. eQUEST is accomplished by combining a building creation wizard, an energy efficiency measure wizard and graphical reporting with a simulation engine derived from the latest version of DOE-2. In general, this simulation tool has been commonly used in commercial and research sectors. In the meantime, the simulation tool is also supported by the department of energy (DOE) in U.S.A.

In this study, a monthly base building electricity consumption with different types of energy consumer components including HVAC system, lighting system, small power system and lift & escalator system were simulated. A thermal performance line model technique was also applied in order to demarcate the weather and non-weather dependent loads for energy saving measures.

3.4.2 Building Description

The modeling building adopted in this study is a commercial building, with sustainable design features, located in Hong Kong. This building consists of 40 floors including basement floor, with gross floor area of 66,000 m², which mainly
embraces shopping mall, food court, sky garden, offices and car parks. Table 3.14 summarizes the general information of the building.

Figure 3.14  Modeling Building Outlook

<table>
<thead>
<tr>
<th>Floor</th>
<th>Function</th>
</tr>
</thead>
<tbody>
<tr>
<td>B4/F – B3/F</td>
<td>Car Park</td>
</tr>
<tr>
<td>B2/F – 15/F</td>
<td>Retail Shops, Arcade and Food &amp; Beverage (F&amp;B)</td>
</tr>
<tr>
<td>16/F – 19/F, 35/F – 36/F</td>
<td>Refuge Floor and E&amp;M Plantroom</td>
</tr>
<tr>
<td>20/F – 34/F, 37/F – 38/F</td>
<td>Office</td>
</tr>
</tbody>
</table>
3.4.3 **Input Parameters for Building Modeling**

The input parameters for simulating the annual building cooling load and annual building electricity consumption are categorized into 7 types which include: (a) building site location, (b) building envelope components, (c) indoor condition, (d) HVAC system, (e) lighting system, (f) equipment loads, and (g) operating schedules.

The details of the input parameters for modeling the building are presented in Appendix B. It is noted that the electrical consumption of the mechanical ventilation system serving the car park floors and the natural gas consumption for F&B areas (hot water supply) are excluded in this study.

**Figure 3.15 Building Modeling Input Parameter Categories**
3.4.4 Results and Discussions

3.4.4.1 Building Electricity Consumption for Different Energy Consumers

The results of the annual building electricity consumption for different energy consumers for the modeled building are tabulated in Table 3.15. As shown in Table 3.15, the annual building electricity consumption for the modeled building is 13,255,747 kWh per annum. The annual chiller plant electricity consumption is 4,690,392 kWh per annum which is the major building electricity consumption of the modeled building (i.e. 35.4% of total) comparing to other building energy consumers. Equipment and lighting system occupy the second and third largest building electricity consumption of the modeled building.
### Table 3.15  Summary of Annual Building Energy Consumption (kWh) for Modeled Building

<table>
<thead>
<tr>
<th>Rank</th>
<th>Component</th>
<th>Jan</th>
<th>Feb</th>
<th>Mar</th>
<th>Apr</th>
<th>May</th>
<th>Jun</th>
<th>Jul</th>
<th>Aug</th>
<th>Sep</th>
<th>Oct</th>
<th>Nov</th>
<th>Dec</th>
<th>Annual Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Equipment</td>
<td>326,762</td>
<td>301,724</td>
<td>341,339</td>
<td>329,607</td>
<td>331,060</td>
<td>329,607</td>
<td>335,582</td>
<td>336,877</td>
<td>325,308</td>
<td>331,060</td>
<td>321,070</td>
<td>332,802</td>
<td>3,942,798</td>
</tr>
<tr>
<td>2</td>
<td>Lighting</td>
<td>286,124</td>
<td>260,382</td>
<td>290,111</td>
<td>279,846</td>
<td>286,093</td>
<td>279,543</td>
<td>287,079</td>
<td>287,955</td>
<td>279,170</td>
<td>286,716</td>
<td>278,457</td>
<td>288,186</td>
<td>3,389,662</td>
</tr>
<tr>
<td>3</td>
<td>Cooling</td>
<td>64,596</td>
<td>61,697</td>
<td>117,835</td>
<td>189,793</td>
<td>272,860</td>
<td>329,049</td>
<td>350,809</td>
<td>344,845</td>
<td>308,248</td>
<td>236,509</td>
<td>150,645</td>
<td>80,241</td>
<td>2,507,127</td>
</tr>
<tr>
<td>4</td>
<td>AC Fans</td>
<td>83,843</td>
<td>78,150</td>
<td>88,297</td>
<td>88,400</td>
<td>93,393</td>
<td>97,688</td>
<td>102,332</td>
<td>100,771</td>
<td>95,079</td>
<td>92,914</td>
<td>85,186</td>
<td>86,627</td>
<td>1,092,680</td>
</tr>
<tr>
<td>5</td>
<td>Lifts and Escalators</td>
<td>91,506</td>
<td>82,650</td>
<td>91,506</td>
<td>88,554</td>
<td>91,506</td>
<td>88,554</td>
<td>91,506</td>
<td>88,554</td>
<td>91,506</td>
<td>88,554</td>
<td>91,506</td>
<td>88,554</td>
<td>1,077,405</td>
</tr>
<tr>
<td>6</td>
<td>AC Pumps</td>
<td>36,443</td>
<td>35,336</td>
<td>51,932</td>
<td>61,084</td>
<td>71,924</td>
<td>85,959</td>
<td>92,614</td>
<td>90,472</td>
<td>77,620</td>
<td>66,237</td>
<td>54,590</td>
<td>41,184</td>
<td>765,395</td>
</tr>
<tr>
<td>7</td>
<td>Heat Rejection</td>
<td>5,171</td>
<td>5,043</td>
<td>11,819</td>
<td>20,200</td>
<td>32,445</td>
<td>50,234</td>
<td>54,793</td>
<td>51,886</td>
<td>40,922</td>
<td>24,373</td>
<td>15,344</td>
<td>6,648</td>
<td>318,878</td>
</tr>
<tr>
<td>8</td>
<td>Façade Lighting</td>
<td>13,206</td>
<td>11,928</td>
<td>13,206</td>
<td>12,780</td>
<td>13,206</td>
<td>12,780</td>
<td>13,206</td>
<td>12,780</td>
<td>13,206</td>
<td>12,780</td>
<td>13,206</td>
<td>155,490</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Heating</td>
<td>3,521</td>
<td>1,833</td>
<td>315</td>
<td>33</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>13</td>
<td>597</td>
<td>6,312</td>
</tr>
<tr>
<td><strong>Monthly Total</strong></td>
<td><strong>911,172</strong></td>
<td><strong>838,743</strong></td>
<td><strong>1,006,360</strong></td>
<td><strong>1,070,297</strong></td>
<td><strong>1,192,487</strong></td>
<td><strong>1,273,414</strong></td>
<td><strong>1,327,921</strong></td>
<td><strong>1,317,518</strong></td>
<td><strong>1,227,681</strong></td>
<td><strong>1,142,521</strong></td>
<td><strong>1,006,639</strong></td>
<td><strong>940,997</strong></td>
<td><strong>13,255,747</strong></td>
<td></td>
</tr>
</tbody>
</table>
Figure 3.16  Annual Building Electricity Consumption Profile for Modeled Building

Figure 3.17  Annual Building Electricity Consumption Breakdown by Percentage for Modeled Building
3.4.4.2 Influence of Climatic Changes on Building Energy Consumption with Different Energy Consumers

After generating the annual building electricity consumption results for each month, a plot between the monthly building electricity consumption for different energy consumers within building and $T_o^2$ as well as $T_o W_o$ can then be carried out.

As illustrated in Figure 3.18 and 3.19, four building energy consumers (including lighting, façade lighting, small power, lifts and escalators) belong to non-weather load since their electricity consumption was not influenced by $T_o^2$ and $T_o W_o$ (i.e. as shown in horizontal line in the graph). Alternatively, all the weather dependent loads were related to the electricity consumption of the chiller plant (i.e. as shown in straight lines with slopes). It can also be observed that the phenomena of the two graphs are more or less the same.

It is noted that Figure 3.18 and 3.19 are used for identifying the weather and non-weather load only while the ranking of the energy consumers within the modeled building is not taken into consideration. For instance, the thermal performance line of heating is on the top when comparing with other energy consumers. It does not mean the electricity consumption of heating is the largest of the modeling building. Instead, the electricity consumption of heating can be determined by subtracting the thermal performance line between heating and AC pumps. For determining the electricity consumption of AC pumps, it can be calculated by subtracting the thermal performance line between AC pumps and heat rejection. For evaluating the electricity consumption of equipment, it can be determined by subtracting the thermal performance line between equipment and
façade lighting.

Figure 3.18  A Plot of Building Electricity Consumption of Different Energy Consumers with Building Against Climatic Variable (T_o^2) for Modeled Building
Figure 3.19  A Plot of Building Electricity Consumption of Different Energy Consumers with Building Against Climatic Variable ($T_o W_o$) for Modeled Building
In order to further investigate how the significant of the climatic variable influencing on the electricity consumption of the chiller plant including its major equipment within the modeled building, additional analysis by correlating the electricity consumption of each major equipment of the chiller plant to the climatic variable (i.e. linear regression) was carried out.

By comparing the slope of the thermal performance line model, the degree of the influence of the climatic variable on the major equipment of the chiller plant can be identified. The thermal performance line model for temperature dependent \( (T_o^2) \) and energy dependent \( (T_o W_o) \) for the modeled building are illustrated in Figure 3.20 and 3.21. Apart from that, the results of the thermal performance line models for the major equipment of the chiller plant are summarized in Table. 3.16.
Figure 3.20  Thermal Performance Line Models – Temperature Dependent for Modeled Building
CHAPTER 3
CHILLER PERFORMANCE ASSESSMENT BASED ON BUILDING THERMAL PERFORMANCE LINE CONCEPT

Figure 3.21 Thermal Performance Line Models – Energy Dependent for Modeled Building

Table 3.16 Summary of Thermal Performance Line Model Results for Major Equipment of Chiller Plant of Modeled Building

<table>
<thead>
<tr>
<th>Type</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature ($T_{o2}$)</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature and Moisture Content ($T_{o2}W_o$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$R^2$</td>
<td>Slope</td>
</tr>
<tr>
<td>Cooling</td>
<td>0.9711</td>
<td>+525.180</td>
</tr>
<tr>
<td>AC Pumps</td>
<td>0.9536</td>
<td>+94.705</td>
</tr>
<tr>
<td>Heat Rejection</td>
<td>0.9511</td>
<td>+87.602</td>
</tr>
<tr>
<td>AC Fans</td>
<td>0.8845</td>
<td>+32.333</td>
</tr>
<tr>
<td>Heating</td>
<td>0.4052</td>
<td>-3.269</td>
</tr>
</tbody>
</table>
Based on the slope of the thermal performance line model of each major equipment of the chiller plant, it was discovered that the electricity consumption of cooling was affected by climatic variable significantly followed by AC pumps and heating rejection. Conversely, the slope of the thermal performance line model for AC fans was low when compared with cooling, AC pumps and heat rejection. Refer to Figure 3.20 and 3.21, it can be clearly observed that the changes of the $T_o^2$ or $T_oW_o$ influencing on the electricity consumption of AC fans was small. Due to this reason, it can be concluded that AC fans may be regarded as non-weather dependent load. For the thermal performance line model of heating, negative slope was investigated. It implies that when the climatic variable increases, the electricity consumption of heating decreases or vice versa. Therefore, this result makes sense since heating is required when the outdoor air dry-bulb temperature is too low (below 15.0°C).

The percentage changes of building energy consumption for increasing 0.5°C against the baseline outdoor air dry-bulb temperature (i.e. the average mean monthly outdoor air temperature over the past three years – 23.3°C), based on the thermal performance line models are summarized in Table 3.18. Since $T_o^2$ can directly reflect how the outdoor thermal environment influences the building energy consumption, the thermal performance line model with $T_o^2$ is selected in this study.
Table 3.17 Prediction of Percentage Changes of Chiller Plant Electricity Consumption for Modeled Building

<table>
<thead>
<tr>
<th>$T_0$</th>
<th>$T_{to}$</th>
<th>Cooling kWh</th>
<th>% Change</th>
<th>Heat Rejection kWh</th>
<th>% Change</th>
<th>AC Pumps kWh</th>
<th>% Change</th>
<th>AC Fans kWh</th>
<th>% Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>198,393</td>
<td>-</td>
<td>61,883</td>
<td>-</td>
<td>24,815</td>
<td>-</td>
<td>90,408</td>
<td>-</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>210,708</td>
<td>+6.2</td>
<td>64,104</td>
<td>+3.6</td>
<td>26,869</td>
<td>+8.3</td>
<td>91,166</td>
<td>+0.8</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>223,286</td>
<td>+12.5</td>
<td>66,372</td>
<td>+7.3</td>
<td>28,967</td>
<td>+16.7</td>
<td>91,940</td>
<td>+1.7</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>236,127</td>
<td>+19.0</td>
<td>68,688</td>
<td>+11.0</td>
<td>31,109</td>
<td>+25.4</td>
<td>92,731</td>
<td>+2.6</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>249,230</td>
<td>+25.6</td>
<td>71,050</td>
<td>+14.8</td>
<td>33,295</td>
<td>+34.2</td>
<td>93,538</td>
<td>+3.5</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>262,596</td>
<td>+32.4</td>
<td>73,461</td>
<td>+18.7</td>
<td>35,524</td>
<td>+43.2</td>
<td>94,361</td>
<td>+4.4</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>276,225</td>
<td>+39.2</td>
<td>75,918</td>
<td>+22.7</td>
<td>37,798</td>
<td>+52.3</td>
<td>95,200</td>
<td>+5.3</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>290,116</td>
<td>+46.2</td>
<td>78,423</td>
<td>+26.7</td>
<td>40,115</td>
<td>+61.7</td>
<td>96,055</td>
<td>+6.2</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>304,269</td>
<td>+53.4</td>
<td>80,976</td>
<td>+30.9</td>
<td>42,475</td>
<td>+71.2</td>
<td>96,926</td>
<td>+7.2</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>318,685</td>
<td>+60.6</td>
<td>83,575</td>
<td>+35.1</td>
<td>44,880</td>
<td>+80.9</td>
<td>97,814</td>
<td>+8.2</td>
</tr>
</tbody>
</table>
3.5  ENGINEERING MODELING ANALYSIS ON CHILLER PERFORMANCE BASED ON BUILDING ENERGY SIGNATURE

In general, chiller or chiller plant optimization can be divided into two categories and they are: (a) operation optimization and (b) maintenance optimization. For the former one, it is related by re-adjusting the control set point(s) in order to maximum the chiller or system \( \text{COP} \), for energy saving purpose, without scarifying the indoor thermal comfort environment (i.e. still can satisfy with the building cooling load). This type of optimization has been elaborated to discuss in Chapter One. For the maintenance optimization, it involves in maintaining the operating condition of the components of a chiller in a proper way so that the chiller or system \( \text{COP} \) can be kept as high as possible without any deterioration or fault occurred. In fact, the common faults occurred in a chiller can be reflected by seven KPIs discussed in Chapter 2.

Previous sections in this Chapter discuss the changes of the climatic variables influencing on the building and chiller plant electricity consumption with adopting the thermal performance line model. In this section, the effects of component faults on the chillers with taking into account the changes of the climatic variable on the electricity consumption of the chiller plant is analyzed.

3.5.1 Methodology

Comstock and Braun [1999a] investigated various types of chiller faults on the performance of the chillers based on the experimental chillers. This study was accredited in ASHRAE research project 1043-RP. The results of the faults data can
also be found [Comstock and Braun 1999b]. With make use of the building model developed in Section 3.4, the chiller plant electricity consumption affected by the faults of the components on the chillers were studied. The influences of the faults of the components of the chillers on the chiller plant electricity consumption were simulated by artificially adjusting some of the input parameters of the HVAC system via eQUEST. After generating the results, thermal performance line model technique was applied in order to explore how the changes of the climatic variables coincident with the chiller faults affecting the electricity consumption of the chiller plant for the modeled building. The electricity consumption of the chiller plant simulated in Section 3.4 was regarded as base case (i.e. free from fault). Four types chiller faults which typically encountered in chiller plants were selected in this study and they are (a) condenser fouling, (b) refrigerant leakage, (c) condensing water flow reduction and (d) chilled water flow reduction. The details of the artificial chiller component faults are summarized in Table 3.18.
<table>
<thead>
<tr>
<th>Fault Type</th>
<th>Severity Level</th>
<th>Fault Simulation</th>
<th>Chiller COP</th>
<th>Chiller Power</th>
<th>Water Pump Flow</th>
<th>Water Pump Power</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condenser Fouling</td>
<td>12% Reduction in Condenser Tube</td>
<td>Reduced by 0.8%</td>
<td>Increased by 0.6%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9%</td>
<td>0.7%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>1.9%</td>
<td>1.9%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>45%</td>
<td>4.1%</td>
<td>3.9%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Refrigerant Leakage</td>
<td>10% Refrigerant Leakage</td>
<td>Reduced by 0.4%</td>
<td>Increased by 0.6%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.2%</td>
<td>0.4%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>1.2%</td>
<td>0.9%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.8%</td>
<td>0.5%</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Condensing Water Flow Reduction</td>
<td>10% Condensing Water Flow Reduction</td>
<td>Reduced by 0.8%</td>
<td>Increased by 0.8%</td>
<td>Reduced by 10.0%</td>
<td>Reduced by 0.5%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>2.5%</td>
<td>2.7%</td>
<td>20%</td>
<td>7.5%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>3.2%</td>
<td>3.9%</td>
<td>30%</td>
<td>12.5%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>6.1%</td>
<td>7.0%</td>
<td>40%</td>
<td>17.5%</td>
<td></td>
</tr>
<tr>
<td>Chilled Water Flow Reduction</td>
<td>10% Chilled Water Flow Reduction</td>
<td>Reduced by 0.2%</td>
<td>-</td>
<td>Reduced by 10%</td>
<td>Reduced by 5.7%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.5%</td>
<td>Increased by 0.4%</td>
<td>20%</td>
<td>8.6%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>1.5%</td>
<td>0.4%</td>
<td>30%</td>
<td>14.3%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>2.4%</td>
<td>0.5%</td>
<td>40%</td>
<td>20.0%</td>
<td></td>
</tr>
</tbody>
</table>
3.5.2 Results and Discussions

3.5.2.1 Condenser Fouling

In this analysis, the condenser fouling fault is divided into four severity levels which includes 12%, 20% 30% and 40% of the condenser tube blockage. The higher the severity level, the lower the overall heat transfer efficiency between the refrigerant tube and condenser water pipe taking place at condenser would be. This phenomena incurs significantly high refrigerant temperature causing the degree of the opening of the expansion device open wider in order to maintain appropriate/avoid excessive refrigerant pressure differential (or temperature lift) between condenser and evaporator.

When comparing with the base case, there was no influence on the electricity consumption for heat rejection and AC pumps under different severity level of condenser fouling fault. Conversely, it was discovered that only the electricity consumption of the chillers was affected by condenser fouling. The percentage changes of the annual electricity consumption of the chillers compared with base case under 12%, 20%, 30% and 40% of condenser fouling were increased to 0.594%, 1.186%, 1.771% and 4.263% respectively of the chiller electricity consumption.
### Table 3.19  Summary of Percentage Changes of Electricity Consumption of Chillers for Modeled Building under Different Severity Levels of Condenser Fouling

<table>
<thead>
<tr>
<th>Month</th>
<th>12%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jan</td>
<td>+0.548</td>
<td>+1.132</td>
<td>+1.688</td>
<td>+4.121</td>
</tr>
<tr>
<td>Feb</td>
<td>+0.631</td>
<td>+1.222</td>
<td>+1.933</td>
<td>+4.457</td>
</tr>
<tr>
<td>Mar</td>
<td>+0.533</td>
<td>+1.120</td>
<td>+1.751</td>
<td>+4.101</td>
</tr>
<tr>
<td>Apr</td>
<td>+0.483</td>
<td>+1.063</td>
<td>+1.480</td>
<td>+3.680</td>
</tr>
<tr>
<td>May</td>
<td>+0.538</td>
<td>+1.114</td>
<td>+1.639</td>
<td>+3.938</td>
</tr>
<tr>
<td>Jun</td>
<td>+0.565</td>
<td>+1.169</td>
<td>+1.741</td>
<td>+4.298</td>
</tr>
<tr>
<td>Jul</td>
<td>+0.686</td>
<td>+1.288</td>
<td>+2.054</td>
<td>+4.723</td>
</tr>
<tr>
<td>Aug</td>
<td>+0.747</td>
<td>+1.362</td>
<td>+2.034</td>
<td>+4.800</td>
</tr>
<tr>
<td>Sep</td>
<td>+0.591</td>
<td>+1.166</td>
<td>+1.678</td>
<td>+4.189</td>
</tr>
<tr>
<td>Oct</td>
<td>+0.479</td>
<td>+1.056</td>
<td>+1.503</td>
<td>+3.749</td>
</tr>
<tr>
<td>Nov</td>
<td>+0.566</td>
<td>+1.155</td>
<td>+1.726</td>
<td>+4.196</td>
</tr>
<tr>
<td>Dec</td>
<td>+0.617</td>
<td>+1.213</td>
<td>+1.848</td>
<td>+4.418</td>
</tr>
<tr>
<td>Annual</td>
<td>+0.594</td>
<td>+1.186</td>
<td>+1.771</td>
<td>+4.263</td>
</tr>
</tbody>
</table>
By applying the thermal performance line model, the influence of the climatic variables as well as condenser fouling of chillers on the electricity consumption of the chiller and chiller plant can be investigated. Figure 3.22 to 3.25 illustrating the thermal performance line models for the electricity consumption of chillers and chiller plant under different severity levels of condenser fouling for modeled building. The results of $R^2$ and slope of each thermal performance line model (i.e. severity levels of condenser fouling) are summarized in Table 3.20. It is noted that the percentage chillers and chiller plant energy consumption for increasing 0.5°C against the baseline outdoor air dry-bulb temperature (i.e. the average mean monthly outdoor air temperature over the past three years – 23.3°C) with climatic variable changes for modeled building, based on the thermal performance line models are summarized in Table 3.21. Since $T_o^2$ can directly reflect how the outdoor thermal environment influences the building energy consumption, the thermal performance line model with $T_o^2$ is selected in this study.
Figure 3.22  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chillers due to Condenser Fouling)
Figure 3.23  Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chillers due to Condenser Fouling)
Figure 3.24  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Condenser Fouling)
Figure 3.25 Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Condenser Fouling)
### Table 3.20
Summary of Thermal Performance Line Model Results for Chillers and Chiller Plant of Modeled Building (due to Condenser Fault)

<table>
<thead>
<tr>
<th>Type</th>
<th>Severity Level</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature ($T_o$)</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature and Moisture Content ($T_oW_o$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>$R^2$</td>
<td>Slope</td>
</tr>
<tr>
<td>Chillers</td>
<td>0%</td>
<td>0.971</td>
<td>+0.0079</td>
</tr>
<tr>
<td></td>
<td>12%</td>
<td>0.9714</td>
<td>+0.0080</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9714</td>
<td>+0.0081</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9718</td>
<td>+0.0082</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.97923</td>
<td>+0.0083</td>
</tr>
<tr>
<td>Chiller Plant</td>
<td>0%</td>
<td>0.9570</td>
<td>0.0096</td>
</tr>
<tr>
<td></td>
<td>12%</td>
<td>0.9574</td>
<td>0.0097</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9575</td>
<td>0.0098</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9580</td>
<td>0.0099</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9587</td>
<td>0.0100</td>
</tr>
</tbody>
</table>

### Table 3.21
Prediction of Chillers and Chiller Plant Electricity Consumption in kWh per m² of Modeled Building (due to Condenser Fault at Different Severity Levels)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_o^2$</th>
<th>Chillers</th>
<th>Chiller Plant</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>12%</td>
<td>20%</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>3.641</td>
<td>3.695</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>3.842</td>
<td>3.898</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>4.046</td>
<td>4.105</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>4.904</td>
<td>4.973</td>
</tr>
</tbody>
</table>
3.5.2.2 Refrigerant Leakage

There are four severity levels (10%, 20%, 30% and 40%) of refrigerant leakage studied. According to the thermodynamic first law, the reduction of the refrigerant flow rate with constant change of specific enthalpy of refrigerant will cause a reduction of the log mean temperature difference of condenser for maintaining the demand of heat rejection load. Due to this reason, the chiller COP will then be decreased (i.e. more energy is required to remove the heat from refrigerant).

Similar to condenser fouling, there was no influence on the electricity consumption for heat rejection and AC pumps under different severity level of refrigerant leakage when compared with base case. Instead, only the electricity consumption of chillers was affected. Nevertheless, it was surprising that the refrigerant leakage at 10%, 20%, 30% and 40% were decreased 0.67%, 0.81%, 1.52% and 2.00% of the chiller electricity consumption. It means that small amount of energy saving could be achieved. The percentage changes of the electricity of chillers for modeled building under different severity levels of refrigerant leakage is summarized in Table 3.22.
Table 3.22  Summary of Percentage Changes of Electricity Consumption of Chillers for Modeled Building under Different Severity Levels of Refrigerant Leakage

<table>
<thead>
<tr>
<th>Month</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Jan</td>
<td>-0.45</td>
<td>-0.45</td>
<td>-1.93</td>
<td>-2.20</td>
</tr>
<tr>
<td>Feb</td>
<td>-0.73</td>
<td>-1.27</td>
<td>-1.95</td>
<td>-2.89</td>
</tr>
<tr>
<td>Mar</td>
<td>-0.17</td>
<td>-0.51</td>
<td>-1.14</td>
<td>-2.31</td>
</tr>
<tr>
<td>Apr</td>
<td>-1.44</td>
<td>-2.28</td>
<td>-2.89</td>
<td>3.54</td>
</tr>
<tr>
<td>May</td>
<td>-0.10</td>
<td>-0.13</td>
<td>-1.16</td>
<td>-1.95</td>
</tr>
<tr>
<td>Jun</td>
<td>-1.86</td>
<td>-2.12</td>
<td>-2.87</td>
<td>-2.97</td>
</tr>
<tr>
<td>Jul</td>
<td>-0.27</td>
<td>-0.56</td>
<td>-1.00</td>
<td>-1.24</td>
</tr>
<tr>
<td>Aug</td>
<td>-0.15</td>
<td>-0.38</td>
<td>-0.79</td>
<td>-1.05</td>
</tr>
<tr>
<td>Sep</td>
<td>-0.31</td>
<td>-0.55</td>
<td>-0.81</td>
<td>-1.62</td>
</tr>
<tr>
<td>Oct</td>
<td>-1.22</td>
<td>-0.38</td>
<td>-1.53</td>
<td>-1.56</td>
</tr>
<tr>
<td>Nov</td>
<td>-0.85</td>
<td>-0.72</td>
<td>-2.50</td>
<td>-2.97</td>
</tr>
<tr>
<td>Dec</td>
<td>-0.14</td>
<td>-0.22</td>
<td>-0.47</td>
<td>-1.38</td>
</tr>
<tr>
<td>Annual</td>
<td>-0.67</td>
<td>-0.81</td>
<td>-1.52</td>
<td>-2.00</td>
</tr>
</tbody>
</table>
Figure 3.26 to 3.29 illustrate the thermal performance line models for the electricity consumption of chillers and chiller plant under different severity levels of refrigerant leakage for modeled building. The results of $R^2$ and slope of each thermal performance line model (i.e. severity levels of refrigerant leakage) are summarized in Table 3.24. In addition, the predictions of the percentage changes of the chillers and chiller plant electricity consumption under different severity levels of refrigerant leakage as well as climatic variable changes for modeled building are presented in Table 3.24. As seen from Table 3.24, it was discovered that the degree of the slope decreases when the severity level of the refrigerant leakage is high which the electricity saving of the chillers and chiller plant could be obtained.
Figure 3.26  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chillers due to Refrigerant Leakage)
Figure 3.27 Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chillers due to Refrigerant Leakage)
Figure 3.28  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Refrigerant Leakage)
Figure 3.29  Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Refrigerant Leakage)
### Table 3.23  Summary of Thermal Performance Line Model Results for Chillers and Chiller Plant of Modeled Building (due to Refrigerant Leakage)

<table>
<thead>
<tr>
<th>Type</th>
<th>Severity Level</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature ($T_o$)</th>
<th>$R^2$</th>
<th>Slope</th>
<th>Y-Intercept</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature and Moisture Content ($T_oW_o$)</th>
<th>$R^2$</th>
<th>Slope</th>
<th>Y-Intercept</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chillers</td>
<td>0%</td>
<td>0.9711</td>
<td>0.0079</td>
<td>-1.2674</td>
<td>0.9717</td>
<td>0.0106</td>
<td>-0.3473</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>0.9739</td>
<td>0.0079</td>
<td>-1.2671</td>
<td>0.9739</td>
<td>0.0106</td>
<td>-0.3469</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9745</td>
<td>0.0078</td>
<td>-1.2655</td>
<td>0.9745</td>
<td>0.0105</td>
<td>-0.3486</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9753</td>
<td>0.0078</td>
<td>-1.2614</td>
<td>0.9749</td>
<td>0.0104</td>
<td>-0.3431</td>
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<tr>
<td></td>
<td>40%</td>
<td>0.9752</td>
<td>0.0077</td>
<td>-1.2770</td>
<td>0.9747</td>
<td>0.0103</td>
<td>-0.3661</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chiller Plant</td>
<td>0%</td>
<td>0.957</td>
<td>0.0096</td>
<td>-1.4392</td>
<td>0.9608</td>
<td>0.0129</td>
<td>-0.2074</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>0.9739</td>
<td>0.0096</td>
<td>-1.4384</td>
<td>0.9739</td>
<td>0.0129</td>
<td>-0.2069</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.959</td>
<td>0.0095</td>
<td>-1.4373</td>
<td>0.9625</td>
<td>0.0125</td>
<td>-0.2087</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9606</td>
<td>0.0095</td>
<td>-1.4434</td>
<td>0.9636</td>
<td>0.0128</td>
<td>-0.2096</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9600</td>
<td>0.0094</td>
<td>-1.4489</td>
<td>0.9630</td>
<td>0.0127</td>
<td>-0.2010</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Table 3.24  Prediction of Chillers and Chiller Plant Electricity Consumption in kWh per m² of Modeled Building (due to Refrigerant Leakage)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_o^2$</th>
<th>Chillers</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
</tr>
</thead>
<tbody>
<tr>
<td>23.3</td>
<td>542.9</td>
<td></td>
<td>3.022</td>
<td>2.969</td>
<td>2.973</td>
<td>2.903</td>
<td>4.865</td>
<td>4.814</td>
<td>4.814</td>
<td>4.737</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td></td>
<td>3.207</td>
<td>3.152</td>
<td>3.157</td>
<td>3.084</td>
<td>5.091</td>
<td>5.038</td>
<td>5.038</td>
<td>4.958</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td></td>
<td>3.398</td>
<td>3.340</td>
<td>3.345</td>
<td>3.270</td>
<td>5.322</td>
<td>5.267</td>
<td>5.267</td>
<td>5.185</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td></td>
<td>3.591</td>
<td>3.532</td>
<td>3.536</td>
<td>3.459</td>
<td>5.557</td>
<td>5.499</td>
<td>5.499</td>
<td>5.415</td>
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<td>25.3</td>
<td>640.1</td>
<td></td>
<td>3.790</td>
<td>3.727</td>
<td>3.731</td>
<td>3.652</td>
<td>5.798</td>
<td>5.738</td>
<td>5.738</td>
<td>5.651</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td></td>
<td>3.991</td>
<td>3.926</td>
<td>3.930</td>
<td>3.848</td>
<td>6.043</td>
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<td>5.980</td>
<td>5.891</td>
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<tr>
<td>27.8</td>
<td>772.8</td>
<td></td>
<td>4.838</td>
<td>4.762</td>
<td>4.766</td>
<td>4.674</td>
<td>7.072</td>
<td>6.999</td>
<td>6.999</td>
<td>6.898</td>
</tr>
</tbody>
</table>
3.5.2.3 **Condensing Water Flow Reduction**

In order to test the influence of the condensing water flow reduction on the electricity consumption of the chiller plant, four severity levels (10%, 20%, 30% and 40%) of condensing water flow rate reduction were studied. The situation of condensing water flow reduction is similar to the condenser fouling since the heat rejection from the refrigerant to the condensing water taking place at condenser is decreased due to inadequate condensing water flow. Consequently, the refrigerant condensing temperature will be increased. As a result, the chiller *COP* will then be decreased.

Unlike previous two chiller faults, condensing water flow reduction affects the electricity consumption of the chiller as well as condensing water pumps (AC pumps) only. Refer to Table 3.25, it can be seen that the condensing water flow reduction at 10%, 20%, 30% and 40% severity level caused an increase in the annual electricity consumption of chillers of 0.132%, 0.824%, 1.964% and 2.516% respectively. For the AC pumps, reduction of the condensing water flow at 10%, 20%, 30% and 40% severity levels affects 1.196%, 1.796%, 2.317% and 2.630% of reduction of annual electricity consumption of AC pumps respectively.
Table 3.25  Summary of Percentage Changes of the Electricity Consumption of Chillers and AC Pumps for Modeled Building under Different Severity Levels of Condensing Water Flow Reduction

<table>
<thead>
<tr>
<th></th>
<th>Severity Level of Condenser Water Flow Reduction Fault</th>
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<tbody>
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<td></td>
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<td>Chillers</td>
<td>AC Pumps</td>
<td>Chillers</td>
<td>AC Pumps</td>
<td>Chillers</td>
<td>AC Pumps</td>
<td>Chillers</td>
</tr>
<tr>
<td>Month</td>
<td></td>
<td>10%</td>
<td>20%</td>
<td>30%</td>
<td>40%</td>
<td>10%</td>
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<tr>
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<td>+0.811</td>
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<td>+2.510</td>
<td>-0.605</td>
<td>-1.040</td>
<td>-1.585</td>
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<td>May</td>
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<td>+0.824</td>
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<td>-1.624</td>
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<td>+0.209</td>
<td>+0.795</td>
<td>+1.881</td>
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<td>-0.435</td>
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<tr>
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<td>+0.187</td>
<td>+0.828</td>
<td>+1.912</td>
<td>+2.488</td>
<td>-0.624</td>
<td>-0.983</td>
<td>-1.342</td>
</tr>
<tr>
<td>Aug</td>
<td></td>
<td>+0.159</td>
<td>+0.905</td>
<td>+1.948</td>
<td>+2.546</td>
<td>-4.446</td>
<td>-4.593</td>
<td>-4.960</td>
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<td>Sep</td>
<td></td>
<td>+0.099</td>
<td>+0.863</td>
<td>+1.938</td>
<td>+2.451</td>
<td>-0.704</td>
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<td>-1.904</td>
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<tr>
<td>Oct</td>
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<td>+0.045</td>
<td>+0.749</td>
<td>+1.975</td>
<td>+2.485</td>
<td>-0.908</td>
<td>-1.611</td>
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<td>+0.106</td>
<td>+0.743</td>
<td>+2.061</td>
<td>+2.649</td>
<td>-0.232</td>
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<td>-2.472</td>
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<td>+0.824</td>
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<td>+2.516</td>
<td>-1.196</td>
<td>-1.796</td>
<td>-2.317</td>
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</table>
Figure 3.30 to 3.35 illustrate the thermal performance line models for the electricity consumption of chillers, AC pumps and chiller plant under different severity levels of condensing water flow reduction for modeled building. The results of $R^2$ and slope of each thermal performance line model (i.e. severity levels of condensing water flow reduction) are summarized in Table 3.26. Furthermore, the predictions of the percentage changes of the chillers, AC pumps and chiller plant electricity consumption under different severity levels of condensing water flow reduction as well as climatic variable changes for modeled building are summarized in Table 3.27. Refer to Table 3.28, it was discovered that the degree of the slope of the thermal performance lines of the AC pumps decreases when the severity level of the condensing water flow reduction is high which the slope of the thermal performance line model of chiller and chiller plant were increased. Despite the electricity consumption of the AC pumps was decreased, it was difficult to reflect from the slope of the thermal performance line model for each severity level of condensing water flow reduction. However, the reduction of the electricity consumption of the AC pumps due to the reduction of the condensing water flow at different severity levels can be reflected by using the y-intercept of the thermal performance line model of AC pumps.
Figure 3.30   Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chillers due to Condensing Water Flow Reduction)
Figure 3.31  Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chillers due to Condensing Water Flow Reduction)
Figure 3.32  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of AC Pumps due to Condensing Water Flow Reduction)
Figure 3.33  Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of AC Pumps due to Condensing Water Flow Reduction)
Figure 3.34 Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Condensing Water Flow Reduction)
Figure 3.35 Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Condensing Water Flow Reduction)
Table 3.26  Summary of Thermal Performance Line Model Results for Chillers, AC Pumps and Chiller Plant of Modeled Building (due to Condensing Water Flow Reduction)

<table>
<thead>
<tr>
<th>Type</th>
<th>Severity Level</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature ($T_{o2}$)</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature and Moisture Content ($T_{oW}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>$R^2$ Slope Y-Intercept</td>
<td>$R^2$ Slope Y-Intercept</td>
</tr>
<tr>
<td>Chillers</td>
<td>0%</td>
<td>0.9711 +0.0079 -1.2674</td>
<td>0.9711 +0.0079 -1.2674</td>
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<tr>
<td></td>
<td>10%</td>
<td>0.9711 +0.0079 -1.2706</td>
<td>0.9711 +0.0079 -1.2706</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9712 +0.0080 -1.2795</td>
<td>0.9712 +0.0080 -1.2795</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9711 +0.0080 -1.2887</td>
<td>0.9711 +0.0080 -1.2887</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9723 +0.0083 -1.3373</td>
<td>0.9723 +0.0083 -1.3373</td>
</tr>
<tr>
<td>AC Pumps</td>
<td>0%</td>
<td>0.9498 +0.0014 +0.1640</td>
<td>0.9376 +0.0017 +0.3307</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>0.9498 +0.0014 +0.1694</td>
<td>0.9335 +0.0017 +0.3328</td>
</tr>
<tr>
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<td>20%</td>
<td>0.9498 +0.0014 +0.1645</td>
<td>0.9344 +0.0017 +0.3273</td>
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<tr>
<td></td>
<td>30%</td>
<td>0.9501 +0.0014 +0.1595</td>
<td>0.9344 +0.0017 +0.3223</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9536 +0.0014 +0.1565</td>
<td>0.9344 +0.0017 +0.3193</td>
</tr>
<tr>
<td>Chiller Plant</td>
<td>0%</td>
<td>0.9717 +0.0106 -1.4392</td>
<td>0.9717 +0.0106 -1.4392</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>0.9714 +0.0106 -1.4370</td>
<td>0.9714 +0.0106 -1.4370</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9714 +0.0107 -1.4508</td>
<td>0.9714 +0.0107 -1.4508</td>
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<td>30%</td>
<td>0.9713 +0.0108 -1.4651</td>
<td>0.9713 +0.0108 -1.4651</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9724 +0.0110 -1.5092</td>
<td>0.9724 +0.0110 -1.5092</td>
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### Table 3.27 Prediction of Chillers and AC Pumps Electricity Consumption in kWh per m$^2$ of Modeled Building (due to Condensing Water Flow Reduction)

<table>
<thead>
<tr>
<th>$T_o$</th>
<th>$T_o^2$</th>
<th>Chillers</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>AC Pumps</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
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</thead>
<tbody>
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<td>542.9</td>
<td>3.018</td>
<td>3.064</td>
<td>3.055</td>
<td>3.169</td>
<td>0.929</td>
<td>0.925</td>
<td>0.920</td>
<td>0.917</td>
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<td></td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>3.204</td>
<td>3.252</td>
<td>3.243</td>
<td>3.364</td>
<td>0.962</td>
<td>0.957</td>
<td>0.952</td>
<td>0.949</td>
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<td>590.5</td>
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<td>3.445</td>
<td>3.435</td>
<td>3.564</td>
<td>0.996</td>
<td>0.991</td>
<td>0.986</td>
<td>0.983</td>
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<tr>
<td>24.8</td>
<td>615.0</td>
<td>3.588</td>
<td>3.641</td>
<td>3.631</td>
<td>3.767</td>
<td>1.030</td>
<td>1.026</td>
<td>1.021</td>
<td>1.018</td>
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<tr>
<td>25.3</td>
<td>640.1</td>
<td>3.786</td>
<td>3.841</td>
<td>3.832</td>
<td>3.976</td>
<td>1.066</td>
<td>1.061</td>
<td>1.056</td>
<td>1.053</td>
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<td></td>
</tr>
<tr>
<td>25.8</td>
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<td>4.045</td>
<td>4.036</td>
<td>4.187</td>
<td>1.101</td>
<td>1.096</td>
<td>1.091</td>
<td>1.088</td>
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<tr>
<td>26.3</td>
<td>691.7</td>
<td>4.194</td>
<td>4.254</td>
<td>4.245</td>
<td>4.404</td>
<td>1.138</td>
<td>1.133</td>
<td>1.128</td>
<td>1.125</td>
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<tr>
<td>26.8</td>
<td>718.2</td>
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<td>4.466</td>
<td>4.457</td>
<td>4.624</td>
<td>1.175</td>
<td>1.170</td>
<td>1.165</td>
<td>1.162</td>
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<tr>
<td>27.3</td>
<td>745.3</td>
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<td>4.683</td>
<td>4.674</td>
<td>4.849</td>
<td>1.213</td>
<td>1.208</td>
<td>1.203</td>
<td>1.200</td>
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<tr>
<td>27.8</td>
<td>772.8</td>
<td>4.835</td>
<td>4.903</td>
<td>4.894</td>
<td>5.077</td>
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<td>1.246</td>
<td>1.241</td>
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### Table 3.28 Prediction of Chiller Plant Electricity Consumption in kWh per m$^2$ of Modeled Building (due to Condensing Water Flow Reduction)

<table>
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<th>$T_o$</th>
<th>$T_o^2$</th>
<th>Chiller Plant</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
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<td>23.3</td>
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<td>4.318</td>
<td>4.358</td>
<td>4.398</td>
<td>4.463</td>
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<tr>
<td>23.8</td>
<td>566.4</td>
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<td>4.610</td>
<td>4.652</td>
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<tr>
<td>24.3</td>
<td>590.5</td>
<td>4.822</td>
<td>4.868</td>
<td>4.912</td>
<td>4.986</td>
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<tr>
<td>24.8</td>
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<td>5.082</td>
<td>5.130</td>
<td>5.177</td>
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<tr>
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<td>5.398</td>
<td>5.448</td>
<td>5.532</td>
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<td>5.671</td>
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<td>5.812</td>
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<td>5.950</td>
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3.5.2.4 Chilled Water Flow Reduction

In order to test the influence of the chilled water flow reduction on the electricity consumption of the chiller plant, four severity levels (10%, 20%, 30% and 40%) of chilled water flow rate reduction were studied.

Similar to condensing water flow reduction, the electricity consumption of the chiller and chilled water pump (AC pumps) were influenced by the reduction of the chilled water flow. Table 3.29 summarizes the percentage of changes of the electricity consumption of the chillers and AC pumps due to the reduction of the chilled water flow at different severity levels at 10%, 20%, 30% and 40%. It was discovered that the annual electricity consumption of the chiller was increased of 0.601%, 2.451%, 3.560% and 6.624% due to the reduction of the chilled water flow at 10%, 20%, 30% and 40% severity level respectively.
### Table 3.29  Summary of Percentage Changes of Electricity Consumption of Chillers and AC Pumps for Modeled Building under Different Severity Levels of Chilled Water Flow Reduction

<table>
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<tr>
<th>Month</th>
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<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
<th>AC Pumps</th>
<th>10%</th>
<th>20%</th>
<th>30%</th>
<th>40%</th>
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<td>+6.573</td>
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<tr>
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<td>-1.951</td>
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<tr>
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<td>-3.544</td>
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<tr>
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<td>+2.453</td>
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<td>+6.785</td>
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<td>-1.672</td>
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<tr>
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<td>+3.978</td>
<td>+7.352</td>
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<td>-2.563</td>
<td>-2.635</td>
<td>-3.783</td>
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</tbody>
</table>
Figure 3.36 to 3.41 illustrate the thermal performance line models for the electricity consumption of chillers, AC pumps and chiller plant under different severity levels of chilled water flow reduction for modeled building. The results of $R^2$ and slope of each thermal performance line model (i.e. severity levels of condensing water flow reduction) are summarized in Table 3.30. In addition, the predictions of the percentage changes of the chillers, AC pumps and chiller plant electricity consumption under different severity levels of chilled water flow reduction as well as climatic variable changes for modeled building are summarized in Table 3.31. As showed in Table 3.32, it was clearly observed that the degree of the slope of the thermal performance line model of the AC pumps decreases when the severity level of the chilled water flow reduction is high which the slope of the thermal performance line of the chiller and chiller plant were increased. Despite the electricity consumption of the AC pumps was decreased, it was difficult to reflect from the slope of the thermal performance line model for each severity level of chilled water flow reduction. However, the reduction of the electricity consumption of the AC pumps due to the reduction of the condensing water flow at different severity levels can be reflected by using the y-intercept of the thermal performance line model of the AC pumps.
Figure 3.36  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chillers due to Chilled Water Flow Reduction)
Figure 3.37  Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chillers due to Chilled Water Flow Reduction)
Figure 3.38  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of AC Pumps due to Chilled Water Flow Reduction)
Figure 3.39  Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of AC Pumps due to Chilled Water Flow Reduction)
Figure 3.40  Thermal Performance Line Models – Temperature Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Chilled Water Flow Reduction)
Figure 3.41 Thermal Performance Line Models – Energy Dependent for Modeled Building (for Electricity Consumption of Chiller Plant due to Chilled Water Flow Reduction)
Table 3.30  Summary of Thermal Performance Line Model Results for Chillers, AC Pumps and Chiller Plant of Modeled Building (due to Chilled Water Flow Reduction)

<table>
<thead>
<tr>
<th>Type</th>
<th>Severity Level</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature ($T_o$)</th>
<th>Product of Mean Outdoor Air Dry-Bulb Temperature and Moisture Content ($T_o W_o$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>$R^2$</td>
<td>Slope</td>
</tr>
<tr>
<td>Chillers</td>
<td>0%</td>
<td>0.9711</td>
<td>-0.0079</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>0.9715</td>
<td>+0.0080</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9720</td>
<td>+0.0081</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9723</td>
<td>+0.0082</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9723</td>
<td>+0.0083</td>
</tr>
<tr>
<td>AC Pumps</td>
<td>0%</td>
<td>0.9536</td>
<td>+0.0014</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>0.9544</td>
<td>+0.0014</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9504</td>
<td>+0.0014</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9523</td>
<td>+0.0014</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9504</td>
<td>+0.0014</td>
</tr>
<tr>
<td>Chiller Plant</td>
<td>0%</td>
<td>0.9717</td>
<td>+0.0106</td>
</tr>
<tr>
<td></td>
<td>10%</td>
<td>0.9719</td>
<td>+0.0107</td>
</tr>
<tr>
<td></td>
<td>20%</td>
<td>0.9720</td>
<td>+0.0108</td>
</tr>
<tr>
<td></td>
<td>30%</td>
<td>0.9724</td>
<td>+0.0109</td>
</tr>
<tr>
<td></td>
<td>40%</td>
<td>0.9720</td>
<td>+0.0108</td>
</tr>
</tbody>
</table>
Table 3.31  Prediction of Chillers and AC Pumps Electricity Consumption in kWh per m$^2$ of Modeled Building (due to Chilled Water Flow Reduction)

<table>
<thead>
<tr>
<th>$T_e$</th>
<th>$T_e^2$</th>
<th>Chillers</th>
<th>AC Pumps</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>10%</td>
<td>20%</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>3.064</td>
<td>3.088</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>3.252</td>
<td>3.278</td>
</tr>
<tr>
<td>24.3</td>
<td>590.5</td>
<td>3.445</td>
<td>3.473</td>
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<tr>
<td>24.8</td>
<td>615.0</td>
<td>3.641</td>
<td>3.672</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>3.842</td>
<td>3.875</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>4.046</td>
<td>4.081</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>4.255</td>
<td>4.293</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>4.467</td>
<td>4.508</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>4.683</td>
<td>4.727</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>4.903</td>
<td>4.950</td>
</tr>
</tbody>
</table>

Table 3.32  Prediction of Chiller Plant Electricity Consumption in kWh per m$^2$ of Modeled Building (due to Chilled Water Flow Reduction)

<table>
<thead>
<tr>
<th>$T_e$</th>
<th>$T_e^2$</th>
<th>Chiller Plant</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>10%</td>
</tr>
<tr>
<td>23.3</td>
<td>542.9</td>
<td>4.353</td>
</tr>
<tr>
<td>23.8</td>
<td>566.4</td>
<td>4.604</td>
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<tr>
<td>24.3</td>
<td>590.5</td>
<td>4.862</td>
</tr>
<tr>
<td>24.8</td>
<td>615.0</td>
<td>5.125</td>
</tr>
<tr>
<td>25.3</td>
<td>640.1</td>
<td>5.393</td>
</tr>
<tr>
<td>25.8</td>
<td>665.6</td>
<td>5.666</td>
</tr>
<tr>
<td>26.3</td>
<td>691.7</td>
<td>5.945</td>
</tr>
<tr>
<td>26.8</td>
<td>718.2</td>
<td>6.229</td>
</tr>
<tr>
<td>27.3</td>
<td>745.3</td>
<td>6.519</td>
</tr>
<tr>
<td>27.8</td>
<td>772.8</td>
<td>6.813</td>
</tr>
</tbody>
</table>
3.6 SUMMARY

In this chapter, building chiller thermal performance lines based on the building energy thermal performance lines concept in real building is discussed and presented. Through this analysis, the effects of the outdoor environment conditions on the building energy consumption can be investigated. The building configurations (including window to wall ratio and U-value) based on using the thermal performance line model could be categorized. The chiller plant within building normally consumes for about 30-40% total of the building electricity consumption. The outdoor environmental conditions and the degradation level of the components of a chiller will definitely affect the performance of the chiller plant and consequently the energy consumption. With adopting the thermal performance line model for analyzing the energy consumption due to the chiller plant, the characteristics of the electricity consumption of the chiller plant can be represented.

The first section of this chapter discusses the energy signature for building energy performance analysis based on statistical model. A thermal-dynamic model was also studied to identify which is the best climatic variable to establish a thermal performance line model for buildings. Secondly, a literature review on the applications of thermal performance line model using statistical model approach for analyzing the energy performance of air-conditioned building is presented. Thirdly, different typical problems encountered when carrying out the building energy performance analysis are highlighted. In addition, the surveyed results on the availability of the essential building information for building energy consumption analysis are also presented and discussed. Fourthly, an engineering modeling technique for building energy performance analysis for dissection of the building
energy consumption is elaborated. The last section of this chapter delivers the method for analyzing the chiller plant performance, with taking into account chiller component faults, based on building energy signature.
Operating Data of the chiller to reflect its performance in terms of energy is a must.

To assess the performance of the chiller, adequate and valid operating data are also vitally important. Without adequate sensing points or operating data with fault or unhealthy conditions, it is impossible to assess the actual performance of the chiller and finally optimize the chiller performance. This arouses a concept of data mining in order to manage the operating data from BMS as well as to mine the knowledge from a huge amount of operating data sets. In this chapter, the significance of the data logic management protocol is highlighted. After that, the second section of this chapter presents how to establish the data logic management protocol for chiller performance analysis. The last section is to implement the data logic management protocol for existing chiller plants with results.

4.1 SIGNIFICANCE OF DATA LOGIC MANAGEMENT PROTOCOL
The main purposes of developing the data management protocol are to inspect and validate the accuracy and reliability of the raw data measured/received from different sensors via BMS as well as to ensure significant data/sensing points are provided for chiller performance analysis. Raw data means that the data are directly measured from different sensors and transmitted as well as logged to the BMS server without any pre-conditioning or pre-processing. It is ascertained that raw data from sensor measurements are easily corrupted by random errors (e.g. noises or outliers) and systematic errors (e.g. fixed bias) due to the dynamic fluctuations of the system/equipment operations, measurement (sensor) faults, signal transmission faults or BMS computer hardware/software problems etc. [Huang et al. 2009]. Moreover, raw data may be collected during the steady or unsteady state conditions. Obviously, the situation occurs when the chiller is starting to switch-on/off. It may also contain noises or outliers etc. In the meantime, the fulfillment of the thermo-physical laws may also be destroyed. Apart from that, the measured operating data during the normal operation of the system/equipment may be out of the normal operating range. Furthermore, since the raw data are normally measured by different sensors, they may not be synchronized at the same time or logged in the same time interval. As mentioned in Chapter 1, inadequacy of data/sensing points always discovered in existing chiller plants in Hong Kong. This phenomenon causes engineers or researchers difficult to conduct accurate chiller performance assessment.

In general, there are a lot of papers recommended to use some simple statistical techniques in order to remove noise or outliers for uni-variate situations (i.e. single variable) [Walczak et al. 1998; Lin et al. 2007; Kadlec et al. 2009; Monfet et al.
However, those techniques are rather simple and no physical relationship assurance among different operating data set of the chiller is guarantee. Recently, it is not surprising that the data pre-conditioning or pre-processing methods for chiller and even entire HVAC systems have been developed elsewhere [Grimmelius et al. 1995; Cui et al. 2005; Cao et al. 2010; Wang et al. 2010; Monfet et al. 2012]. Nonetheless, the data pre-conditioning methods suggested by those authors just focused on removing out the fault data and making sure the pre-conditioned data set can satisfy with the physical laws only. They all neglected the fundamental thing that ensuring sufficient data sensing points for assessing the chiller performance is critical. No matter how the actual of the data collected, it is impossible to carry out chiller performance analysis without sufficient sensing/data points. In the recent ten decades, a more powerful technique “Principle Component Analysis (PCA)” is a multi-variate approach (i.e. multi-variables) that has been widely adopted to deal with the sensor faults detection and diagnosis. The approach is also adopted for HVAC System including air-side, water-side and refrigeration system [Xiao et al. 2003; Cui et al. 2004; Chen et al. 2010]. Despite how PCA is powerful and sophisticated, it is still difficult to apply it in exiting chiller plant. Lee [2011] also criticized PCA method for existing plant and proposed a simple method for sensor faults detection and diagnosis.

As the above mentioned problems in existing chiller plants and BMS as well as no fully integration of the data pre-conditioning method for actual chiller performance assessment, it is crucial to establish a method for pre-conditioning and proper manage the raw data before carrying out the chiller performance assessment. In this study, data logic management protocol are divided into eight steps (with five stages)
and they include: (a) Building Management System Accountability, (b) Data Acquisitioning, (c) Data Synchronization, (d) Steady-State Conditioning, (e) Thermo-Physical Laws Validity Check, (f) Outlier Conditioning, (g) Range Validity Conditioning and as well as (h) uncertainty analysis on heat rejection rate. The data conditioning process part of the data logic analysis is programmed by adopting a high-level programming language “GNU Octave Version 3.2.3” [Long 2005].

Figure 4.1  Flow Chart of Data Logic Analysis
4.2 ESTABLISHMENT OF DATA LOGIC MANAGEMENT PROTOCOL

4.2.1 Building Management System (BMS) Accountability

Sufficient sensing/data points for measuring the operating parameters/variables of chiller or chiller plant are critical for assessing the actual performance of the system/equipment. It can also truly reflect whether the system/equipment is in healthy condition or not. As reported in Chapter 1, there are a lot of papers discussing the inadequacy and the inaccuracy of sensors or sensing points for monitoring the chiller(s) and the entire chiller plant in Hong Kong. In addition, most of the people concern about the operating parameters of water-side and electrical side for chillers and plant only.

Calder [1994] discussed the issues of monitoring the performance of the chiller system in a practical way. The author suggested how to select the monitoring equipment (i.e. portable measuring instruments and sensors) for chiller plant and the sampling rate for recording/logging the operating data of the plant. In the meantime, the author also classified what parameters are essential or non-essential that should be monitored for chilled water and condenser water loops serving chillers (constant
volume type). The essential parameters include: (a) chilled water flow temperature, (b) chilled water return temperature, (c) chilled water flow rate, (d) compressor running amps, (e) cooling tower fan running amps and (f) condenser pump running amps. In Hong Kong, the EMSD addressed BEC (2012 edition) and specified the requirement of the energy metering for HVAC system [EMSD 2012]. When a chiller, heat pump, unitary air-conditioner or chilled/heated water plant is equal to/greater than 350kW cooling/heating capacity, continuous monitoring facilities for measuring its power [kW] & energy input [kW], cooling/heating power [kW] & energy output [kWh] and $COP$. For chiller water plant, the measurement should include chiller compressors, circulation pumps of condensers or cooling towers, condenser fans, cooling tower fans, radiator fan etc. while chilled water pumps should be excluded. However, this document does not specify which parameters of the chiller or chiller plant should be measured or logged. To summarize the above two documents, the focusing point of monitoring the parameters suggested by Calder [1994] and EMSD [2012] is just to calculate the operating chiller $COP$ and system $COP$.

As mentioned previously in Chapter 2, chiller component fault or degradation can decrease the $COP$ of chiller seriously. Therefore, monitoring the parameters included in calculating the KPI for each major component of a chiller is vitally important.
Instead of monitoring the entire chiller plant, this section suggests the BMS Accountability for assessing the adequacy of sensing points for monitoring the actual performance of each major component of a chiller prior to entire chiller plant. The BMS Accountability regarding chiller KPI summary are shown in Table 4.1.

In conjunction with Chapter 3, 20 buildings selected for conducting the BMS Accountability regarding chiller KPI. There are 12 buildings with air-cooled chiller plant, 3 buildings with water-cooled chiller plant, 1 building with air-cooled radiators, 1 building with air-cooled and water-cooled chiller plant and 3 buildings with air-cooled DX packaged unit (see Table 3.1). It is noted that the sensing points available for each chiller within the same chiller plant were the same. Therefore, assessing the BMS Accountability for one chiller could represent the others within the chiller plant. Since some buildings (without BMS) were also to have log sheets for recording some operating variables of the chiller regularly, the records of the log sheets regarding the recorded operating variables were also treated as BMS sensing points in this analysis.

Table 4.1  Summary of BMS Accountability regarding Chiller KPIs
### Operating Variables

<table>
<thead>
<tr>
<th>Evaporator Overall Heat Transfer Value</th>
<th>Condenser Overall Heat Transfer Value</th>
<th>Electro-Mechanical Loss Efficiency</th>
<th>Compressor Polytropic Efficiency</th>
<th>Refrigerant Mass Flow Rate</th>
<th>Expansive Device Pressure Drop/Flow Characteristic</th>
<th>Coefficient of Performance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Author</td>
<td>Ref A</td>
<td>Author</td>
<td>Ref A</td>
<td>Author</td>
<td>Ref A</td>
<td>Author</td>
</tr>
<tr>
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</tr>
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<td>✓</td>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>✓</td>
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</tr>
<tr>
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</tr>
<tr>
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<td>✓</td>
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<td>-</td>
<td>✓</td>
<td>X</td>
<td>-</td>
</tr>
<tr>
<td>✓</td>
<td>X</td>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>X</td>
<td>✓</td>
</tr>
</tbody>
</table>

### Condenser (Air-Cooled Chiller)

<table>
<thead>
<tr>
<th>Condenser Inlet Ambient Dry-Bulb Temperature</th>
<th>Condenser Outlet Ambient Dry-Bulb Temperature</th>
<th>Condenser Heat Rejection Air Flow Rate</th>
<th>Refrigerant Condensing Temperature</th>
<th>Condenser Heat Rejection Air Fan Power Input</th>
<th>No. of Condenser Heat Rejection Air Fan in Operation</th>
</tr>
</thead>
<tbody>
<tr>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>O</td>
<td>-</td>
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</tr>
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<td>-</td>
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<td>O</td>
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<td>-</td>
<td>-</td>
<td>✓</td>
<td>X</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>X</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>-</td>
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<tr>
<td>-</td>
<td>-</td>
<td>✓</td>
<td>-</td>
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</table>
# Chapter 4: Data Acquisition System in Real Buildings for Individual Chiller Optimization

<table>
<thead>
<tr>
<th><strong>Condenser (Water-Cooled Chiller)</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Condenser Water Outlet Temperature</td>
</tr>
<tr>
<td>Condenser Water Inlet Temperature</td>
</tr>
<tr>
<td>Condensing Water Flow Rate</td>
</tr>
<tr>
<td>Refrigerant Condensing Temperature</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Compressor</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant Discharge Temperature</td>
</tr>
<tr>
<td>Compressor Power Input</td>
</tr>
<tr>
<td>No. of Compressors in Operation</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Building Management System (BMS) Accountability Result</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>✓</td>
</tr>
</tbody>
</table>

**Note:**

- **Essential**: ✓
- **Non-Essential**: O
- **Not Considered**: X
- **Not Related**: -

**Ref A**: Calder 1994
From the survey result revealed that (see Table 4.2), most of the buildings concerned the operating conditions of the water-side system only. For Building B, F, M and S, the measured values of the chilled water supply ($T_{chws}$) & return ($T_{chwr}$) temperature and the condensing water outlet ($T_{cdws}$) & inlet ($T_{cdwr}$) temperature of the chillers were recorded on the log sheets or logged into the BMS servers. For the operating variables of the refrigerant (i.e. refrigerant evaporating temperature – $T_{ev}$, refrigerant condensing temperature – $T_{cd}$ and refrigerant discharge temperature – $T_{dis}$), no record or trend data were provided for most of the buildings except Building S. In addition, it was also observed that no measured operating variables of the chiller for the buildings with air-cooled chiller plant or air-cooled DX package units was recorded or logged. Furthermore, it was also discovered that only 4 buildings measured and logged the operating variables of the compressors of a chiller (i.e. numbers of compressor in operation and power consumption).

As reported by the respective building operators, their interest only focused on the COP of a chiller. Therefore, they did not record or measure the operating variables of refrigerant for the chiller. Instead of measuring the power consumption of air-cooled condenser fans and chiller compressors, they normally monitor the power consumption of the entire chiller plant via the sub-meters. For the buildings having
air-cooled DX package units provided, it was surprising that the building operators even did not interested in \textit{COP} of each chiller. Their concern was only focusing on whether the cooling capacity of the units is capable to deal with the cooling load of the building. As a result, once the fault(s) of a chiller occurred, the building operators normally relied on the fault alarm generated by the local control panel of a chiller to repair the chiller. In order words, most of the surveyed buildings relied on corrective maintenance while no conditioned-based maintenance approach was adopted with respect to individual chiller. This phenomenon can also be reflected on Table 4.3. Only Building S could have adequate sensing points and logged operating data for assessing the key performance indices of a chiller.

Therefore, it is suggested to the building operators to record the essential operating variables of each chiller or DX units regularly with optimal time interval of 30 minutes or 60 minutes per each set of operating data in order to monitor the key performance indices of each chiller so that down time of a chiller due to adverse component fault(s) can be minimized and even avoided. It is also noted that this suggestion may have cost implication since additional sensing devices and associated accessories (e.g. cable etc.) may be required if no existing sensing point is provided. Regarding the refrigerant operating data of a chiller, the chiller vendor or contractor
may have the logged data via the local control panel of a chiller privately and they normally keep record during the regular inspection for the chiller. These data will normally not be opened to public. Due to this reason, it is also suggested that the building operators can discuss with the chiller vendor or contractor to obtain these data for chiller performance analysis.
### Table 4.2  Summary of BMS (Sensing Points) Accountability regarding Chiller KPIs for 20 Surveyed Buildings

| Building | A | B | C | D | E | F | G | H | I | J | K | L | M | B | O | P | Q | R | S | T | Total |
| **Evaporator** | | | | | | | | | | | | | | | | | | | | | | | | | | |
| $T_{chws}$ | X | ✓ | X | X | - | ✓ | X | X | X | - | - | X | ✓ | X | ✓ | - | X | ✓ | ✓ | ✓ | X | 37.5% |
| $T_{chwr}$ | X | ✓ | X | X | - | ✓ | X | X | X | - | - | X | X | X | X | - | X | ✓ | ✓ | ✓ | X | 37.5% |
| $m_w$ | X | ✓ | X | X | - | X | X | X | X | - | - | X | X | X | X | - | X | ✓ | X | X | 12.5% |
| $T_{ev}$ | X | X | X | X | - | X | X | X | X | X | X | - | X | X | X | X | - | X | ✓ | X | 5.6% |
| **Condenser (Air-Cooled Chiller)** | | | | | | | | | | | | | | | | | | | | | | | | | | |
| $T_{adbi}$ | X | X | X | X | - | X | X | X | X | X | X | - | - | X | - | - | X | 0.0% |
| $T_{adbe}$ | X | X | X | X | - | X | X | X | X | X | X | - | - | X | - | - | X | 0.0% |
| $m_{a}$ | X | X | X | X | - | X | X | X | X | X | X | - | - | X | - | - | X | 0.0% |
| $T_{cd}$ | X | X | X | X | - | X | X | X | X | X | X | - | - | X | - | - | X | 0.0% |
| $P_{cf}$ | X | X | X | X | - | X | X | X | X | X | X | - | - | X | - | - | X | 0.0% |
| $N_{cf}$ | X | X | X | X | - | X | X | X | X | X | X | - | - | X | - | - | X | 0.0% |
| **Condenser (Water-Cooled Chiller)** | | | | | | | | | | | | | | | | | | | | | | | | | | |
| $T_{adws}$ | - | ✓ | - | - | - | - | - | - | - | - | - | - | - | - | - | ✓ | - | ✓ | ✓ | - | 100.0% |
| $T_{adwr}$ | - | ✓ | - | - | - | - | - | - | - | - | - | - | - | - | - | ✓ | - | ✓ | ✓ | - | 100.0% |
| $m_{chw}$ | - | X | - | - | - | - | - | - | - | - | - | - | - | - | - | - | X | - | X | ✓ | - | 25.0% |
| $T_{cd}$ | - | X | - | - | - | - | - | - | - | - | - | - | - | - | - | - | X | - | X | ✓ | - | 25.0% |
### Data Acquisition System in Real Buildings for Individual Chiller Optimization

| Building | A | B | C | D | E | F | G | H | I | J | K | L | M | B | O | P | Q | R | S | T | Total |
| **Compressor** | | | | | | | | | | | | | | | | | | | | | |
| $T_{\text{dis}}$ | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |
| $P_{cc}$ | X | ✓ | ✓ | ✓ | - | X | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 22.2% |
| $N_{cc}$ | X | ✓ | ✓ | ✓ | - | X | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 22.2% |

**Note:**

- Accountable ✓
- Not Accountable X
### Table 4.3  Summary of BMS (KPIs) Accountability regarding Chiller KPIs for 20 Surveyed Buildings

| Building | A | B | C | D | E | F | G | H | I | J | K | L | M | B | O | P | Q | R | S | T | Total |
| UAev     | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |
| UAcd     | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |
| η_m     | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |
| η_poly  | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |
| m_r     | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |
| C_{o}/A_o | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |
| COP      | X | X | X | X | - | X | X | X | X | X | X | X | X | X | X | X | - | X | X | ✓ | X | 5.6% |

**Note:**

- Accountable  ✓
- Not Accountable  X
4.2.2 Data Acquisitioning

There are different formats/structures of data acquisitioning system provided by different control vendors (e.g. Siemens, Honeywell or Johnson Controls etc.). A numbers of channels are normally provided for monitoring various types of system parameters. The DC signal from different measuring devices/sensors can be scaled into their real physical values of the measured parameters by using a logging and control supervisory programme (i.e. direct digital controllers and measuring devices/sensors) which is commonly integrated into the BMS [Li 2007]. Therefore, the measured value of the parameters can be monitored and logged via the BMS. To obtain the logged data, a Microsoft Excel format is commonly used in most of the BMS. Apart from the formats of generating the logged data, time interval setting is also important in the data acquisitioning part since it will affect the performance interruption of the system. In general, 0.5 to 1.0 hour is commonly adopted in most of the chiller plants.
4.2.3 Data Synchronization

Data Synchronization is the process that develops consistency among dataset from the source to the storage location and then vice versa. Also, it generates the continuous harmonization of the data over time. In existing chiller plant with its BMS, it consists a lot of sensors to monitor different operating parameters of the equipment or system of the chiller plant. The sensors monitor the operating variables and convert them into electrical signals (include analogue, digital or both). After that, these signals will be transferred to the BMS station (i.e. server) via the direct digital control (DDC). The time interval settings, carried out in DDC or BMS station, for logging different operating variables of each of the equipment or system may not be the same. Moreover, sensors for each operating data with errors may incur data lost or log the data with time lag. Therefore, the time stamp of each signal (i.e. operating variable) may be inconsistent. For instance, the time interval setting of the sensors for monitoring and logged the chilled water supply temperature, chilled water return temperature and chilled water flow rate are the same. Unfortunately, the time stamp of some of these three operating parameters may not be the same (see Figure 4.2). Therefore, it is difficult to carry out the analysis (e.g. cooling load) due to the time of a set of data are inconsistent. As a result, data synchronization before assessing the energy performance of the chiller or chiller plant is a must.
In general, the three typical methods for data synchronization and they are (a) Intersection Method, (b) Cubic Interpolation and (c) Linear Interpolation. These three methods were selected for data synchronization study. Intersection Method is to extract the common data sets that exist in all relevant data set. In data synchronization, this method extracts common data set from time data and uses it as the base of extraction. In other words, it is simply put all data in a big pool regardless whether the time is synchronized or not. Cubic Interpolation is a form of interpolation where the interpolant is a piecewise polynomial. Linear Interpolation is a method of curve fitting using linear polynomials.
4.2.4 Steady-State Conditioning

The operation of a chiller or chiller plant may involve steady or dynamic state in nature. When a chiller or chiller plant is not steady, the thermodynamic operating variables of it are highly unstable (see Figure 4.3). For chiller performance analysis, steady-state data is adequate since the operations of dynamic situation are not interested. Basically, there are two approaches for steady-state conditioning: (a) Data Removal for a Period with Rationale and (b) Steady-State Conditioner.

<table>
<thead>
<tr>
<th>Date/Time</th>
<th>Chiller 1 - CHW Supply Water Temp</th>
<th>Date/Time</th>
<th>Chiller 1 - CHW Supply Water Flow</th>
<th>Date/Time</th>
<th>Chiller 1 - CHW Supply Water Temp</th>
<th>Date/Time</th>
<th>Chiller 1 - CHW Supply Water Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>5/2/2008 5:30</td>
<td>14.06°C</td>
<td>5/2/2008 5:30</td>
<td>0</td>
<td>5/2/2008 5:30</td>
<td>16.14°C</td>
<td>5/2/2008 5:30</td>
<td>0</td>
</tr>
<tr>
<td>5/2/2008 6:00</td>
<td>14.06°C</td>
<td>5/2/2008 6:00</td>
<td>0</td>
<td>5/2/2008 6:00</td>
<td>16.14°C</td>
<td>5/2/2008 6:00</td>
<td>0</td>
</tr>
<tr>
<td>5/2/2008 7:00</td>
<td>14.06°C</td>
<td>5/2/2008 7:00</td>
<td>0</td>
<td>5/2/2008 7:00</td>
<td>16.14°C</td>
<td>5/2/2008 7:00</td>
<td>0</td>
</tr>
<tr>
<td>5/2/2008 7:30</td>
<td>14.06°C</td>
<td>5/2/2008 7:30</td>
<td>0</td>
<td>5/2/2008 7:30</td>
<td>16.14°C</td>
<td>5/2/2008 7:30</td>
<td>0</td>
</tr>
<tr>
<td>5/2/2008 8:00</td>
<td>14.06°C</td>
<td>5/2/2008 8:00</td>
<td>0</td>
<td>5/2/2008 8:00</td>
<td>16.14°C</td>
<td>5/2/2008 8:00</td>
<td>0</td>
</tr>
<tr>
<td>5/2/2008 8:30</td>
<td>14.06°C</td>
<td>5/2/2008 8:30</td>
<td>0</td>
<td>5/2/2008 8:30</td>
<td>16.14°C</td>
<td>5/2/2008 8:30</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure 4.3 Example of Trend Log Data for a Chiller with Transient Condition
4.2.4.1 Data Removal for a Period with Rationale

This data removal is to avoid the dynamic data caused by on/off situations of the chiller operation. In most of literatures, the data within one hour after/before on/off operation of the chiller system are normally excluded [Cui et al. 2005; Yu et al. 2008]. As a result, this criterion is applied.

4.2.4.2 Steady-State Conditioner

There are two types of steady-state filters commonly used and they are: (a) Moving Window Average and (b) Exponentially Weighted Moving Average. For moving window average, it uses the calculation of the standard deviation of parameters in a recursive fashion by averaging over samples in time with a fixed time window length [Cao et al. 1995; Jaing et al. 2003; Yu et al. 2008]. Zhou [2009] did a research on a systematic fault diagnosis strategy for building and HVAC systems, the author adopted the fixed time-window length with a four-interval in order to smooth the data (i.e. minimizing the dynamic data and removing the noisy data). For exponentially weighted moving average, the concept of this technique is very similar to the moving window average. However, the main difference between them is that the former takes into account the weighting factor. In the meantime, there is no absolute rule to set this factor for chiller system. This makes the technique too
complicated to use and even vague. Therefore, moving window average with a
four-interval window length is adopted.

Assuming that at any time instant $k$, the average of the latest $n$ samples of a data
sequence, $x_i$, is given by:

$$\bar{x}_k = \frac{1}{n} \sum_{i=k-n+1}^{k} x_i$$

(4.1)

A difference between two averages of the latest $n$ samples at the current time, $k$ and
at the previous time instant, $k-1$ is:

$$\bar{x}_k - \bar{x}_{k-1} = \frac{1}{n} \left[ \sum_{i=k-n+1}^{k} x_i - \sum_{i=k-n}^{k-1} x_i \right] = \frac{1}{n} (x_k - x_{k-n})$$

(4.2)

By rearranging, the current average is determined by:

$$\bar{x}_k - \bar{x}_{k-1} = \frac{1}{n} (x_k - x_{k-n})$$

(4.3)
4.2.5 Thermo-Physical Validity Check

As discussed in Chapter 2, the interaction among components of a chiller follows the thermodynamic behaviour (i.e. energy and mass balance conservation). Without fault, the operating variables of a chiller can obey the thermo-physical laws under normal operation. Therefore, using the thermo-physical laws with the operating data set to validate the whether the chiller is operating under normal or abnormal conditions is applicable. Thus, the Thermo-Physical Validity check is to guarantee the correlation among the operating variables which can satisfy with the universal thermo-physical laws. The Thermo-Physical Validity Check involves two processes and they are (a) Variable Check and (b) Energy Balancing Check.
Figure 4.5  Energy Transfer Paths within Chiller Unit at Typical Design Condition

- Refrigerant Discharge Temperature = 50.0°C
- Refrigerant Condensing Temperature = 45.0°C
- Condensing Water Outlet Temperature = 37.0°C
- Outdoor Air Temperature = 33.0°C (DB)/28.0°C (WB)
- Condensing Water Inlet Temperature = 32.0°C
- Indoor Air Temperature = 25.5°C
- Chilled Water Return Temperature = 12.0°C
- Refrigerant Evaporating Temperature = 4.5°C
- Chilled Water Supply Temperature = 7.0°C

Condenser-Side Heat Transfer Path
Evaporator-Side Heat Transfer Path
4.2.4.3 Variables Check

Variables check employs the fundamental of thermo-physical concepts to validate the raw data (simple data – i.e. the operating data of the chiller without any calculation required). It is accepted that the variations of each measurement among different variables (i.e. operating parameters) should be subject to thermo-physical laws. By using this judgment, the variables check can be implemented.

<table>
<thead>
<tr>
<th>Variable Type</th>
<th>Variable Nature</th>
<th>Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>Refrigerant, Water</td>
<td>$0 &lt; T_{ev} &lt; T_{chw}s &lt; T_{chwr} &lt; T_{chw} &lt; T_{cd} &lt; T_{dis}$</td>
</tr>
<tr>
<td>Pressure</td>
<td>Refrigerant</td>
<td>$0 &lt; P_{ev} &lt; P_{cd}$</td>
</tr>
<tr>
<td>Chiller Power Input</td>
<td>Electricity</td>
<td>$0 &lt; P_{cc}$</td>
</tr>
</tbody>
</table>

4.2.4.4 Energy Balancing Check

It is based on the conservation of energy balancing within refrigeration cycle to validate a set of raw data (derived data – i.e. the operation data of the chiller calculated from individual raw data set) during normal operation of a chiller. It is subject to the universal concept of “energy input equals to energy output” to implement.
Table 4.5  Thermal-Physical Law Validity Check (Energy Balancing Check) for Chillers

<table>
<thead>
<tr>
<th>Principle</th>
<th>Variable Nature</th>
<th>Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Balance</td>
<td>Cooling Load, Chiller</td>
<td>$q_{cd} = q_{rl} + E_{cc}$</td>
</tr>
<tr>
<td></td>
<td>Compressor Power Input,</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Heat Rejection Rate</td>
<td></td>
</tr>
</tbody>
</table>

4.2.6 Outlier Check

Outliers are commonly defined as observations that are not consistent with the majority of the data [Lin et al. 2007], including missing data points or blocks and observations that deviate significantly from normal values. The famous method adopted in dealing with the outlier removal is the rule of three standard deviations away from the mean over a series of time [Cui et al. 2005; Wang et al. 2010; Lin et al. 2007; Ratchliff 1993; Pearson 2002; Xiao et al. 2009; Wang et al. 2005; Kadlec et al. 2009]. Unfortunately, this method often fails in practice because the presence of outliers tends to inflate the variance estimation and incurring too few outliers to be detected. The Hampel Identifier [Davies et al. 1981] replaces the outliers-sensitive means and standard deviation estimates with the outlier-resistant median and median absolute deviation from the median. Due to this reason, Hampel identifier was adopted in the study.

$$\text{MAD} = 1.4826 \, \text{median}(|x_i - x^*|)$$  (4.4)
where $x$ is the median of the data sequence. The factor 1.4826 is chosen such that the expected $MAD$ is equal to the standard deviation $\sigma$ for normally distributed data.

### 4.2.7 Range Validity Conditioning

The main concept of the range validity conditioning is to inspect the variations of each measurement variable during the chiller operation by engineering and statistical judgment. An expert knowledge is required. In case the magnitude of the measurement is out of the normal operating range, the measured variable or data may be treated as suspiciously abnormal and it will be rejected. In this conditioning, it embraces two parts: (a) Engineering Approach; and (b) Statistical Approach. A brief description is summarized in Table 4.6:

<table>
<thead>
<tr>
<th>Approach</th>
<th>Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engineering Approach</td>
<td>Based on the possibility of the operating range of the measured variables.</td>
</tr>
<tr>
<td>Statistical Approach</td>
<td>Based on the statistical technique to investigate the operating range of the measured variables (mean; standard deviation; and frequency distribution plot).</td>
</tr>
</tbody>
</table>
4.2.8 Uncertainty Analysis on Heat Rejection Rate of Chillers

In reality, uncontrollable heat loss/gain from/to skin of a chiller and sensor fault(s) will affect the accuracy of the prediction of the amount of the heat rejection rate of a chiller. Apart from that, sensor error will also contribute the measurement error for calculating the cooling load as well as heat rejection rate of a chiller. Therefore, it is essential to carry out the uncertainty analysis in order to check how the accuracy of the operating variables is and whether they can be used for analyzing chiller performance.

With using the concept of energy balancing between cooling load \( q_{rl} \) and heat rejection rate \( q_{cd} \) as well as power input \( E_{cc} \) of a chiller, the heat rejection rate \( q_{cd} \) can be calculated as follow:

\[
q_{cd} = q_{rl} + E_{cc} \tag{4.5}
\]

After that, the residual heat rejection rate can be calculated as:

\[
R_h = q_{cd} - (q_{rl} + E_{cc}) \tag{4.6}
\]

where \( R_h \) is a residual heat rejection rate [kW].
With using residual heat rejection rate uncertainly analysis, its uncertainty limit (i.e. threshold) at a specific time can then be evaluated. If there is no any error occurred, Equation (4.6) should be zero. Based on Equation (2.11) and (2.16), the residual heat rejection rate can be further expressed as:

\[ R_h = (m_{cdw} C_{pw} (T_{chws} - T_{cdwr})) - ((m_w C_{pw} (T_{chwr} - T_{chws})) + E_{cc}) \] (4.7)

The uncertainties (\( \delta R \)) in an estimation (\( R \)) made from measurement (\( V_i \)) that are themselves subject to uncertainties (\( \delta V_i \)) can be quantified by using Kline and McClintock’s Method [Yik and Chiu 1998]. The formula for implementing the uncertainty analysis is shown as below:

\[ R = f(V_1, V_2, \ldots, V_n) \] (4.8)

\[ \delta R = \sqrt{\sum_{i=1}^{n} \left( \frac{\partial R}{\partial V_i} \delta V_i \right)^2} \] (4.9)

where \( R \) is an estimate made from variable \( V_1 \) to \( V_n \); \( \delta R \) is uncertainty in the estimate \( R \).

With substituting Equation (4.7) into (4.9), the uncertainty in the residual heat rejection rate can be evaluated by using the following equation:
\[ \delta R_h = (\left( C_{pw} (T_{cdws} - T_{cdwr}) \left( m_{cdw} \delta m_{cdw} \right) / 100 \right)^2 + (m_{cdw} C_{pw} \delta T_{cdws})^2 + (m_{cdw} C_{pw} \delta T_{cdwr})^2 \]

\[ + (C_{pw} (T_{chws} - T_{chwr}) \left( m_w \delta m_w \right) / 100)^2 + (m_w C_{pw} \delta T_{chws})^2 + (m_w C_{pw} \delta T_{chwr})^2 \]

\[ + (E_{cc} \delta E_{cc})^2 \right)^{0.5} \quad (4.10) \]
4.3 SELECTED EXISTING CHILLER PLANTS

DESCRIPTION

Two separated chiller plants serving for institutional building (named as Chiller Plant A) and commercial building (named as Chiller Plant B) were selected for the study. These two chiller plants have adequate sensing points which the key performance indices of each chiller can be assessed. The general information of these two chiller plants are summarized as below:

<table>
<thead>
<tr>
<th></th>
<th>CH-01 to CH-04</th>
<th>CH-05</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number</td>
<td>4 (Duty)</td>
<td>1 (Light Mode Chiller)</td>
</tr>
<tr>
<td>Compressor Type</td>
<td>Centrifugal</td>
<td>Screw</td>
</tr>
<tr>
<td>(Open Type)</td>
<td></td>
<td>(Semi-Hermetic Type)</td>
</tr>
<tr>
<td>Compression Stage Number</td>
<td>Single</td>
<td>Single</td>
</tr>
<tr>
<td>Heat Rejection Method</td>
<td>Water-Cooled</td>
<td>Water-Cooled</td>
</tr>
<tr>
<td>Refrigerant Type</td>
<td>R134a</td>
<td>R134a</td>
</tr>
<tr>
<td>Capacity Control Method</td>
<td>Inlet-Guide Vane</td>
<td>Sliding Valve</td>
</tr>
<tr>
<td>Chilled Water Flow Type</td>
<td>Constant Flow</td>
<td>Constant Flow</td>
</tr>
<tr>
<td>Condensing Water Flow Type</td>
<td>Constant Flow</td>
<td>Constant Flow</td>
</tr>
<tr>
<td>Rated Cooling Capacity (kW)</td>
<td>1513</td>
<td>704</td>
</tr>
<tr>
<td>Rated Compressor Power (kW)</td>
<td>300</td>
<td>140</td>
</tr>
<tr>
<td>Chiller Efficiency at Full Load (kW/TR)</td>
<td>0.502</td>
<td>0.482</td>
</tr>
<tr>
<td>Heat Rejection Capacity (kW)</td>
<td>2090</td>
<td>990</td>
</tr>
<tr>
<td>Design Chilled Water Supply/Return Temperature (°C)</td>
<td>7.0/12.0</td>
<td>7.0/12.0</td>
</tr>
<tr>
<td>Design Condenser Water Inlet/Outlet Temperature (°C)</td>
<td>33.0/38.0</td>
<td>33.0/38.0</td>
</tr>
<tr>
<td>Design Chilled Water Flow Rate (L/s)</td>
<td>72.0</td>
<td>33.0</td>
</tr>
<tr>
<td>Design Condensing Water Flow Rate (L/s)</td>
<td>87.0</td>
<td>72.0</td>
</tr>
</tbody>
</table>
### Table 4.8  General Information of Chiller Plant B

<table>
<thead>
<tr>
<th></th>
<th>CH-01 to CH-04</th>
<th>CH-05</th>
<th>CH-06 to CH-07</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number</td>
<td>4 (Duty)</td>
<td>4 (Standby/Night)</td>
<td>2 (Essential)</td>
</tr>
<tr>
<td>Compressor Type</td>
<td>Centrifugal</td>
<td>Centrifugal</td>
<td>Centrifugal</td>
</tr>
<tr>
<td>(Open Type)</td>
<td>(Open Type)</td>
<td>(Open Type)</td>
<td>(Open Type)</td>
</tr>
<tr>
<td>Compression Stage Number</td>
<td>Single</td>
<td>Single</td>
<td>Single</td>
</tr>
<tr>
<td>Heat Rejection Method</td>
<td>Water-Cooled</td>
<td>Water-Cooled</td>
<td>Water-Cooled</td>
</tr>
<tr>
<td>Refrigerant Type</td>
<td>R134a</td>
<td>R134a</td>
<td>R134a</td>
</tr>
<tr>
<td>Capacity Control Method</td>
<td>Inlet-Guide Vane</td>
<td>Inlet-Guide Vane</td>
<td>Inlet-Guide Vane</td>
</tr>
<tr>
<td>Chilled Water Flow Type</td>
<td>Variable Flow</td>
<td>Variable Flow</td>
<td>Variable Flow</td>
</tr>
<tr>
<td>Condensing Water Flow Type</td>
<td>Variable Flow</td>
<td>Variable Flow</td>
<td>Variable Flow</td>
</tr>
<tr>
<td>Rated Cooling Capacity (kW)</td>
<td>6336</td>
<td>3625</td>
<td>1830</td>
</tr>
<tr>
<td>Rated Compressor Power (kW)</td>
<td>1161</td>
<td>659</td>
<td>358</td>
</tr>
<tr>
<td>Chiller Efficiency at Full Load (kW/TR)</td>
<td>0.65</td>
<td>0.65</td>
<td>0.65</td>
</tr>
<tr>
<td>Heat Rejection Capacity (kW)</td>
<td>7920</td>
<td>4530</td>
<td>2288</td>
</tr>
<tr>
<td>Design Chilled Water Supply/Return Temperature (°C)</td>
<td>7.0/14.5</td>
<td>7.0/14.5</td>
<td>6.0/11.5</td>
</tr>
<tr>
<td>Design Condenser Water Inlet/Outlet Temperature (°C)</td>
<td>32.0/39.0</td>
<td>32.0/39.0</td>
<td>32.0/39.0</td>
</tr>
<tr>
<td>Design Chilled Water Flow Rate (L/s)</td>
<td>202.1</td>
<td>115.6</td>
<td>79.6</td>
</tr>
<tr>
<td>Design Condensing Water Flow Rate (L/s)</td>
<td>270.7</td>
<td>148.7</td>
<td>78.2</td>
</tr>
</tbody>
</table>

### Table 4.9  Specification of the Required Sensors for Chiller Plant A and Chiller Plant B

<table>
<thead>
<tr>
<th>Type</th>
<th>Type</th>
<th>Measured Variable</th>
<th>Unit</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water Temperature</td>
<td>Platinum Resistance Type (PT100)</td>
<td>$T_{\text{chow}}, T_{\text{chwr}}, T_{\text{cdws}}, T_{\text{cdwr}}$</td>
<td>°C</td>
<td>(a) Uncertainty of $\pm 0.15$°C (at 0°C) (b) Uncertainty of $\pm 0.35$°C (at 100°C)</td>
</tr>
<tr>
<td>Refrigerant Temperature Sensor</td>
<td>Ditto</td>
<td>$T_{\text{ev}}, T_{\text{cd}}, T_{\text{dis}}$</td>
<td>°C</td>
<td>(a) Uncertainty of $\pm 0.15$°C (at 0°C) (b) Uncertainty of $\pm 0.26$°C (at 55°C)</td>
</tr>
<tr>
<td>Refrigerant Pressure Transmitter</td>
<td>Ditto</td>
<td>$P_{\text{ev}}, P_{\text{cd}}$</td>
<td>kPa</td>
<td>Uncertainty of $\pm 0.4$% of Full Scale</td>
</tr>
<tr>
<td>Water Flow Meter</td>
<td>Magnetic Flow Meter</td>
<td>$m_{\text{ch}}, m_{\text{cdw}}$</td>
<td>L/s</td>
<td>Uncertainty of $\pm 1.0$% of Full Scale</td>
</tr>
<tr>
<td>Power Meter</td>
<td>Kilowatt Meter</td>
<td>$P_{\text{cc}}$</td>
<td>kW</td>
<td>Uncertainty of $\pm 0.5$% of Full Scale</td>
</tr>
</tbody>
</table>

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4.4 IMPLEMENTATION OF DATA LOGIC MANAGEMENT PROTOCOL

In order to test the performance of the data logic management protocol regarding the data pre-conditioning, a statistical approach for the comparison of the operating data sets of each chiller before and after applying the data logic management protocol was applied. It includes: (a) mean, (b) medium, (c) maximum, (d) minimum and (e) standard deviation as well as (f) frequency distribution. In addition, a graphical presentation was also utilized for visualization purpose.

4.4.1 Stage 1 (Data Acquisition)

There were 13 sensing points for monitoring different variables for each chiller (see Table 4.9) for both selected chiller plants. These measured variables could also be logged/read in/via BMS. For time interval setting of each chiller plant, 0.5 hour for each chiller was set in Chiller Plant A while 1.0 was set in Chiller Plant B. For both chiller plants, the file of the logged data could be generated in DBF format. Therefore, transformation of the file format from DBF to Microsoft Excel CSV was required.
4.4.2 Stage 2 (Data Synchronization)

There are three data synchronization methods considered and the results for these three methods were presented (including union intersection, cubic interpolation and linear interpolation) in this section. A simple statistical method based on the lowest standard deviation, for selecting the data synchronization method for data logic analysis, is selected. The reason behind is that the lowest deviation indicates a smaller dispersion/variation of the data away from the mean (i.e. the data points tend to be very close to the mean). The results presented in this part are based on the period from 01-05-2008 to 01-01-2009 (for Chiller Plant A) which covers a half year operating data (including summer, autumn and winter season) for each chiller. In the meantime, the results presented in the following tables are based on the raw data with implementing the data synchronization process only. Table 4.8 to 4.12 summarize the statistical results for CH-01 to CH-05 for Chiller Plant A respectively. Since the data acquisition of the BMS for logging the chiller operating trend data of Chiller Plant B has the data synchronization function, there is no result presented in stage 2. In addition, for logging the operating data of each chiller into the BMS for both chiller plants, the logging data time interval for Chiller Plant A is 30 minutes while Chiller Plant B is 60 minutes.
As the results shown in Table 4.10 to 4.14, it was observed that the mean and standard deviation of the data synchronization results with using Cubic and Linear Interpolation method for each chiller for Chiller Plant A were more or less the same. In the meantime, when comparing to the results of standard deviation for each chiller, using Cubic/Linear Interpolation method was lower than Union Intersection method. It implies that the data point dispersion (compared to mean value) with using Cubic/Linear Interpolation method for data synchronization was smaller than using Union Intersection method. Moreover, in order to make the process more simpler, Linear Interpolation method was selected for data synchronization in this data logic management protocol.
Table 4.10 Summary of Statistical Results of Data Synchronization for CH-01 (Chiller Plant A)

<table>
<thead>
<tr>
<th>Input</th>
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<th>Standard Deviation</th>
</tr>
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<tr>
<td></td>
<td>Union</td>
<td>Cubic</td>
</tr>
<tr>
<td>m_w</td>
<td>37.456</td>
<td>37.284</td>
</tr>
<tr>
<td>T_chwr</td>
<td>12.713</td>
<td>12.710</td>
</tr>
<tr>
<td>T_chws</td>
<td>12.121</td>
<td>12.135</td>
</tr>
<tr>
<td>m_cdw</td>
<td>87.000</td>
<td>87.000</td>
</tr>
<tr>
<td>T_cdw</td>
<td>27.688</td>
<td>27.661</td>
</tr>
<tr>
<td>T_cdw</td>
<td>38.718</td>
<td>30.679</td>
</tr>
<tr>
<td>T_cd</td>
<td>22.890</td>
<td>22.849</td>
</tr>
<tr>
<td>E_cc</td>
<td>111.290</td>
<td>110.758</td>
</tr>
<tr>
<td>T_dis</td>
<td>35.323</td>
<td>35.277</td>
</tr>
<tr>
<td>P_ev</td>
<td>339.785</td>
<td>339.912</td>
</tr>
<tr>
<td>P_cd</td>
<td>547.561</td>
<td>546.765</td>
</tr>
</tbody>
</table>
### Table 4.11  Summary of Statistical Results of Data Synchronization for CH-02 (Chiller Plant A)

<table>
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<th>Standard Deviation</th>
</tr>
</thead>
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<td>Union</td>
<td>Cubic</td>
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<td>$T_{chwr}$</td>
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<td>14.777</td>
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<td>-16.044</td>
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<tr>
<td>$T_{ev}$</td>
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<td>16.581</td>
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<td>$m_{cdw}$</td>
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<td>$T_{cdws}$</td>
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<td>$E_{cc}$</td>
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<td>$T_{dis}$</td>
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<td>$P_{ev}$</td>
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<td>419.663</td>
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<tr>
<td>$P_{cd}$</td>
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Table 4.12  Summary of Statistical Results of Data Synchronization for CH-03 (Chiller Plant A)

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<th>Standard Deviation</th>
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<td>Cubic</td>
</tr>
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<td>$m_{w}$</td>
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<td>43.678</td>
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<td>$T_{chwr}$</td>
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<td>13.294</td>
</tr>
<tr>
<td>$T_{ev}$</td>
<td>10.676</td>
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<tr>
<td>$m_{cdw}$</td>
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<td>87.000</td>
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<td>$T_{cdwr}$</td>
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<td>31.192</td>
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<tr>
<td>$E_{cc}$</td>
<td>129.217</td>
<td>129.217</td>
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<tr>
<td>$T_{dis}$</td>
<td>42.977</td>
<td>42.977</td>
</tr>
<tr>
<td>$P_{ev}$</td>
<td>327.662</td>
<td>327.662</td>
</tr>
<tr>
<td>$P_{cd}$</td>
<td>566.347</td>
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</tbody>
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### Table 4.13 Summary of Statistical Results of Data Synchronization for CH-04 (Chiller Plant A)

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<td>Cubic</td>
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<td>$T_{chws}$</td>
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<td>12.158</td>
</tr>
<tr>
<td>$T_{ev}$</td>
<td>11.682</td>
<td>11.695</td>
</tr>
<tr>
<td>$m_{cdw}$</td>
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<td>87.000</td>
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<td>$T_{cdwr}$</td>
<td>30.089</td>
<td>30.090</td>
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<td>$T_{cdws}$</td>
<td>32.070</td>
<td>32.061</td>
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<td>40.380</td>
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<tr>
<td>$P_{ev}$</td>
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<td>342.179</td>
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<td>$P_{cd}$</td>
<td>556.963</td>
<td>556.091</td>
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<td>Mean</td>
<td>Standard Deviation</td>
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<td>Union</td>
<td>Cubic</td>
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<tr>
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<td>14.676</td>
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<td>$T_{chws}$</td>
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<td>13.724</td>
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<td>15.021</td>
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<td>72.000</td>
<td>72.000</td>
</tr>
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<td>$T_{cdws}$</td>
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<td>31.852</td>
</tr>
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<td>40.880</td>
</tr>
<tr>
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<td>395.718</td>
<td>395.110</td>
</tr>
<tr>
<td>$P_{cd}$</td>
<td>570.309</td>
<td>570.990</td>
</tr>
</tbody>
</table>
4.4.3 Stage 3 (Data Conditioning)

After implementing the data conditioning process, the statistical results and the trends of the operating variables of five chillers for Chiller Plant A and seven chillers for Chiller Plant B are generated. Table 4.15 (Chiller Plant A) and 4.16 (Chiller Plant B) show the statistical results of each operating variable, before and after conducting the data conditioning process, of five selected chillers. The plots of the trends of the operating variable of CH-01 for Chiller Plant A are presented in this section (see Figure 4.6 (a) to (x)). The trends of the operating variables of CH-01 to CH-05 (before and after data conditioning process) for Chiller Plant A and CH-01 to CH-07 (before and after data conditioning process) for Chiller Plant B are illustrated in Appendix C.

As Table 4.15 and 4.16 show that the standard deviation of each operating variable of the chillers after implementing data conditioning process for both chiller plants were lower when comparing to the operating variables of the chillers before implementing the data conditioning process. Apart from that, it was also easily identified that the fluctuations of each variable of the chillers for both chiller plants were small when compared with the operating variables before the data conditioning process. These results indicated that each operating variable of the chillers for both
chiller plants after the data conditioning process are more accurate for chiller performance assessment since the unwanted variations due to the measurement errors etc. have been minimized (i.e. the trend of the curve of each operating data of chiller for these two chiller plants after the data conditioning process was smoothen when compared with the operating data set before data conditioning process). It is noted that no value of the standard deviation of the condensing water flow rate of each chiller for Chiller Plant A is indicated since there was no sensing point (i.e. sensor) provided for monitoring and measuring the operating variable of condensing water flow rate. In addition, the mean values of each operating variable of the chillers were close to their rated values after implementing the data conditioning process.
### Table 4.15 Summary of Statistical Results of Data Conditioning Process of Each Chiller (Chiller Plant A)

<table>
<thead>
<tr>
<th>Chiller</th>
<th>$T_{chws}$</th>
<th>$T_{chwr}$</th>
<th>$T_{cdws}$</th>
<th>$T_{cdwr}$</th>
<th>$T_{ev}$</th>
<th>$P_{ev}$</th>
<th>$T_{cd}$</th>
<th>$P_{cd}$</th>
<th>$T_{dis}$</th>
<th>$m_s$</th>
<th>$m_{abs}$</th>
<th>$P_{ce}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH-01</td>
<td>Mean</td>
<td>8.565</td>
<td>12.278</td>
<td>29.795</td>
<td>6.215</td>
<td>263.908</td>
<td>35.783</td>
<td>807.018</td>
<td>48.056</td>
<td>71.2</td>
<td>87.000</td>
<td>220.608</td>
</tr>
<tr>
<td></td>
<td>(12.120)</td>
<td>(12.713)</td>
<td>(30.723)</td>
<td>(27.696)</td>
<td>(11.568)</td>
<td>(339.739)</td>
<td>(22.881)</td>
<td>(547.400)</td>
<td>(35.322)</td>
<td>(37.428)</td>
<td>(-)</td>
<td>(111.185)</td>
</tr>
<tr>
<td></td>
<td>Standard</td>
<td>0.377</td>
<td>0.621</td>
<td>0.815</td>
<td>0.470</td>
<td>5.971</td>
<td>1.431</td>
<td>36.155</td>
<td>0.843</td>
<td>0.000</td>
<td>(-)</td>
<td>25.518</td>
</tr>
<tr>
<td></td>
<td>Deviation</td>
<td>(4.117)</td>
<td>(1.386)</td>
<td>(4.450)</td>
<td>(3.115)</td>
<td>(4.426)</td>
<td>(10.221)</td>
<td>(204.880)</td>
<td>(10.981)</td>
<td>(35.555)</td>
<td>(-)</td>
<td>(107.763)</td>
</tr>
<tr>
<td>CH-02</td>
<td>Mean</td>
<td>10.982</td>
<td>13.106</td>
<td>34.706</td>
<td>5.110</td>
<td>250.285</td>
<td>35.861</td>
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<td>64.379</td>
<td>87.000</td>
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<td>0.845</td>
<td>1.226</td>
<td>2.188</td>
<td>1.292</td>
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<td>13.073</td>
<td>59.480</td>
<td>1.210</td>
<td>4.711</td>
<td>(-)</td>
<td>42.174</td>
</tr>
<tr>
<td></td>
<td>Deviation</td>
<td>(53.944)</td>
<td>(2.401)</td>
<td>(3.483)</td>
<td>(2.892)</td>
<td>(5.877)</td>
<td>(91.396)</td>
<td>(133.286)</td>
<td>(7.727)</td>
<td>(26.113)</td>
<td>(-)</td>
<td>(76.306)</td>
</tr>
<tr>
<td>CH-03</td>
<td>Mean</td>
<td>9.092</td>
<td>13.015</td>
<td>33.648</td>
<td>5.545</td>
<td>255.467</td>
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<td>0.544</td>
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<td>1.149</td>
<td>0.797</td>
<td>0.508</td>
<td>6.446</td>
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<td>0.957</td>
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<td>22.312</td>
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<td>Deviation</td>
<td>(21.555)</td>
<td>(1.587)</td>
<td>(3.211)</td>
<td>(1.979)</td>
<td>(4.835)</td>
<td>(69.400)</td>
<td>(209.288)</td>
<td>(5.889)</td>
<td>(33.537)</td>
<td>(-)</td>
<td>(103.303)</td>
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<td>Standard</td>
<td>0.370</td>
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<td>0.825</td>
<td>0.358</td>
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<td>0.922</td>
<td>2.108</td>
<td>(-)</td>
<td>24.867</td>
</tr>
<tr>
<td></td>
<td>Deviation</td>
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<td>(1.207)</td>
<td>(2.471)</td>
<td>(1.337)</td>
<td>(4.780)</td>
<td>(68.358)</td>
<td>(207.974)</td>
<td>(8.468)</td>
<td>(34.627)</td>
<td>(-)</td>
<td>(105.748)</td>
</tr>
<tr>
<td></td>
<td>$T_{chws}$</td>
<td>$T_{chwr}$</td>
<td>$T_{cdws}$</td>
<td>$T_{cdwr}$</td>
<td>$P_{ev}$</td>
<td>$P_{cd}$</td>
<td>$T_{dis}$</td>
<td>$m_c$</td>
<td>$m_{abe}$</td>
<td>$P_{cc}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-------</td>
<td>------------</td>
<td>------------</td>
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<td>----------</td>
<td>---------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>CH-05</td>
<td></td>
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<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Mean</td>
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<td>1.163</td>
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<td>13.580</td>
<td>2.265</td>
<td>58.339</td>
<td>1.466</td>
<td>0.476</td>
<td>(-)</td>
<td>20.093</td>
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<tr>
<td>Deviation</td>
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<td>(2.375)</td>
<td>(3.419)</td>
<td>(1.997)</td>
<td>(6.081)</td>
<td>(93.472)</td>
<td>(8.817)</td>
<td>(184.786)</td>
<td>(4.538)</td>
<td>(16.454)</td>
<td>(-)</td>
<td>(53.279)</td>
</tr>
</tbody>
</table>

**Note:**

(a) The value indicated in the bracket denotes the operating data (raw data) of a chiller before implementing the data conditioning process.

(b) The value indicated in the bracket highlighted by red colour denotes the operation data of a chiller after implementing the data conditioning process.

(c) Since there was no sensing point (i.e. sensor) provided for monitoring and measuring the operating parameter of condensing water flow rate ($m_{cw}$), no value of the standard deviation was calculated. In addition, the mean value of $m_{cw}$ indicated in the above table is based on the design value of the chiller.
### Table 4.16 Summary of Statistical Results of Data Conditioning Process of Each Chiller (Chiller Plant B)

| Standard Deviation | 0.221 | 0.774 | 2.321 | 2.198 | 0.319 | 3.926 | 2.570 | 65.108 | 1.915 | 23.392 | 8.049 | 139.284 |
| Deviation | (3.943) | (1.331) | (4.141) | (3.090) | (4.561) | (66.597) | (9.413) | (195.345) | (10.248) | (95.468) | (118.630) | (402.890) |
| CH-02 | Mean | 6.478 | 12.275 | 32.927 | 27.530 | 5.823 | 259.038 | 38.657 | 881.888 | 49.132 | 215.256 | 282.770 | 1017.540 |
| Standard Deviation | 0.273 | 0.846 | 2.789 | 2.288 | 0.387 | 4.815 | 1.835 | 47.366 | 1.380 | 19.105 | 8.909 | 160.922 |
| CH-03 | Mean | 6.621 | 12.214 | 29.530 | 24.255 | 5.708 | 257.564 | 34.334 | 773.037 | 45.446 | 221.445 | 275.504 | 927.645 |
| Standard Deviation | 0.161 | 0.557 | 2.145 | 2.187 | 0.193 | 2.385 | 2.646 | 64.033 | 2.318 | 18.433 | 4.599 | 102.805 |
| Standard Deviation | 0.191 | 0.743 | 2.411 | 2.227 | 0.224 | 2.724 | 2.391 | 60.286 | 1.884 | 17.939 | 9.009 | 132.754 |
## Data Acquisition System in Real Buildings for Individual Chiller Optimization

<table>
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<th></th>
<th>$T_{chws}$</th>
<th>$T_{chwr}$</th>
<th>$T_{cdws}$</th>
<th>$T_{cdwr}$</th>
<th>$P_{ev}$</th>
<th>$P_{cd}$</th>
<th>$T_{dis}$</th>
<th>$m_e$</th>
<th>$m_{ab}$</th>
<th>$P_{cc}$</th>
</tr>
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<td><strong>CH-06</strong></td>
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<td>8.995</td>
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<td>21.490</td>
<td>5.548</td>
<td>255.553</td>
<td>31.811</td>
<td>716.086</td>
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<td>2.242</td>
<td>1.954</td>
<td>0.177</td>
<td>2.184</td>
<td>4.078</td>
<td>96.042</td>
<td>2.710</td>
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<tr>
<td><strong>CH-07</strong></td>
<td>Mean</td>
<td>6.107</td>
<td>8.954</td>
<td>28.052</td>
<td>23.938</td>
<td>5.059</td>
<td>249.501</td>
<td>32.603</td>
<td>733.327</td>
<td>48.112</td>
</tr>
<tr>
<td></td>
<td>Standard</td>
<td>0.266</td>
<td>0.431</td>
<td>2.616</td>
<td>2.676</td>
<td>0.356</td>
<td>4.436</td>
<td>3.584</td>
<td>83.039</td>
<td>3.648</td>
</tr>
</tbody>
</table>

**Note:**

(a) The value indicated in the bracket denotes the operating data (raw data) of a chiller before implementing the data conditioning process.

(b) The value indicated in the bracket highlighted by red colour denotes the operation data of a chiller after implementing the data conditioning process.
(c) Since there was no sensing point (i.e. sensor) provided for monitoring and measuring the operating parameter of condensing water flow rate \( (m_{cdw}) \), no value of the standard deviation was calculated. In addition, the mean value of \( m_{cdw} \) indicated in the above table is based on the design value of the chiller.
Figure 4.6 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-01)

Figure 4.6 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-01)

Figure 4.6 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-01)

Figure 4.6 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-01)

Figure 4.6 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-01)

Figure 4.6 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-01)
Figure 4.6 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-01)

Figure 4.6 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-01)

Figure 4.6 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-01)

Figure 4.6 (j) Refrigerant Evaporating Profile after Data Conditioning (CH-01)

Figure 4.6 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-01)

Figure 4.6 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-01)
Figure 4.6 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-01)

Figure 4.6 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-01)

Figure 4.6 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-01)

Figure 4.6 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-01)

Figure 4.6 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-01)

Figure 4.6 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-01)
### Table 4.17 Summary of Data Acquisition Management Protocol Results for Chiller Plant A and Chiller Plant B

<table>
<thead>
<tr>
<th>Chiller Plant</th>
<th>Chiller No.</th>
<th>Period</th>
<th>Total Data Set (Rows)</th>
<th>Filtered Data Set (Rows)</th>
<th>Filtered Data Set (%)</th>
<th>Retained Data Set (Rows)</th>
<th>Retained Data Set (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>1</td>
<td>2008-01-01 to 01-01-2009</td>
<td>8,760</td>
<td>6,755</td>
<td>77.112</td>
<td>3,139</td>
<td>35.833</td>
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<tr>
<td></td>
<td>2</td>
<td></td>
<td>7,587</td>
<td>7,053</td>
<td>92.962</td>
<td>534</td>
<td>17.038</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>01-01-2008 to 01-01-2009</td>
<td>8,383</td>
<td>5,021</td>
<td>59.895</td>
<td>3,362</td>
<td>40.105</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td></td>
<td>7,010</td>
<td>6,347</td>
<td>90.542</td>
<td>3,391</td>
<td>48.374</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td></td>
<td>8,731</td>
<td>7,327</td>
<td>83.919</td>
<td>2,452</td>
<td>28.084</td>
</tr>
<tr>
<td>B</td>
<td>1</td>
<td></td>
<td>8,784</td>
<td>7,105</td>
<td>80.886</td>
<td>1,679</td>
<td>19.114</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td></td>
<td>8,784</td>
<td>7,695</td>
<td>87.602</td>
<td>1,089</td>
<td>12.398</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td></td>
<td>8,784</td>
<td>8,458</td>
<td>96.289</td>
<td>326</td>
<td>10.711</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>01-01-2009 to 01-01-2010</td>
<td>8,784</td>
<td>7,105</td>
<td>80.886</td>
<td>1,679</td>
<td>19.114</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td></td>
<td>8,784</td>
<td>7,059</td>
<td>80.362</td>
<td>1,725</td>
<td>19.638</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td></td>
<td>8,784</td>
<td>8,317</td>
<td>94.684</td>
<td>647</td>
<td>10.366</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td></td>
<td>8,784</td>
<td>7,803</td>
<td>88.832</td>
<td>981</td>
<td>11.168</td>
</tr>
</tbody>
</table>
4.4.4 Stage 4 (Range Validity Conditioning)

Figure 4.7 (a) to (l) and Figure 4.8 (a) to (l) illustrate the highest percentage of the occurrence of frequency distribution plot for each operating variable (after the data conditioning process) for various chillers for Chiller Plant A and Chiller Plant B respectively. In conjunction with Section 4.4.3, the results indicate that all of the operating variables of the chillers for both chiller plants were close to their rated values or set point values after implementing the data conditioning process for each set of the operating data of each chiller.

Moreover, some of the measured values of the particular operating data of a chiller with the highest percentage of frequency occurrences were different from others (or set point values) within the same chiller plant due to the set point values of the respective operating variable were different. The percentages of the frequency distribution plot for each operating variable for both chiller plants are summarized in Appendix D.
Chiller Plant A

Figure 4.7 (a) Highest Percentage of Occurrence of Frequency Distribution Plot for Chilled Water Supply Temperature for Various Chillers (Chiller Plant A)

Figure 4.7 (b) Highest Percentage of Occurrence of Frequency Distribution Plot for Chilled Water Return Temperature for Various Chillers (Chiller Plant A)

Figure 4.7 (c) Highest Percentage of Occurrence of Frequency Distribution Plot for Condensing Water Outlet Temperature for Various Chillers (Chiller Plant A)

Figure 4.7 (d) Highest Percentage of Occurrence of Frequency Distribution Plot for Condensing Water Inlet Temperature for Various Chillers (Chiller Plant A)
Figure 4.7 (e) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Evaporating Temperature for Various Chillers (Chiller Plant A)

Figure 4.7 (f) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Condensing Temperature for Various Chillers (Chiller Plant A)

Figure 4.7 (g) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Discharge Temperature for Various Chillers (Chiller Plant A)

Figure 4.7 (h) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Evaporating Pressure Various Chillers (Chiller Plant A)
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Figure 4.7 (i) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Condensing Pressure for Various Chillers (Chiller Plant A)

Figure 4.7 (j) Highest Percentage of Occurrence of Frequency Distribution Plot for Chilled Water Flow Rate for Various Chillers (Chiller Plant A)

Figure 4.7 (k) Highest Percentage of Occurrence of Frequency Distribution Plot for Condensing Water Flow Rate for Various Chillers (Chiller Plant A)

Figure 4.7 (l) Highest Percentage of Occurrence of Frequency Distribution Plot for Chiller Power Input for Various Chillers (Chiller Plant A)
Chiller Plant B

Figure 4.8 (a) Highest Percentage of Occurrence of Frequency Distribution Plot for Chilled Water Supply Temperature for Various Chillers (Chiller Plant B)

Figure 4.8 (b) Highest Percentage of Occurrence of Frequency Distribution Plot for Chilled Water Return Temperature Various Chillers (Chiller Plant B)

Figure 4.8 (c) Highest Percentage of Occurrence of Frequency Distribution Plot for Condensing Water Outlet Temperature for Various Chillers (Chiller Plant B)

Figure 4.8 (d) Highest Percentage of Occurrence of Frequency Distribution Plot for Condensing Water Inlet Temperature Various Chillers (Chiller Plant B)
Figure 4.8 (e) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Evaporating Temperature for Various Chillers (Chiller Plant B)

Figure 4.8 (f) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Condensing Temperature Various Chillers (Chiller Plant B)

Figure 4.8 (g) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Discharge Temperature for Various Chillers (Chiller Plant B)

Figure 4.8 (h) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Evaporating Pressure Various Chillers (Chiller Plant B)
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Figure 4.8 (i) Highest Percentage of Occurrence of Frequency Distribution Plot for Refrigerant Condensing Pressure for Various Chillers (Chiller Plant B)

Figure 4.8 (j) Highest Percentage of Occurrence of Frequency Distribution Plot for Chilled Water Flow Rate for Various Chillers (Chiller Plant B)

Figure 4.8 (k) Highest Percentage of Occurrence of Frequency Distribution Plot for Condensing Water Flow Rate for Various Chillers (Chiller Plant B)

Figure 4.8 (l) Highest Percentage of Occurrence of Frequency Distribution Plot for Chiller Power Input for Various Chillers (Chiller Plant B)
4.4.5 Stage 5 (Uncertainty Analysis)

After going through from step 1 to step 4, this section presents the results of the uncertainty analysis on heat rejection rate of a chiller. Figure 4.9 to 4.13 illustrate the uncertainty analysis results on heat rejection of CH-01 to CH-05 for Chiller Plant A respectively while Figure 4.14 to 4.20 present the uncertainty analysis results on heat rejection of CH-01 to CH-07 for Chiller Plant B respectively.

4.4.5.1 Chiller Plant A

As showed in Figure 4.9, the majority of the operating data of the calculated residual heat rejection rate (about 87%) of CH-01 were fallen within the residual threshold. It means that the annual operating data set of CH-01 within the residual threshold could be used for further chiller performance analysis since the influences of the heat loss/gain from/to the skin of a chiller and sensor fault(s) on the accuracy of the heat rejection rate were minimal.

Unlike CH-01, it was discovered that the calculated residual heat rejection rate of CH-02 for the sample number from 0 to 500 and from 2600 to 3000 as well as 1500 exceeded the residual threshold (see Figure 4.10). It implies that error(s) of the operating data for these sample numbers might exist. Therefore, they could not be
used for further chiller performance analysis and should be rejected.

For CH-03 and CH-04 (see Figure 4.11 and 4.12 respectively), their results of the uncertainty analysis on heat rejection rate were similar. Only one data set of the calculated heat rejection rate residual was out of their threshold. These two outliers (i.e. at the sample number 2603 for CH-03 and 3094 for CH-04) should be rejected when implementing the further chiller performance analysis.

Refer to Figure 4.13, it was observed that all the operating data set of the heat rejection rate residual after the implementation of data conditioning process were inside the threshold of the heat rejection rate residual for CH-05. Therefore, no operating data set is required to be rejected.
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Figure 4.9  Uncertainty Analysis on Heat Rejection Rate of CH-01
(Chiller Plant A)

q_{cd} calculated by:
\[ m_{cdw} \times C_{pw} \times (T_{cdws} - T_{cdwr}) \]

q_{cd} calculated by:
\[ q_{rl} + E_{cc} \]

q_{cd} Residual

Threshold

Figure 4.10  Uncertainty Analysis on Heat Rejection Rate of CH-02
(Chiller Plant A)
Figure 4.11  Uncertainty Analysis on Heat Rejection Rate of CH-03
(Chiller Plant A)

Figure 4.12  Uncertainty Analysis on Heat Rejection Rate of CH-04
(Chiller Plant A)
Figure 4.13 Uncertainty Analysis on Heat Rejection Rate of CH-05

(Chiller Plant A)
4.4.5.2 Chiller Plant B

It was clearly found that the calculated heat rejection rate residual for CH-02, CH-03, CH-06 and CH-07 (see Figure 15, 16, 19 and 20 respectively) were fallen within their threshold of the heat rejection rate residual. Therefore, no operating data set of these four chillers is required to be rejected.

Figure 4.14 illustrates the result of the uncertainty analysis on heat rejection rate of CH-01. It was discovered that the calculated residuals of the heat rejection rate were out of the boundary of the threshold at the sample number 1493, 1501 and 1504. Therefore, these three operating data set of CH-01 should be rejected.

For CH-04, it was identified that three outliers (i.e. out of the boundary of the residual threshold) of the calculated heat rejection rate residual were found at the sample number 276, 313, 1431 and 1604 (see Figure 4.21). These three outliers should be rejected before conducting further analysis.

Similar situation also occurred in CH-05. As illustrated in Figure 4.18, the outliers were occurred at the sample number 1221 and 1679. Therefore, they should also be refused since these operating data set might have error(s) existed.
Figure 4.14  Uncertainty Analysis on Heat Rejection Rate of CH-01

(Chiller Plant B)

Figure 4.15  Uncertainty Analysis on Heat Rejection Rate of CH-02

(Chiller Plant B)
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Figure 4.16  Uncertainty Analysis on Heat Rejection Rate of CH-03
(Chiller Plant B)

Figure 4.17  Uncertainty Analysis on Heat Rejection Rate of CH-04
(Chiller Plant B)
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Figure 4.18  Uncertainty Analysis on Heat Rejection Rate of CH-05
(Chiller Plant B)

Figure 4.19  Uncertainty Analysis on Heat Rejection Rate of CH-06
(Chiller Plant B)
Figure 4.20 Uncertainty Analysis on Heat Rejection Rate of CH-07

(Chiller Plant B)
4.5 SUMMARY

Optimizing of individual chiller by analyzing the operating data of a chiller is, no doubt, necessary. Nonetheless, inadequate and invalid data are always found in most of the chiller plants in Hong Kong. This inherent problem incurs difficult to assess the actual performance of the chillers. In this chapter, the needs of the data logic management protocol for individual chiller optimization are highlighted. In order to suit for the real situation of the chiller plant in Hong Kong, A Five Steps of Data Logic Management Protocol is, therefore, established and presented. The progress of this protocol involves: (a) Data Acquisition, (b) Data Synchronization, (c) Data Conditioning, (d) Range Validity Conditioning and (e) Uncertainty Analysis. An application of the Data Logic Management Protocol with selecting two existing chiller plants, including 12 chillers, are demonstrated and presented in this chapter.
A HES APPROACH FOR INDIVIDUAL CHILLER OPTIMIZATION

In this chapter 5, the chiller performance optimization scheme based on chiller parametric analysis approach is introduced and implemented. The chiller parametric analysis means using the relationship among different operating parameters of a chiller with regard to low power consumption and high COP under normal operation to develop a chiller performance optimization scheme. The first section of this chapter is chiller parameter analysis followed by the HES approach for chiller optimization. The third section is chiller HES optimized control case study.

5.1 CHILLER PARAMETRIC ANALYSIS

It is well realized that the performance of a chiller can deviate from the specification or catalogue provided by the manufacturer when a chiller is newly installed in the building [Liu et al. 2011]. In addition, the actual performance of the as-installed chiller depends on the structure of the pipeline and the modes of the operation. In
order to maintain the operating chiller with the highest efficiency and the lowest energy consumption, it is necessary to understand the actual performance by analyzing the logged chiller operating data from BMS. The chiller efficiency (COP) relies on the interaction among different components of a chiller with different operating parameter settings during the normal operation [Ng 2004]. Before developing a scheme for chiller performance optimization, it is better to explore how different operating parameters of a chiller vary with time sequence during its normal operation. In addition, understanding the correlation of different operating parameters of a chiller is necessary for judging how the level of each operating parameter influences the others and in terms of energy saving that could be achieved by adjusting some of the key controlled parameters.

![Figure 5.1 Flow Chart of Chiller Parametric Analysis](image-url)
5.1.1 Selected Existing Chiller Plants Description

In continuation of Chapter 4, two chiller plants named as Chiller Plant A for institutional building and Chiller Plant B for commercial building were also selected in this study.

5.1.2 Simple Plot Analysis

A simple plot analysis involves: (a) time series plot analysis and (b) part load chiller performance. The former one is to carry out an analysis on the variations of the derived data (i.e. KPI obtained by calculations). As a result, the variations of the derived data due to the degradation of the major component(s) of a chiller can be identified via the time series variations analysis. The second one in this analysis is to investigate the part load performance regarding the variations of COP of a chiller. This analysis enables us to truly understand the actual part load operating characteristics of a chiller against to the typical chiller modeling results with assumptions.

5.1.2.1 Time Series Variations

(a) Part Load Ratio

From Figure E.1.1, it was observed that the normal operating range of part load ratio
of CH-01, CH-03 and CH-04 was very broad (i.e. from 0.4 to 0.9) unlike CH-02 and CH-05. It was because that CH-01 to CH-04 are duty chillers which were used to deal with the cooling load of the building at daytime. During the daytime period, the cooling load of the building normally changed from time to time. For CH-02, the normal operating range of the part load ratio was from 0.17 to 0.58 which was at the lower side when compared with other chillers. For CH-05 (night mode chiller), the operating range of the part load ratio was from 0.62 to 0.98 in summer season and from 0.32 to 0.80 in autumn to winter season which the range was narrow in summer season (from 06-2008 to 10-2008) when compared with others. This phenomenon can be explained due to the cooling load of the building dealt by Chiller Plant A was steady at night and only CH-05 was turned-on to deal with the cooling load at night. Besides, it was also investigated that the part load ratio of each chiller occurred in winter season was low. It may be due to the outdoor air dry-bulb temperature was low and the cooling load of the building became low as well.

Figure E.1.2 shows the time serious plot of part load ratio of each chiller for Chiller Plant B. The normal operating ranges of part load ratio of the duty chillers (from CH-01 to CH-04) fell within 0.50 to 1.15. For night mode chiller (CH-05), the normal operating range of part load ratio was from 0.60 to 1.03 in summer season.
and from 0.38 to 0.58 in winter season. For essential chillers (CH-06 and CH-07), their normal operating range of the part load ratio were very similar (i.e. from 0.50 to 0.90). Similar to Chiller Plant A, the part load ratio of each chiller in Chiller Plant B was at the low side during autumn and winter season.

It is noted that little operating data of CH-03 and CH-06 were presented due to the off-line of both chillers for condenser tubes cleaning from 10-09-2009 to 23-01-2010 and from 08-09-2009 to 21-01-2010 respectively.
(b) Evaporator Overall Heat Transfer Value

As discussed in Chapter 2, chilled water pipework system is a closed-loop system in which the evaporator(s) of a chiller is connected without contacting with ambient air. Therefore, the degradation rate of the evaporator overall heat transfer value due to the evaporator fouling is normally lower when compared with open-loop system (i.e. condenser fouling). Refer to Figure E.2.1 and E.2.2, a distinct degradation trend of the evaporator overall heat transfer value of chiller for CH-01, CH-03 & CH-05 (Chiller Plant A) and CH-01, CH-02, CH-05 & CH-07 (Chiller Plant B) could be identified. For CH-02 & CH-04 (Chiller Plant A) and CH-04 & CH-06 (Chiller Plant B), no cleaning action on evaporator tubes of the chillers was conducted.
(c) **Condenser Overall Heat Transfer Value**

As presented in Chapter 3, it has discussed about the influence of the degradation of the condenser overall heat transfer due to condenser fouling on COP of chillers by artificial fault simulation. In this part, Figure E.3.1 and E.3.2 illustrate the trend of the condenser overall heat transfer value of each chiller for both chiller plants over a year. It was discovered that a distinct trend of condenser overall heat transfer value of each chiller could be observed. Apart from that, the condenser overall heat transfer value of each chiller increase sharply after the cleaning action on the condenser tubes had been carried out. When each chiller had been run for a certain period of time, it was clearly identified that the condenser overall heat transfer value of each chiller dropped. For some of the chillers in Chiller Plant A and Chiller Plant B, at least two cleaning actions on the condenser tubes were carried out in a year. Therefore, the trend of each cleaning cycle (i.e. before and after the cleaning action on the condenser tubes) of each chiller was repeated similarly within a year.

In order to have fully picture on the improvement of the condenser overall heat transfer value of a chiller after the cleaning action taking place, cleaning schedule for condensing tubes of each chiller for Chiller Plant B is presented in Table 5.1 for counter checking.
## Table 5.1  Cleaning Schedule for Condensing Tubes of Chillers (Chiller Plant B)

<table>
<thead>
<tr>
<th>Chiller Designation</th>
<th>Times</th>
<th>Start Date</th>
<th>Finish Date</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH-01</td>
<td>First</td>
<td>01-04-2009</td>
<td>25-04-2009</td>
</tr>
<tr>
<td>CH-02</td>
<td>First</td>
<td>26-05-2009</td>
<td>16-06-2009</td>
</tr>
<tr>
<td>CH-03</td>
<td>First</td>
<td>02-03-2009</td>
<td>12-03-2009</td>
</tr>
<tr>
<td></td>
<td>Second</td>
<td>10-09-2009</td>
<td>23-01-2010</td>
</tr>
<tr>
<td></td>
<td>Second</td>
<td>31-08-2009</td>
<td>08-09-2009</td>
</tr>
<tr>
<td>CH-05</td>
<td>First</td>
<td>12-03-2009</td>
<td>28-03-2009</td>
</tr>
<tr>
<td>CH-06</td>
<td>First</td>
<td>29-12-2008</td>
<td>17-02-2009</td>
</tr>
<tr>
<td></td>
<td>Second</td>
<td>08-09-2009</td>
<td>21-01-2010</td>
</tr>
<tr>
<td>CH-07</td>
<td>First</td>
<td>22-04-2009</td>
<td>06-02-2010</td>
</tr>
</tbody>
</table>
(d) Refrigerant Mass Flow Rate

In fact, the part load ratio of a chiller is directly proportional to the refrigerant mass flow rate passing through the evaporator. As illustrated in Figure E.4.1 and E.4.2, it was discovered that the trend of the refrigerant mass flow rate of each chiller for both chillers plants over a year was more or less the same as the trend of the part load ratio of each chiller (see Figure E.1.1 and E.1.2). Although the operating range of the refrigerant mass flow rate of each chiller for both chiller plants was large, no any refrigerant leakage fault was reported. As a result, the reason of the large variation of the refrigerant mass flow rate of each chiller was mainly due to the fluctuation of the cooling load of the building. The large the cooling load of the building, the higher the refrigerant mass flow rate passing through the evaporator is.
(e) **Compressor Polytropic Efficiency**

Figure E.5.1 and E.5.2 shows the trend of the compressor polytropic efficiency of each chiller for Chiller Plant A and Chiller Plant B respectively. For Chiller Plant A, the operating range of the compressor polytropic efficiency of CH-01 to CH-04 (duty chiller) was normally within 0.75 to 0.94 while CH-05 (night mode chiller) was higher (i.e. from 0.80 to 1.00). For Chiller Plant B, it was found that the normally operating range of the compressor polytropic efficiency of CH-01 to CH-04 (duty chiller) was within 0.80 to 1.00. For CH-05 (night mode chiller), the compressor polytropic efficiency range was within 0.75 to 1.00 while was similar to CH-05 of Chiller Plant A. For CH-6 and CH-07 (essential chiller), the operating range of the compressor polytropic efficiencies were within 0.70 to 1.00 and 0.61 to 0.85 respectively. It could be concluded that the normal operating range of the compressor polytropic efficiency for centrifugal chiller was within 0.70 to 1.00. However, the operating range of the compressor polytropic efficiency of CH-07 for Chiller Plant B occurred in 01-01-2009 to 01-02-2009 was below 0.70. Therefore, it was suspected that fault might be occurred.
(f) **Electro-Mechanical Loss Efficiency**

As shown in Figure E.6.1 and E.6.2, it was clearly demarcated that the fluctuation of the electro-mechanical loss efficiency of each chiller for Chiller Plant A was more stable when compared with Chiller Plant B. The normal operating range of the electro-mechanical loss efficiency of each chiller for Chiller Plant A was from 0.70 to 1.00 except CH-02. For Chiller Plant B, it was clearly observed that the efficiency of the electro-mechanical loss for CH-01, CH-02, CH-03 and CH-05 dropped when they were run for a certain period of time while it was stable for CH-04, CH-06 and CH-07. One of this phenomena might be due to the smaller the running/operating hours of the chiller is, the less the electro-mechanical loss is.
(g) **Expansion Device Discharge Coefficient**

Figure E.7.1 and E.7.2 presented the operating trend of the discharge coefficient of the expansion device of each chiller for Chiller Plant A and Chiller Plant B respectively. As discussed in Chapter 2, the refrigerant mass flow rate is directly proportional to the degree of opening of expansion device. Hence, the part load ratio of a chiller also has a relationship with the degree of opening of expansion device. Due to this reason, it was discovered that the operating trends of the discharge coefficient of the expansion device of each chiller for both chiller plants were similar to that of the part load ratio of each chiller (see Figure E.1.1 and E.1.2).
(h) **Coefficient of Performance**

Based on Figure E.8.1 and E.8.2, it was observed that the operating trend of COP of each chiller for both chiller plants followed the operating trend of condenser overall heat transfer value. This results explained that the degradation of condenser overall heat transfer value influenced the COP of a chiller seriously. Refer to Figure E.8.2, the drop of the COP of each chiller for Chiller Plant B was dominant comparing to Chiller Plant A (see Figure E.8.1). Therefore, regular monitoring the overall heat transfer value of condenser and COP of each chiller is highly recommended. In fact, an automatic cleaning system on condenser tubes including washing balls is now available in the market. This system has also been installed in Chiller Plant A. Nonetheless, Chiller Plant B was based on pre-set time schedule to carry out the cleaning action on the condenser tubes. Therefore, the degradation rate of COP or condenser heat transfer value of a chiller within Chiller Plant B was comparatively fast.
Based on the above investigation, it was found that the COP of a chiller is governed by several key factors: (a) key performance indices, (b) part load ratio and (c) season. They should be considered simultaneously when assessing the COP of a chiller. For key performance indices, it will affect the COP of a chiller due to the degradation of the performance of the major components of a chiller. A distinct operating trend of evaporator overall heat transfer value, condenser overall heat transfer value and electro-mechanical loss efficiency of the chillers could be easily identified. In addition, the condenser heat transfer value affecting the COP of a chiller was dominant when compared with others. For other KPIs, their operating trends were similar to that of part load ratio.

For part load ratio, it is noted that different types of chillers will have different performance characteristics. For instance: the peak COP of oil-free chiller normally occurs at its part load operation while the peak COP of conventional chiller normally occurs at or near to its full load condition. In order to allow the chiller running at or near its peak COP condition, in Hong Kong, the building operators will prefer to control the chiller operation manually before starting up the additional idle chiller to deal with the cooling load of the building. Apart from that, as discussed in Chapter 3 as well, the COP of a chiller will also be governed by the outdoor air condition. In
sub-tropical climate regions (e.g. Hong Kong), the outdoor air dry-bulb temperature is hot while it is cold in winter. Thus, the cooling load of a building (i.e. heat gain via building façade and ventilation load as well as infiltration load) and the loading (i.e. part load ratio) of a chiller within a chiller plant occurred in different seasons will be different.

Therefore, when assessing or modeling the performance of the chiller for optimization or fault detection, selecting the operating data set, with taking into account the above mentioned three key factors, for training or calibrating the chiller model should be identified first. Otherwise, using wrong operating data set of a chiller to train or calibrate the model may have wrong interpretation of the actual performance of a chiller.
5.1.2.2 Part Load Performance

As discussed in the previous part of time series variations, there are three major factors affecting the COP of a chiller (i.e. KPIs, part load ratio and season). This part is to investigate the influence of part load ratio and season on the COP of a chiller. Figure E.9.1 (a) to (e) and Figure E.9.2 (a) to (g) present the part load performance of different chillers with monthly bin groups for Chiller Plant A and Chiller Plant B respectively. For Chiller Plant A, some of the monthly operating data of the respective chiller were missing due to the data loss or after the data logic analysis process while the complete monthly operating data set of each chiller in Chiller Plant B could be guaranteed.

It was clearly investigated that a distinct cluster of each chiller within Chiller Plant A could be identified. In addition, the cluster of each chiller was similar and their patterns of the cluster of each chiller were similar to typical simulation results of centrifugal chiller model (empirical, semi-empirical or physical base).

For Chiller Plant B, different clusters of various chillers were identified. Apart from that, it was discovered that two distinct clusters of CH-01 and CH-02 were observed (see Figure E.9.2 (a) and (b)). Besides, it was also observed that the upper cluster of
CH-01 and CH-02 occurred in between winter and spring season (from January to May). The percentage of the difference of the peak COP occurred in these two clusters was about 30% which can affect the power consumption of the chiller significantly. For other chillers within Chiller Plant B, the peak COP of each chiller still occurred in between winter and spring season. These observations prove that using typical simulation centrifugal chiller model cannot truly reflect the actual performance of all chillers since the operation of the chillers together with their associated equipment as well as the changes of the outdoor air condition can influence the actual performance of chillers. In fact, low outdoor air dry-bulb temperature can reduce the refrigerant condensing temperature/pressure and compressor power input which relieves the compression ratio (i.e. refrigerant condensing pressure over refrigerant evaporating pressure) without scarifying the refrigeration effect. In addition, it was also discovered that the part load ratios of each chiller in Chiller Plant B and CH-05 in Chiller Plant A were higher than 1.0. This phenomenon may be due to the chiller has lower compression ratio and higher cooling effect occurred in cooler days.
5.1.3 Chiller Efficiency Signature Analysis

This part is to understand the operating characteristics of the chillers in order to seek for individual chiller optimization. The relationship between the coefficient of performance (COP) and the part load ratio (PLR) with the variations of different operating variables (i.e. controllable variables and KPIs) for each chiller was examined. It is noted that a “BIN” method was adopted in this study and this method was used to divide an operating variable into a series of strips (i.e. bin groups) for facilitating analysis and simple application. In addition, since the operating data point of each chiller were widely scattered, averaging the operating data point with respect to the PLR (i.e. x-axis) was applied in order to minimize that effect.

---

**BIN Method:**

In HVAC field, the BIN method was established for building energy calculation with cooling and heating system by dividing the outdoor air temperature into a series of bins [ASHRAE 1981; Hanby 1995; Akbari and Konopacki 2005]
5.1.3.1 Chilled Water Supply Temperature

In Hong Kong, the building operators normally attempt to increase the chilled water supply temperature of the chiller when the outdoor air dry-bulb temperature is low as they believe that increasing in chilled water supply temperature of the chillers will incur enhancing the COP of each chiller and hence energy saving can be achieved. Apart from that, some researchers also recommend to reset the chilled water supply temperature for energy saving purpose.

As shown in Figure E.10.1 and E.10.2, it was discovered that the COP of each chiller in both chiller plants decreased when the chilled water supply temperature increased. This phenomenon deviates from the thermodynamic principle. In fact, the COP of each chiller is influenced by a lot operating variables (e.g. chilled water supply temperature and condensing water inlet temperature etc.), outdoor environmental condition (e.g. outdoor air dry-bulb and wet-bulb temperature), occupants’ habit on controlling the set point of the indoor air temperature of air-conditioning systems or units and control logic of the chiller plant etc. coincidentally. In theory, if the other operating variables (except chilled water supply temperature) of a chiller are operated/kept in constant, increasing in chilled water supply temperature may have energy saving due to enhance in COP. It can be interpreted by using typical chiller...
models. Nevertheless, Figure E.10.1 and E.10.2 proved that using typical chiller models cannot truly/wrongly interpret the actual characteristics of the performance of the chiller.

When looking at the correlation between chilled water supply temperature and condensing water inlet temperature of all chillers installed in Chiller Plant A, it was observed that the condensing water inlet temperature increased when the chilled water supply temperature increased (see Figure 5.2 (a) to (e)). Due to this phenomenon, it could be explained the reason why the COP of each chiller in Chiller Plant A decreased despite the chilled water supply temperature increased. In addition, this phenomenon also proved that the COP of each chiller in Chiller Plant A was more sensitive to the condensing water inlet temperature when comparing with chilled water supply temperature.

The operating ranges of the chilled water supply temperature of all chillers installed in Chiller Plant B were comparatively lower (i.e. from 5.0°C to 8.0°C) when compared with Chiller Plant A (from 7.0°C to 14.0°C). Based on the plot of COP of chiller against part load ratio with different chilled water supply temperature BIN groups for Chiller Plant B, it is difficult to justify whether increasing the chilled
water supply temperature could enhance or reduce the COP of each chiller in Chiller Plant B. It is noted that the chilled water supply temperature decreased when the part load ratio of the chillers in Chiller Plant B increased (see Figure 5.3 (a) to (g). On the other hand, Figure 5.3 (a) to (g) also illustrated that the low chilled water supply temperature of the chillers was normally occurred at high part load ratio. This phenomenon were quite often in CH-01, CH-02, CH-04 to CH-07 of Chiller Plant B. Due to this reason, it was suspected that the part load ratio of the chillers in Chiller Plant B outweighs the chilled water supply temperature to the effect of COP.
Chiller Plant A

Figure 5.2 (a) Condensing Water Inlet Temperature Vs Chilled Water Supply Temperature (CH-01)

Figure 5.2 (b) Condensing Water Inlet Temperature Vs Chilled Water Supply Temperature (CH-02)

Figure 5.2 (c) Condensing Water Inlet Temperature Vs Chilled Water Supply Temperature (CH-03)

Figure 5.2 (d) Condensing Water Inlet Temperature Vs Chilled Water Supply Temperature (CH-04)

Figure 5.2 (e) Condensing Water Inlet Temperature Vs Chilled Water Supply Temperature (CH-05)
Figure 5.3 (a) Chilled Water Supply Temperature Vs Part Load Ratio (CH-01)

Figure 5.3 (b) Chilled Water Supply Temperature Vs Part Load Ratio (CH-02)

Figure 5.3 (c) Chilled Water Supply Temperature Vs Part Load Ratio (CH-03)

Figure 5.3 (d) Chilled Water Supply Temperature Vs Part Load Ratio (CH-04)

Figure 5.3 (e) Chilled Water Supply Temperature Vs Part Load Ratio (CH-05)

Figure 5.3 (f) Chilled Water Supply Temperature Vs Part Load Ratio (CH-06)
Figure 5.2 and 5.3  Condensing Water Inlet Temperature Vs Chilled Water Supply Temperature (Chiller Plant A) and Chilled Water Supply Temperature and Part Load Ratio (Chiller Plant B)

Figure 5.3 (g) Chilled Water Supply Temperature Vs Part Load Ratio (CH-07)
5.1.3.2 **Condensing Water Inlet Temperature**

Figure E.11.1 and E.11.2 present the chiller efficiency signature with condensing water inlet temperature binned for Chiller Plant A and Chiller Plant B respectively. Despite the trend of some condensing water inlet temperature curves at a particular temperature range were distorted when compared with others, it was still clearly identified that the lower the condensing water inlet temperature curve, the higher the COP is. Therefore, this observation enables the concept of chiller optimization (i.e. enhance the chiller COP) by reducing the condensing water inlet temperature.
5.1.3.3 Chilled Water Flow Rate

In fact, all the chillers installed in Chiller Plant A are in constant flow (including chilled water and condensing water). Apart from that, the variation of the chilled water flow rate reflected by the BMS logged operating data is mainly due to the sensor error while those data were still accepted after the data logic analysis processing. Therefore, the variation of the chilled water flow rate affecting the COP of each chiller within Chiller Plant A was not studied.

In Figure E.12, almost all COP of each chiller, except CH-05, within Chiller Plant B increased when the chilled water flow rate increased. Therefore, the COP of a chiller can be enhanced by increasing the chilled water flow rate via the evaporator.
5.1.3.4 Condensing Water Flow Rate

As mentioned that a heat rejection system installed in Chiller Plant A is based on a constant condensing water flow and no sensor is provided for monitoring and logging the actual condensing water flow rate for each chiller, no analysis on the changes of the condensing water flow rate with regard to the COP of each chiller within Chiller Plant A was studied.

Among seven chillers installed in Chiller Plant B (see Figure E.13), it was investigated that the COP of each chiller decreased when the condensing water flow rate increased. Again, this phenomenon violates the thermodynamic principle since increasing condensing water flow rate can increase the overall heat transfer value of condensing of each chiller and hence the COP. As also discussed on the change of COP of a chiller with respect to the chilled water supply temperature previously, the change of the COP of a chiller is subject to different operating variables (e.g. chilled water supply temperature, condensing water inlet temperature and outdoor environmental condition etc.) coincidentally. Therefore, this phenomenon gives a picture that using typical chiller models cannot truly/fully describe the actual performance of a chiller.
5.1.3.5 **Evaporator Overall Heat Transfer Value**

Figure E.14.1 show the chiller efficiency signature with evaporator overall heat transfer value for Chiller Plant A. It was identified that with the increasing in overall heat transfer value of evaporator, the COP of each chiller was also increased. This phenomenon was also occurred for some chillers (CH-03, CH-04, CH-06 and CH-07) installed in Chiller Plant B.

Unlike Chiller Plant A, a variable chilled water flow chiller system is adopted in Chiller Plant B. Referring to Figure E.14.2, it was quite surprising that increasing in overall heat transfer value of evaporator incurred decreasing in COP of some chillers (CH-01, CH-02 and CH-05). It may be due to the reduction of chilled water flow rate flowing through the evaporator of the chillers.
5.1.3.6 Condenser Overall Heat Transfer Value

As shown in Figure E.15.1 and 15.2, high COP could be achieved when the overall heat transfer value of condenser was high. This phenomenon always occurs in both chiller plants except CH-02 in Chiller Plant A. For CH-02 in Chiller Plant A, the reason was that since inadequate operating data were obtained, due to data loss and data with fault (after DLA), to generate the chiller efficiency signature causing wrong interpretation. In general, it can be proved that increasing or maintaining the overall heat transfer value of condenser of each chiller to a certain level can optimize the performance of chillers.
5.1.3.7 Compressor Polytropic Efficiency

Refer to Figure E.16.1, it was discovered that the operating range of compressor polytropic efficiency was governed by the part load ratio of a chiller. The higher the part load ratio of a chiller, the higher the value of the operating range of compressor polytropic efficiency of each chiller within Chiller Plant A is. This situation was also observed in Chiller Plant B (see Figure E.16.2). Due to this reason, it can be concluded that the compressor polytropic efficiency is low when the chiller (water-cooled centrifugal chiller) is running at its part load condition.

In addition, it was also investigated that the COP of each chiller for both chiller plants was high when the compressor polytropic efficiency, with running at high part load ratio, was high except the operating range of compressor polytropic efficiency curve at 1.0 to 1.1 or 1.0 to 1.2 for Chiller Plant B (from CH-05 to CH-07). In fact, the compressor polytropic efficiency of a chiller is universally not greater than 1.0. This phenomenon was mainly due to sensor error. In order to identify the actual operating characteristics of each chiller, the curve can be neglected when the operating range of the compressor polytropic efficiency of a chiller is greater than 1.0.
5.1.3.8 **Electro-Mechanical Loss Efficiency**

Figure E.17.1 and E.17.2 illustrate the operating characteristics of each chiller with regard to the electro-mechanical loss efficiency for both chiller plants respectively. It was easily observed that the higher the electro-mechanical loss efficiency the higher the COP of a chiller is. Therefore, maintaining high electro-mechanical loss efficiency can enhance the COP of a chiller.
5.1.3.9 Refrigerant Mass Flow Rate

The chiller efficiency signatures with different refrigerant mass flow rate binned groups of each chiller for Chiller Plant A and Chiller Plant B were presented in Figure E.18.1 and E.18.2. It was observed that the refrigerant mass flow rate is also governed by the part load ratio of a chiller. The higher the part load ratio, the higher the refrigerant mass flow rate is. In addition, it was also discovered that the refrigerant mass flow rate also influences the COP of each chiller. The higher the refrigerant mass flow rate, the higher the COP of a chiller is.
5.1.3.10 Expansion Device Discharge Coefficient

Refer to Figure E.19.1 and E.19.2, it was clearly identified that the phenomena of increasing the expansion device discharge coefficient was similar to refrigerant mass flow rate (see Figure E.18.1 and E.18.2). With increasing the part load ratio, the expansion device discharge coefficient will also increase. In addition, the higher the expansion device discharge coefficient of a chiller, the higher the COP is.
5.1.4 Chiller Parameters Sensitivity Analysis

It is understood that the variations of the operating variables of a chiller will affect its COP and in terms of power consumption. Therefore, it is worthwhile to investigate the variations of each operating variable of a chiller to the changes of its COP. The sensitivity analysis based on the sensitivity coefficient \((SC)\) was adopted in this study [Spitler et al. 1989]. The sensitivity coefficient is an indicator that reflecting the rate of change of the COP (i.e. output) of a chiller with respect to the variation of each operating variable (i.e. input). The higher the \(SC\), the greater variation in chiller COP for a unit of relative change in the operating variable. In order to provide a common base for assessing different chillers, normalization to both input and output variables based on their mean values was adopted. It is noted that the operating variables (i.e. output) of a chiller will be divided into two groups. Group 1 is the controllable variables and Group 2 is the key performance indices. The calculation of \(SC\) is expressed below:

\[
SC = \frac{|O - O_{\text{mean}}|}{O_{\text{mean}}} + \frac{|I - I_{\text{mean}}|}{I_{\text{mean}}}
\]  

(5.1)

where \(O_{\text{min}}, O_{\text{max}}\) and \(O_{\text{mean}}\) are the minimum, maximum and mean value of the Chiller COP respectively. \(I_{\text{min}}, I_{\text{max}}\) and \(I_{\text{mean}}\) are the minimum, maximum and mean value of the operating variable of a chiller respectively. \(\left| O - O_{\text{mean}} \right| = \)
Indeed, investigating the sensitivity of each controllable variable influencing on the COP of a chiller can provide a room for energy saving opportunity by re-adjusting the set point of each controllable variable in order to enhance the chiller COP for each chiller plant (i.e. operation optimization). While for assessing the sensitivity of each KPI, it is to enhance the chiller COP by monitoring the performance of each component of each chiller in order to maintain an acceptable performance of each chiller before running to poor performance or faulty condition (i.e. maintenance optimization).

Figure 5.4 and 5.5 illustrate the results of sensitivity analysis of each controllable variation influencing on the coefficient of performance of chillers for Chiller Plant A and Chiller Plant B respectively. For Chiller Plant A, it is noted that due to no sensing point for monitoring and logged the operating data of condensing water flow rate and chilled water flow rate, these two controllable variables were excluded in this study. On the other hand, it was observed that the ranking of the influence of controllable variations on the COP of each chiller for Chiller Plant A was led by

$$|O_{min} - O_{mean}| \text{ or } |O_{max} - O_{mean}| \text{ whichever is higher.}$$

$$|I_{min} - I_{mean}| \text{ or } |I_{max} - I_{mean}| \text{ whichever is higher.}$$

Indeed, investigating the sensitivity of each controllable variable influencing on the COP of a chiller can provide a room for energy saving opportunity by re-adjusting the set point of each controllable variable in order to enhance the chiller COP for each chiller plant (i.e. operation optimization). While for assessing the sensitivity of each KPI, it is to enhance the chiller COP by monitoring the performance of each component of each chiller in order to maintain an acceptable performance of each chiller before running to poor performance or faulty condition (i.e. maintenance optimization).
condensing water inlet temperature and followed by chilled water flow rate and chilled water supply temperature as well as part load ratio. For Chiller Plant B, the ranking result was led by condensing water flow rate, followed by condensing water inlet temperature, chilled water supply temperature and chilled water flow rate as well as part load ratio.

Therefore, the first priority of energy saving for Chiller Plant A is to decrease the condensing water inlet temperature by increasing the fan speed of the cooling tower. The energy saving of the chiller power may compensate the additional power consumption due to increasing the fan speed of the cooling tower. Alternatively, using fixed approach method (normally at 3.5°C or 4.0°C for sub-tropical climate regions) based on the changes of the outdoor wet-bulb temperature to control the set point of the condensing water inlet temperature can also have energy saving opportunities. The second consideration for energy saving for Chiller Plant A is to decrease the chilled water supply temperature of each chiller. This strategy can be adopted when the outdoor air dry-bulb temperature is low especially in winter season or the building cooling load is low especially at night mode operation. The last consideration for energy saving is to control the part load ratio of each chiller. This approach is to allow the running chiller(s) to deal with the cooling load and start up
another chiller until the running chiller(s) has/have been run at full load condition.

For Chiller Plant B, it was discovered that the condensing water flow rate was the most sensitive controllable variable to all chillers, except CH-04 to CH-07, within Chiller Plant B. For CH-04 to CH-07, the most sensitive variable to its COP was condensing water return temperature. The second sensitive controllable variable to chiller COP was the condensing water inlet temperature followed by chilled water supply temperature and chilled water flow rate as well as part load ratio.
Figure 5.4  Sensitivity Analysis of Each Controllable Variable Influencing on Coefficient of Performance of Chillers (Chiller Plant A)

Figure 5.5  Sensitivity Analysis of Each Controllable Variable Influencing on Coefficient of Performance of Chillers (Chiller Plant B)
Table 5.2  Summary of Sensitivity Coefficient of Each Controllable Variable Influencing on Coefficient of Performance of Chillers (Chiller Plant A)

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<td></td>
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<td>0.641</td>
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Table 5.3  Summary of Sensitivity Coefficient of Each Controllable Variable Influencing on Coefficient of Performance of Chillers (Chiller Plant B)

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Figure 5.6 and 5.7 show the results of sensitivity analysis of each KPI influencing on the coefficient of performance of each chiller for Chiller Plant A and Chiller Plant B. It was discovered that the most sensitive KPI with respect to chiller COP for both chiller plants was the compressor polytropic efficiency followed by the electro-mechanical loss efficiency. Nevertheless, the condenser overall heat Transfer value of each chiller for both chiller plants was less sensitive to the chiller COP which is unlike the results did by many researchers based on laboratory chiller test or unitary packaged unit [Jia and Reddy 2003; Breuker and Braun 1998; Comstock and Braun 1999a and 1999b]. This result may prove that the as-installed condition of chillers is different from the chiller test under controlled environment.

After examining the results of sensitivity analysis of the KPIs influencing on the chiller COP, regular monitoring the degradation level of KPIs for each chiller is suggested for the building operators for both chiller plants. It also serves as a base for building operators to make a decision on condition-based maintenance strategy (whether maintenance action should be firstly taken – e.g. compressor and motor etc.).
Figure 5.6 Sensitivity Analysis of Each Key Performance Index Influencing on Coefficient of Performance of Chillers (Chiller Plant A)

Figure 5.7 Sensitivity Analysis of Each Key Performance Index Influencing on Coefficient of Performance of Chillers (Chiller Plant B)
Table 5.4  Summary of Sensitivity Coefficient of Each Key Performance Index Influencing on Coefficient of Performance of Chillers (Chiller Plant A)

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<td>(\eta_{\text{m}})</td>
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Table 5.5  Summary of Sensitivity Coefficient of Each Key Performance Index Influencing on Coefficient of Performance of Chillers (Chiller Plant B)

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5.2 HEURISTIC-ENGINEERING-STATISTICAL APPROACH FOR CHILLER OPTIMIZATION

In general, the objective of the optimization based on HES approach is to move the operating performance from low optimization zone to high optimization zone. In this chapter, the procedures of the general HES optimization protocol is firstly presented. After that, the HES approach is applied for chiller optimized control and it discusses in the second part of this section.

5.2.1 HES Optimization Protocol

Step 1 Identification of Optimization Mapping

For all optimization processes, there must be a clear objective or objectives for an assessed goal. In the Chapter, this HES protocol is developed for optimized operation of chillers. For cooling system, energy consumption is obviously the key concern. Also, the effectiveness of heat rejection should be a key objective as well. Therefore, the optimization map is defined as the mapping of the data pair ($P_{cc}$, COP). The target is to move the operating performance on the upper left hand corner towards the bottom right hand corner heuristically.
The philosophy of this study is that working experience in chillers in buildings in use, it is futile to target for the most optimum point at all time because it is almost impossible to adjust the parameters in the parameter groups (micro-climate, load and machine inherent) to their corresponding most optimum values where these values cannot be achieved due to the interlocking relationships among these parameters.

![Figure 5.8 Example of Chiller Optimization Map](image-url)
Step 2  **Transformation of Optimization Map**

After plotting $P_{cc}$ Vs COP, a “Trapezium” pattern formed by the data points is normally observed. Therefore, the classification of the points is taken along the diagonal from the top left hand corner to the bottom right hand corner. The data within an interval between two corresponding orthogonal lines are taken as a zone.

For easily classifying the zones, the original two axes (i.e. $P_{cc}$ and COP) are transformed by rotating the Y axis (i.e. $P_{cc}$) to be aligned with this diagonal. With applying the equations of polar transformation, the optimized zones can then be easily classified by using excel sheet or computing program.

**Figure 5.9**  Optimization Map with Classification of Data Points before Transformation

*Note: The data are classified according to the range of the optimization and the number of data available in each zone.*
Figure 5.10 Transformation of Optimization Map

Figure 5.11 Optimization Map with Classification of Data Points after Transformation
Step 3  **Identification of Controllable Variables**

In optimizing the chiller operation by controlling the variable set points, it is important that these variables can be set and controlled. These variables can be divided into three groups:

- Group 1 – Environment Group;
- Group 2 – Equipment Group;
- Group 3 – Physical Group.

Group 1 is the act of God and is considered as independent conditions not controllable. Group 2 pertains to the pre-setting of the equipment or a component in the equipment. In Group 3, the physical parameters are resettable and hence controllable for optimized control of the chiller.

The essence is to get more controllable parameters for chiller optimization. With Group 1, the environmental resources can be maximized by design and installation. Some parameters in parameters in Group 2 pertaining to the factory pre-setting of the chiller can be fine-tuned with the collaboration of the supplier. Parameters in Group 3 are the normal system operation parameters and therefore can be feasibly re-set for optimized operation. Very often, the re-set values can be ascertained with the change in operation group and sequence.
Step 4  **Determination of Set Points Setting by Prevalence Distribution**

By extracting each controllable variable in the candidate set from each zone, the frequency distribution graph (named as “Prevailing Distribution Graph”) for each controllable variable can be generated.

In this Prevailing Distribution Graph, the parametric value with the highest prevalence in the most optimum zone (i.e. from right bottom corner zone) represents the most likely condition that the chiller is operating in this zone. The output for each controllable variable in each zone will be the optimized range setting instead of single point setting.

![Figure 5.12 Example of Prevailing Distribution Graph for Specified Controllable Variable](image-url)
Step 5  **Determination of Chiller Optimized Range Settings**

When an annual operational cycle is mapped, the optimized operation of a chiller can be predicted and planned. For automatic optimized control, a statistical algorithm can be set and the chiller can be controlled according to the environmental and load conditions. It is more effective in the beginning of this HES control algorithm by manual. When the optimized operation is ascertained and the side effects monitored and reinstate, the chiller and chiller group can be put into automatic operation. The procedure is illustrated in the next section.

![Optimization Map after Transformation](image)

**Figure 5.13**  Example of Determination of Chiller Optimized Range Settings
Figure 5.14 Flow Chart of Chiller HES Optimization
5.3 CHILLER HES OPTIMIZED CONTROL CASE STUDY

After the process of the optimization zones classification for the selected 12 chillers of both chiller plants have been implemented, the results of the optimal operating range settings under different optimization zones for the chillers can be identified. In this section, a set of the optimal operating range settings, based on controllable variables, under different optimization zones for each chiller are determined. In the meantime, the graphical results showing the optimal operating range settings under different efficiency zones are also presented.

5.3.1 Optimization Zones of Each Chiller for Both Chiller Plants

As discussed in Section 5.2, after gone through the process of the data logic management, the chiller performance optimization can then be proceeded. Table 5.6 and 5.7 summarize the optimal operating range settings based on controllable variables under different optimization zones for Chiller Plant A and Chiller Plant B respectively. The figures illustrating the chiller optimization zones identification before and after transformation and the controllable variables under different chiller optimization zones for Chiller No. 1 in both chiller plants are presented Figure 5.15 to 5.16. Figure E.20 and E.21 presented for all chillers in both chiller plants.
Table 5.6  Optimal Operating Range Settings based on Controllable Variables under Different Optimization zones of Chillers

(Chiller Plant A)

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Table 5.7  Optimal Operating Range Settings based on Controllable Variables under Different Optimization zones or Chillers

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<td>94 – 105</td>
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CHAPTER 5

A HES APPROACH FOR INDIVIDUAL CHILLER OPTIMIZATION

Chiller Plant A

Figure 5.15 (a) Chiller Optimization Map before Transformation (CH-01)

Figure 5.15 (b) Chiller Optimization Map after Transformation (CH-01)

Figure 5.15 (c) Chilled Water Supply Temperature under Different Chiller Optimization Zones (CH-01)

Figure 5.15 (d) Condensing Water Inlet Temperature under Different Chiller Optimization Zones (CH-01)

Figure 5.15 (e) Part Load Ratio under Different Chiller Optimization Zones (CH-01)

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**Chiller Plant B**

**Figure 5.16 (a) Chiller Optimization Map before Transformation (CH-01)**

**Figure 5.16 (b) Chiller Optimization Map after Transformation (CH-01)**

**Figure 5.16 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-01)**

**Figure 5.16 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-01)**

**Figure 5.16 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-01)**

**Figure 5.16 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-01)**
Figure 5.16 (g) Part Load Ratio under Different Chiller Optimization Zones (CH-01)
5.4 SUMMARY

The philosophy in this study lies in the fact that chiller performance in a real site is a complicated issue. The performance is affected by its inherent design and setting of the components, the environmental factor and the operating conditions. It is therefore difficult to control a chiller always operates at an optimum point. Also, it is easy to evaluate the dependence of its performance relative to the operating parameters because it is almost impossible to obtain the same set of coincidental operating conditions even in this study that the most optimum value can be determined by the prevalence in high optimum zones in the optimization mapping.

The HES approach is further confirmed by data of the chiller operation itself. The approach not only truly produces results of ‘steady state experimental results’, it identifies optimum values for pragmatic setting and self-verifiable predicted results. As proven by the data acquired from two chiller plants, optimization strategies can be set up with results not simulated but revealed by data. To facilitate this approach, an intelligent facility is set up as described in Section six.
Firstly, a chiller parametric analysis based on simple plot is conducted. Through this analysis, each controllable variable and KPI of a chiller operated over a period of time can be investigated. For controllable variables, the control settings of a chiller in different seasons can be clearly identified. The degradation of the performance of a chiller via monitoring the variations of the KPIs of each component of a chiller can also be analyzed. Secondly, another chiller parametric analysis based on chiller efficiency signature with utilizing different family curves (i.e. COP Vs PLR with different BIN groups of the controllable variable and KPI) is studied. This study provides a preliminary investigation on seeking opportunities for individual chiller performance optimization. Thirdly, a sensitivity analysis of controllable variables and KPIs on the COP of a chiller is carried out. The most deterministic controllable variable and KPI can be identified.

After discussing the chiller parametric analysis, the HES approach for chiller optimization is presented. For HES optimization protocol, the general procedures are detail discussed. The main concept is to move the operating condition from low optimized zone to high optimized zone. With applying this HES approach to chiller optimization, the procedures for chiller HES optimized control are then presented. The objective of this chiller performance optimization protocol is to seek for optimal
control settings for chiller operation by establishing optimization map for the chiller with moving the operating performance of the chiller from the upper left hand corner towards to the bottom right hand corner. Lastly, a cast study of chiller HES optimized control for two existing chiller plants is studied. The optimal range of control settings with graphical illustration is also presented in this chapter.
Having studied the previous two chapters focusing on the data logic management to deal with the raw data from BMS database and the HES approach for data mining process for individual chiller optimization, this chapter presents the development of an electronic web-based integrated analysis scheme for chiller performance optimization (named as Chiller Server) with its applications.

6.1 CHILLER SERVER DEVELOPMENT

With combining the functions of the data logic management protocol and the chiller performance optimization scheme based on chiller parametric analysis approach presented in Chapter 4 and 5, an integrated scheme for individual chiller performance optimization can be achieved. Nowadays, using electronic web-base to monitor and control the performance of chillers has been widely adopted for more than twenty decades in Hong Kong.

The architectural structure of an electronic web-based integrated chiller server for individual chiller optimization is illustrated in Figure 6.1. It is noted that this chiller server is an off-line based. This chiller server can be used as a commercial tool for monitoring and optimizing the actual performance of chillers. The input of the chiller
server is divided into three parts: (a) BMS database for logging the operating data of chiller plant, (b) Data Logic Analysis criteria for addressing the rules for validating the chiller data whether the data can fulfill the pre-set rules or not and (c) refrigerant models for modeling the thermo-properties of different refrigerant at different working conditions. For the output of the chiller servers, it consists of five functions which conclude: (a) time series plot for all operating variables, (b) monthly plot for COP against PLR of chillers, (c) energy efficiency signature for plotting family curves of chillers, (d) chiller optimal control set point for individual chiller optimization and (e) chiller model calibration.

Figure 6.1 Architectural Structure of Electronic Web-Based Integrated Chiller Server for Individual Chiller Optimization
6.2 CHILLER SERVER PLATFORM WITH APPLICATIONS

6.2.1 Time Series Plot

As discussed in Chapter 3 and 5, the operating control settings and the degradation of the KPIs will influence the chiller performance. Regular monitoring the changes of the operating control set points and the degradation levels of the KPIs over a certain period of time can enable the building operators to understand the actual operating conditions of the chillers. This action is a decision making process for the building operators whether re-adjusting the control set points and maintenance task for the chillers should be carried out or not. One of the functions of the chiller server provides a time series plot for each operating variables (including all operating variables as well as KPIs) of a chiller. Figure 6.2 illustrates an example of the chiller server interface showing the degradation trends of the overall heat transfer value of condenser for CH-01 within Chiller Plant B.
6.2.2 Monthly Plot

In previous Chapters, it is mentioned that the part load ratio of a chiller and the outdoor environmental condition are also the deterministic factors governing its COP. Thus, realizing the variations of the COP, with respect to part load ratio, of the chillers over a period of time is vital for building operators to understand the actual performance, in terms of energy efficiency, of the chillers. It is also understood that the changes of the outdoor environmental conditions is correlated to different seasons. As a result, it is beneficial to investigate the changes of the COP at different part load ratios of the chillers with monthly base. One of the functions of the developed chiller server enables this task as well. Figure 6.3 shows the chiller interface for plotting the monthly variations of the COP with respect to part load ratio of CH-01 in Chiller Plant B.
6.2.3 Family Curves Plot (Chiller Efficiency Signature)

Chiller efficiency signature is a set of family curves showing the actual operating characteristics, with regard to the energy performance, of a chiller. It is well realized that the actual performance of a chiller can deviate from the specification. With make use of the curves, it allows the building operators to understand the actual variations of each operating variables influencing on the chiller electricity consumption as well as its COP coincidentally. It provides aid and is a first step for considering chiller optimization. Due to its importance, this task is also included in the chiller server. Figure 6.4 shows the chiller server interface for plotting the chiller family curves of refrigerant condensing temperature for CH-01 in Chiller Plant B.
6.2.4 Chiller Efficiency Optimizer

The core part of the chiller server is the chiller efficiency optimizer. The function of the chiller efficiency optimizer is an integrated part of the data logic management and the individual chiller efficiency optimization. By extracting the operating data of the chillers from the BMS database, different optimization zones can be identified and hence, the optimal set points of the controllable variables and the KPIs for each chiller can be obtained. Figure 6.5
Figure 6.5  Chiller Server Interface for Identifying the Optimal Control Settings (Condensing Water Inlet Temperature and Overall Heat Transfer Value of Condenser) for Different Chiller Optimization Zones for CH-01 (Chiller Plant B)
6.2.5 Chiller Model Calibration

Other than the above mentioned functions of the chiller server, an additional function for modeling the performance of the chiller based on Gordon-Ng’s chiller model is also included in the server [Gordon et al. 1995]. An automatic calibration procedure for determining the coefficients of the model can be proceeded via the server. In addition, a graphical presentation for highlighting the accuracy of the prediction of the COP of the calibrated chiller model comparing with the actual measured data can also be implemented via the chiller server. Figure 6.6

![Figure 6.6 Chiller Server Interface for Chiller Model Calibration before and after Calibration for EIRFPLR Curve for CH-04 (Chiller Plant B)](image)
6.3 SUMMARY

This chapter presents the development of an electronic web-based integrated analysis scheme for individual chiller performance optimization (i.e. chiller server). This chiller server is established based on the chiller parametric approach with combining the function of the data logic management protocol and chiller performance optimization scheme for online based analysis. Three input data of the chiller are required (including chiller operating data, criteria setting for data logic analysis and refrigerant model) which can generate five outputs (including time series plot for all operating variables of chiller, monthly plot of COP Vs PLR, energy efficiency signature of chiller, determination of chiller optimal control set points and chiller model calibration). The applications of the chiller server are also highlighted in this chapter.
The main purpose for this study intends to develop a pragmatic approach for chiller optimization for its operation and maintenance. It differs from the contemporary studies in the sense that on one hand, the HES analysis is solely based on the data trend of the chiller itself rather than the generic modeling. Secondly, a user interface is conceived and constructed in this study to facilitate the real life optimization.

The HES approach is firstly demonstrated by the comprehensive analysis of energy consumption data in buildings. Engineering modeling contours are then imposed on these data trends to assess the parametric impacts. The parametric trends inspire an optimization strategy which can then be developed into a protocol for optimization settings of operating conditions. In this study, the HES approach is applied into two aspects, the total energy consumptions of a building with respect to the outdoor climatic conditions and the chiller performance with respect to the optimization mapping of energy consumption and COP. The former analysis is important because of the breakdown of the energy consumption reveals the heat rejection efficiency of
the air-conditioning systems which in turn will have an impact to the optimization setting of the chiller. The latter is very effective way to derive the more precise settings of parameters for highest optimization operation and maintenance. Hence, the approach is named Heuristic-Engineering-Statistical approach. This approach is particularly useful for knowledge mining based on the massive data acquisition from the BMS in building air-conditioning systems.

Energy conservation measures are norms for building system operation. The quality and capability of facility management teams today are much higher than the last few decades. In general, the modern buildings are more sustainable in design and the buildings are more efficiently operated. Nonetheless, the improvement in design is more on the deployment of renewable energy and passive design such as façade enhancement. The improvement in operational efficiency is more on scheduling and replacement of less efficient devices and equipment. In fact, more energy can be saved from a more precise and optimum setting of operational parameters of equipment and system. The BMS data are not fully utilized by the facility managers because of no in-depth analytic tools for such use exists. Therefore, the objective of this research will not be completed within the interface between the data acquired, analyzed and a user interface for displaying
results.

Chapter 6 is dedicated to the description of a web based platform for such purpose. This kind of tools is not meant to be real time optimization. The computation technique and the accuracy of the sensors are still not yet up to such requirement. However, it is the direction to go for real time optimization adjustment. The development in this project is implementable in real buildings. It can be considered as a third party analysis in-depth platform for HES analysis, fine tuning of operational parameters, simulated real time display and terminal reporting. It is expected that it will become a third party tool interfacing the BMS vendor and the facility managers.

This research study established a genic protocol in assessing the electricity consumption of chiller plant within building. Moreover, in order to assess the chiller performance for optimization, an integrated HES approach based on data mining technique was introduced. Owing to assessing the characteristics of chiller, understanding the thermal behaviour of the refrigerant is essential. Chapter 2 presents a modified refrigerant model for refrigerant R134a. Chapter 3 introduces the utilization of the building chiller thermal performance lines based on building
energy thermal performance lines for the assessment of energy consumption due to chiller plant. Chapter 4 highlights the establishment and application of the data logic management protocol for individual chiller performance analysis while Chapter 5 introduces a new concept of HES approach based on data mining process for individual chiller optimization. With combining the function of the data logic management protocol and HES approach for individual chiller optimization, an electronic web-based integrated analysis scheme for chiller performance optimization is presented in Chapter 6.

7.1 SUMMARY OF SIGNIFICANT ACHIEVEMENTS

7.1.1 Development of Advanced Refrigerant Model (R134a)

In order to assessing the actual operating performance of a chiller, it is vitally important to understand the thermo-properties of the working fluid (i.e. refrigerant) at different working conditions within the refrigerant circuit of a chiller. In this study, refrigerant R134a was selected since this type of refrigerant has been commonly adopted in buildings in recently decades due to zero ozone depletion on the global environment. An advanced refrigerant model for predicting the thermo-properties of R134a under different working conditions was established. Moreover, the accuracy
of this model based on percentage of maximum error between the modeled data and lab data was also validated and the results showed that the accuracy of the model was significantly high.

7.1.2 Building Chiller Thermal Performance Lines Based on Building Energy Thermal Performance Lines

An energy signature reflecting the building energy performance by utilizing a thermal performance line based on statistical model with applications was introduced. With using this approach, the outdoor air condition influencing on the building energy consumption could be identified. In addition, the major functions of using this thermal performance line could be divided into two parts: (a) compare different types of buildings and (b) compare baseline for individual building. For former part, different types of building with different types of building configurations, in turns of window to wall ratio and U-value, could be identified and categories into different groups (including office building, commercial building, building complex, community hall, theatre, exhibition centre, institutional building and laboratory building) by demarcating their slopes of the thermal performance line model. 20 surveyed buildings were selected for the study.
Furthermore, in order to analyze the deterministic climatic variable to the building energy consumption, the climatic variables (i.e. $T_o$, $T_o^2$, $W_o$, $W_o^2$ and $T_o W_o$) were selected to determine their correlations to building energy consumption. With using thermo-dynamic model, the result revealed that $T_o W_o$ and $T_o^2$ are the best climatic variable that could be used to reflect how the weather-dependent load affecting the building energy consumption when applying thermal performance line model. In order to investigate the relationship between $T_o^2$ and $T_o W_o$, another study, with using the outdoor weather data provided by HKO with covering from Year 2002 to Year 2011, by correlating them was carried out. The correlation between them was extremely high enough. This provides a room to consider the priority of the selection of the climatic variables (i.e. $T_o^2$ and $T_o W_o$) for reflecting the weather-dependent loads of a building. For maintaining higher accuracy, using $T_o W_o$ is better. For convenience purpose, adopting $T_o^2$ is possible for presenting the weather-dependent loads while it will not violate the accuracy of the result.

With applying the thermal performance line model, 5 surveyed buildings out of 20 were selected for building electricity consumption analysis over the past three years (i.e. energy audit). The selected 5 surveyed buildings could be used for highlighting the typical cases in Hong Kong. With using the model, the changes of the weather
and non-weather dependent load over the past three years could be identified by using the slope and position of the thermal performance line model. When carrying out the building electricity consumption analysis with using thermal performance line model approach, the problems due to the collection of the essential building information encountered were also highlighted and discussed.

A case study on realizing the building energy signature with regard to the weather and non-weather dependent loads by using an engineering modeling approach with dissection of the building energy consumption was studied. In this study, eQUEST building energy performance tool was selected. The detailed energy breakdown due to different types of loads of a building could be identified. For weather dependent load, the result showed that the building electricity consumption due to cooling was significantly affected by $T_o^2$ and $T_oW_o$ following by AC pumps, heat rejection and AC fans as well as heating. Furthermore, the effects of components faults of the chillers with taking into account the changes of the climatic variable on the electricity consumption of the chiller plant within building by simulating artificial chiller component faults (including condenser fouling, refrigerant leakage, condensing water flow reduction and chilled water flow reduction) were also analyzed.
7.1.3 Data Acquisition System in Real Buildings for Individual Chiller Optimization

In order to assess or optimize the performance of a chiller, adequate sensing points and valid operating data of a chiller should be required. Nonetheless, it was discovered that the sensing points in most of the chiller plant in Hong Kong were insufficient. Apart from that, the operating data collected by log sheet or BMS server were normally subject to sensor faults or errors. Therefore, the phenomena were discussed in the significant of the data logic management protocol.

An established of the data logic management protocol with 5 steps (including data acquisition, data synchronization, data conditioning, range validity conditioning and uncertainty analysis) was introduced. The application of the data logic management protocol for two selected chiller plants (i.e. institutional building and commercial building) located in Hong Kong were presented.

7.1.4 HES Approach for Data Mining Process for Individual Chiller Optimization

Before introducing the HES approach for chiller performance optimization protocol, a chiller parametric analysis based on Simple plot was conducted. This analysis
enables that each controllable variable of a chiller operated over a period time could
be investigated. For controllable variables, the control settings of a chiller different
seasons could be clearly identified. In addition, the degradation of the performance
of a chiller via monitoring the variations of the KPIs of each component of a chiller
could also be analyzed.

Another parametric analysis based on chiller signature with utilizing different family
curves was also studied. This study provides a preliminary investigation on seeking
opportunities for individual chiller performance optimization. A sensitivity analysis
of both controllable variables and KPIs on the COP of a chiller was presented. The
most deterministic controllable variable and KPI of each chiller were identified.

Finally, an establishment of HES approach for chiller optimization was detailed
discussed. The main concept is to move the operating condition from low optimized
zone to high optimized zone. With applying this HES approach to chiller
optimization, the procedures for chiller HES optimized control are then presented.
Two chiller plants were selected for the case study.
7.1.5 Chiller Server Development with Applications

An establishment of chiller server with combining the functions of the data logic management protocol and chiller performance optimization scheme based on chiller parametric analysis approach was presented. This chiller server can be used as a commercial tool for monitoring and optimizing the actual performance of chillers. The functions of this chiller server includes: (a) time series for all operating variables, (b) monthly plot for COP against PLR of chillers, (c) energy efficiency signature for plotting family curves of chillers, (d) chiller optimal control set point for individual chiller optimization and (e) chiller model calibration. An application example of adopting the chiller server was also highlighted.
CHAPTER 7 CONCLUSIONS AND RECOMMENDATIONS

7.2 FUTURE RESEARCH

Major efforts of this thesis are made on the concept of the protocol for individual chiller optimization with using data mining technique. It would be very desirable and valuable to make further efforts on the following three aspects related to the research presented in this thesis.

7.2.1 Simulation Models for Common Types of Refrigerants Adopted in Different Types of Vapour Compression Chillers

In this thesis, a modified refrigerant model for simulating the thermal-dynamics characteristic of R134a under different working conditions was developed and presented. Since different types of chillers (including air-cooled and water-cooled chiller) may have adopting different types of refrigerants (for instances: R22, R407c, or R410a etc.). It is suggested to future develop the refrigerant model to simulate the thermal-dynamic performance of common types of refrigerants adopted in vapour compression chillers.

7.2.2 Chiller Plant Optimization with Using DLA and HES Approach

For the centralized chiller plant in Hong Kong, multiple-chiller type is commonly
adopted in most of buildings. Therefore, chiller plant optimization, with taking into consideration of multiple-chillers with their associated major equipment (i.e. chilled water pumps, condensing water pumps and cooling towers), by using DLA and HES approach can be further studied. The chiller plant optimization may include: (a) reset chiller water supply temperature, (b) reset condensing water return temperature, (c) optimized numbers of chilled water pumps operation or (d) optimized numbers of cooling tower fans / condensing water pumps operation etc. This recommendation can fill-in the gap between individual chiller and chiller plant for optimization analysis.

7.2.3 An Electronic Web-Based Integrated Chiller Plant Server for Chiller Plant Performance Optimization

As presented in Section 6, an electronic web-based integrated analysis scheme for chiller performance optimization (i.e. Chiller Server) is discussed. With combining the suggestion in Section 7.2.3, this Chiller Server can be further established with storing massive of operating data of the chiller plant for system optimization.
A.1 OVERVIEW ON INTERNATIONAL STANDARDS FOR STANDARD RATING CONDITIONS AND MINIMUM REQUIREMENT OF CHILLER EFFICIENCY

A.1.1 Hong Kong EMSD Building Energy Code 2012

Since 1998, the Electrical and Mechanical Department (EMSD) addressed a Code of Practice on Energy Efficiency of Air-Conditioning Installations (CoP for AC 1998) governing the rating conditions and the minimum requirement of the efficiency (i.e. Coefficient of Performance) of different types and sizes of chillers based on survey results and the Air-Conditioning and Refrigeration Association of Hong Kong [EMSD 1998]. This code focused on the full load operation of chillers while the part load operation of chillers was not considered as inadequate data were available from manufacturers in Hong Kong. This is because of the rapid change in technology and energy prices, the requirements stated in CoP for AC 1998 were stringent and therefore, the code was updated by two versions continuously (i.e. CoP for AC 2005
and CoP for AC 2007) [EMSD 2005; EMSD 2007]. At that time, this code was still a voluntary. In September 2012, the EMSD addressed a new code named “Code of Practice for Energy Efficiency of Building Services Installations (BEC 2012)”. This new code is under Part 9 of the Building Energy Efficiency Ordinance, Chapter 610 and this is a mandatory code. Now, the code is in effective [EMSD 2012].

A.1.2 USA AHRI Standard 551/591 (SI)

The two standards governing the performance/testing rating conditions for different types of chillers published by the Air-Conditioning and Refrigeration Institute (ARI) were ARI Standard 550-1992 “Centrifugal and Rotary Screw Water Chilling Packages” and ARI Standard 590-1992 “Positive Displacement Compressor Water Chilling Packages” [ARI 1992a; ARI 1992b]. These two standards took into consideration of full load and part load operation of chillers that introduced Application Part Load Value (APLV) and Integrated Part Load Value (IPLV). Indeed, APLV was firstly introduced in ARI Standard 550-1988 while IPLV was firstly launched in ARI Standard 550-1986 [Trane 1999]. In December 1998, ARI combined these two standards, ARI Standard 550-1992 and ARI Standard 590-1992, with some modifications (e.g. weighting factors for calculating IPLV etc.) that were superseded by ARI Standard 550/590-1998 “Standard for Water Chilling Packages
Using the Vapor Compression Cycle” [ARI 1998]. At the same time, APLV was deleted and was replaced by Non-Standard Part Load Value (NPLV).

Later on, ARI Standard 550/590-1998 was replaced and published by the Air-Conditioning, Heating and Refrigeration Institute (AHRI) with the new standard AHRI 550/590-2003 “Performance Rating of Water-Chilling Packages Using the Vapor Compression Cycle” [AHRI 2003]. Recently, this standard was updated to AHRI Standard 550/590-2011 for IP unit and AHRI Standard 551/591-2011 for SI unit. It is noted that most of the chiller manufacturers nowadays claim that products can meet the requirement of the standard and therefore, this standard becomes a base for justifying whether a single chiller is efficient or not. Moreover, no energy efficiency requirement of chillers is stated in this standard.
The equation of IPLV or NPLV is shown as:

\[
IPLV \text{ or } NPLV = 0.01 \ COP_{R \text{ at } 100\%} + 0.42 \ COP_{R \text{ at } 75\%} + 0.45 \ COP_{R \text{ at } 50\%} + 0.12 \ COP_{R \text{ at } 25\%}
\]  
(A.1)

where \( COP_{R \text{ at } 100\%} \) is the COP at 100% of chiller load; \( COP_{R \text{ at } 75\%} \) is the COP at 75% of chiller load; \( COP_{R \text{ at } 50\%} \) is the COP at 50% of chiller load; and \( COP_{R \text{ at } 25\%} \) is the COP at 25% of chiller load. The standard rating conditions of IPLV is summarized in Appendix A.2.1.
## Table A.1: Development of Part Load Value for Chiller Efficiency

<table>
<thead>
<tr>
<th>Chiller Efficiency Value</th>
<th>Description</th>
<th>Start from</th>
<th>Deleted from</th>
</tr>
</thead>
<tbody>
<tr>
<td>Application Part Load Value</td>
<td>It is a single number to act as part load efficiency indicator calculated using ARI method referenced to selected conditions.</td>
<td>ARI Standard 550-1998</td>
<td>ARI Standard 550/590-1998</td>
</tr>
<tr>
<td>Integrated Part Load Value</td>
<td>It is a single number to act as part load efficiency indicator calculated using ARI method at standard rating conditions.</td>
<td>ARI Standard 550-1986</td>
<td>-</td>
</tr>
<tr>
<td>Non-Standard Part Load Value</td>
<td>It is a single number to act as part load efficiency indicator calculated using the ARI Method referenced rating conditions other than ARI standard.</td>
<td>ARI Standard 550/590-1998 (replaced APLV)</td>
<td>-</td>
</tr>
</tbody>
</table>
A.1.3 USA ASHRAE Standard 90.1-2010

Originally, ASHRAE Standard 90 published in 1975 was a standard that provides minimum requirements for energy efficient designs for building except for low-rise residential buildings [WIKI ASHRAE STD 90.1; ASHRAE STD 90.1-2004]. This old ASHRAE Standard had multiple editions to it in years after. In 1999, the Board of Directors for ASHRAE voted and placed the standard on continuous maintenance which enables it to be updated multiple times in a year. As the advanced technology development and energy crisis concern, the ASHRAE 90.1 was first addressed and started in 2001. It has been updated in 2004, 2007 and 2010 [ASHRAE STD 90.1-2004; ASHRAE STD 90.1-2007; ASHRAE STD 90.1-2010]. It is noted that this standard considers both full load and part load operation of different types and sizes of chillers by COP and IPLV respectively. In ASHRAE Standard 90.1-2010, the minimum energy efficiency of chillers are firstly divided by two paths named Path A and Path B. For Path A, it is for governing the requirement of chillers with constant speed while Path B is for variable speed chillers.
A.1.4 Canada CSA-C743-02 2004

The Energy Efficiency Act passed in 1992 provided for the making and enforcement of regulations concerning minimum energy performance standards (MEPS) for energy-using products as well as the labeling of energy-using products and the collection of data [CSA 2002]. In 2003, Canada Standards Association (CSA) proposed MEPS, with incorporating the minimum requirement of COP and IPLV specified in ASHRAE Standard 90.1 2001 and the standard rating conditions stated in ARI 550/590-1998, for chillers that were intended for application in the air-conditioning of buildings. After that, the standard CSA-C753-02 “Performance Standard for Rating Packaged Water Chillers” was published [CSA 2002]. The latest standard was updated in October 2004. The standard is also under the Energy Efficiency Regulations in Canada.
A.1.5 Singapore SS530:2006

The first of the Singapore Standard “Code of Practice for Energy efficiency Standard for Building Services and Equipment” prepared by the Technical Committee on Building Services under the direction of the Construction Industry Practice Committee was published in 1982 [CP 24: 1999]. This standard was divided into three parts: CP 24: Part 1: 1982, Part 2: 1983 and Part 3: 1982. After seventeen years, the standard was revised to CP 24: 1999. In 2006, this standard was updated by the Working Group under the direction of the Technical Committee on Facilitates Management and has been renumbered as SS530: 2006 [SS530: 2006]. This standard specified the minimum efficiency requirements (including COP for full load and IPVL for part load operation) for water chilling packages (i.e. chillers) and the performance requirements for heat rejection equipment (i.e. cooling towers).

A.1.6 Chinese Taipei CNS 12575 2005 and CNS 12812 2005

The Bureau of Energy, Ministry of Economic Affairs (BEEA) has developed MEPS for a number of products. In most of cases, the energy tests are detailed in Chinese National Standards (CNS) of Chinese Taipei and the MEPS requirements are published by BEEA. Two mandatory standards for water commercial water chillers CNS 12575 (for Volumetric Water Chillers) and CNS 12812 (for Centrifugal Water
Chillers) were firstly introduced since 1989 and 1990 respectively [CNS 1989 and CNS 1990]. They covered minimum energy efficiency requirements of chillers in full load operation and testing standards (full load and part load operation) for chillers. These two standards were reviewed in 2003 (Phase 1) and 2005 (Phase 2). The most updated version of CN 12575 was named as Vapor Compression Water Chiller in February 2007 [TRACEA 2012].

A.1.7 Australia Greenhouse Office 2004

The first MEPS proposals for chillers published by the Australia government through the National Appliance and Equipment Energy Efficiency Program (NAEEEP) were in October 2004 [AGO 2004]. The detailed document can also be found from the Air-Conditioning and Refrigeration Equipment Manufacturers Association of Australia (AREMA). These proposals considered using the specifications in ASHRAE Standard 90.1-2004 [ASHRAE STD 90.1-2004]. The proposed minimum energy efficiency levels of the chillers were generally 15% higher than ASHRAE Standards 90.1-2004. Nevertheless, only the minimum energy efficiency of the chillers in full load operation (i.e. COP) was considered. After receiving the recommendations from the industry, consultation on and publication of the draft standard by Standards Australia were held and launched in 2005. The finalized
version of the standard with regulatory impact statement was introduced and undertaken starting from 2006. In 2007, the MEPS for chillers were enforced in the public.

A.1.8 China GB 19577-2004

The State General Administration of the People’s Republic of China for Quality Supervision and Inspection and Quarantine legislated and announced a new mandatory standard called “the minimum allowable values of the energy efficiency and energy efficiency grades for water chillers (GB 19577-2004) in 2003 and 2004 respectively [GB 19577-2004]. This standard has been implemented in March 2005. Only the full load operation (COP) of the chiller was considered. Apart from that, this standard considered water-cooled chillers and air-cooled/evaporative-cooled chillers only. With regard to the energy efficiency grades for water chillers, it classifies into give grades (i.e. from Grade 1 to Grade 5). Grade 1 is the most energy efficient requirements for chillers which is a target level [GB 50189-2005]. Grade 2 is the baseline level of the energy efficient product for chillers. Grade 3 and Grade 4 is the standard level of the energy efficiency requirements for chillers while Grade 5 is the products that will be phased-out in future. The standard rating conditions for chillers can be found from the national standard “The Methods of Performance Test.
for Positive Displacement and Centrifugal Water-Chilling Units and Heat Pump (GB/T10870)“.

A.1.9 European Union Eurovent 2005

Similar to Mainland China GB 19577-2004, the Eurovent introduced the minimum requirement of the energy efficiency (Energy Efficiency Ratio) for liquid chilling packages and heat pumps with seven classes (from Class A to Class G) [Eurovent 2005]. The requirement is under the supervision of the Eurovent Certification Programme. The highest energy efficiency level is represented by Class A while the lowest level is Class G. It is noted that the meaning of the Energy Efficiency Ratio (EER) is the same as the Coefficient of Performance (COP) of a chiller. Furthermore, Eurovent also studied the part load efficiency of different chillers. They introduced a term called “European Seasonal Energy Efficiency Ratio (ESEER)” for governing the part load efficiency of chillers in European countries. Generally, ESEER is equivalent to IPLV except the weighting factors. The study was completed in 2004 and the experimental applicable was started in 2005. Later, this requirement was mandatory implemented since June 2006 [Saheb et al. 2006].

The equation of ESEER is shown as:
\[ \text{ESEER} = 0.03 \text{EER}_{\text{at } 100\%} + 0.33 \text{EER}_{\text{at } 75\%} + 0.41 \text{EER}_{\text{at } 50\%} + 0.23 \text{EER}_{\text{at } 25\%} \]  
(A.2)

where \( \text{EER}_{\text{at } 100\%} \) is the EER at 100% of chiller load; \( \text{EER}_{\text{R at } 75\%} \) is the EER at 75% of chiller load; \( \text{EER}_{\text{at } 50\%} \) is the EER at 50% of chiller load; and \( \text{EER}_{\text{at } 25\%} \) is the EER at 25% of chiller load. The standard rating conditions of ESEER is summarized in Appendix A.2.1.

A.1.10 USA DOE 2003

The U.S. Department of Energy (DOE) launched a Federal Energy Management Program (FEMP) which provides guidelines on purchasing energy efficient products including chillers [DOE 2003a; DOE 2003b]. In this FEMP, air-cooled and water-cooled were taken into consideration. Recommended and Best Available IPLV at part load condition as well as COP at full load condition are specified. In general, it is not a mandatory program while it is to encourage the users to buy an energy efficient chiller for energy saving.
A.2 SUMMARY OF INTERNATIONAL STANDARDS FOR
STANDARD RATING CONDITIONS AND MINIMUM
REQUIREMENT OF CHILLER EFFICIENCY
## A.2.1 STANDARD RATING CONDITIONS FOR CHILLERS

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<td>7.0°C</td>
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<td>-</td>
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<td>≥ 1.0°C</td>
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<td>0.000018 m²/°C/W</td>
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<tr>
<td>Chilled Water Supply Temperature</td>
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<td>2.0°C to 16.0°C</td>
<td>7.0°C</td>
<td>7.0°C</td>
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<td>7.0°C</td>
<td>-</td>
<td>7.0°C</td>
<td>-</td>
</tr>
<tr>
<td>Chilled Water Return Temperature</td>
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<td>12.0°C</td>
<td>-</td>
<td>-</td>
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<td>-</td>
<td>12.0°C</td>
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<td>-</td>
<td>-</td>
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<td>-</td>
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<td>Condenser Water Inlet Temperature (Fresh Water)</td>
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</tr>
<tr>
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<td>-</td>
<td>-</td>
<td>30.0°C</td>
<td>≥ 0.5°C</td>
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<td>75% PLR (with weighting factor = 0.42 at 75% Part Load Ratio)</td>
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<td>-</td>
<td>19.0°C</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>23.0°C</td>
<td>≥ 0.5°C</td>
</tr>
<tr>
<td>25% PLR (with weighting factor = 0.12 at 25% Part Load Ratio)</td>
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<td>-</td>
<td>-</td>
<td>19.0°C</td>
<td>-</td>
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<td>-</td>
<td>19.0°C</td>
<td>≥ 0.5°C</td>
</tr>
<tr>
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<td>-</td>
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</tr>
<tr>
<td>Condenser Water Inlet Temperature (Sea Water)</td>
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<td>-</td>
<td>-</td>
<td>-</td>
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<tr>
<td>Evaporator Water Flow Rate</td>
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<td>0.0478 L/s/kW</td>
<td>-</td>
<td>0.0478 L/s/kW</td>
<td>0.0478 L/s/kW</td>
<td>-</td>
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<td>10.0 L/min/RT</td>
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<tr>
<td>Evaporator Flow Rate</td>
<td>-</td>
<td>Determined by Water Temperatures at Rated Capacity at Standard Rating Conditions only.</td>
<td>Determined by Water Temperatures at Rated Capacity at Standard Rating Conditions only.</td>
<td>Determined by Water Temperatures at Rated Capacity at Standard Rating Conditions only.</td>
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<td>-</td>
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<tr>
<td>Evaporator Fouling Factor (Water Side)</td>
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<td>0.000018 m²/°C/W</td>
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<td>0.000018 m²/°C/W</td>
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### Heat Rejection Requirement

|--------|-------------|---------------------------|-------------------------------|--------------------------------------|----------------------------------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|------------|

**Evaporatively-cooled**

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<th>Chilled Water Supply Temperature</th>
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<th>Condenser Inlet Ambient Dry-bulb Temperature</th>
<th>Condenser Inlet Ambient Wet-bulb Temperature</th>
<th>Temperature 100% PLR (with weighting factor = 0.01 at 100% Part Load Ratio)</th>
<th>75% PLR (with weighting factor = 0.42 at 75% Part Load Ratio)</th>
<th>50% PLR (with weighting factor = 0.45 at 50% Part Load Ratio)</th>
<th>25% PLR (with weighting factor = 0.12 at 25% Part Load Ratio)</th>
<th>Evaporator Water Flow Rate</th>
<th>Evaporator Fouling Factor (Water Side)</th>
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<td>7.0 °C</td>
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<td>3.0 °C</td>
<td>7.0 °C</td>
<td>No Rating Requirement</td>
<td>No Rating Requirement</td>
<td>No Rating Requirement</td>
<td>No Rating Requirement</td>
<td>0.0478 L/s/kW</td>
<td>0.000018 m²/°C/W</td>
</tr>
<tr>
<td>Requirement</td>
<td>12.0 °C</td>
<td>-</td>
<td>-</td>
<td>-</td>
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<td>-</td>
<td>-</td>
<td>0.0478 L/s/kW</td>
<td>0.000018 m²/°C/W</td>
</tr>
<tr>
<td>Requirement</td>
<td>35.0 °C</td>
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<td>24.0 °C</td>
<td>20.5 °C</td>
<td>17.0 °C</td>
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**Evaporative Water Flow Rate**

- 0.0478 L/s/kW
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
- 0.000018 m²/°C/W
## A.2.2 MINIMUM REQUIREMENT OF CHILLER EFFICIENCY

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<th>China</th>
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<td>All Capacities</td>
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<td>Water-cooled</td>
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<td>5.718</td>
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<td>≥ 1005 kW 5.547</td>
<td>5.901</td>
<td>5.504</td>
<td>7.815</td>
</tr>
<tr>
<td>-----------------------</td>
<td>-----------------</td>
<td>-------------------------</td>
<td>--------------------------</td>
<td>-----------------------------</td>
<td>--------------------------------------------</td>
</tr>
<tr>
<td></td>
<td>Cooling Capacity (kW)</td>
<td>Minimum COP</td>
<td>Cooling Capacity (kW)</td>
<td>Minimum COP</td>
<td>Minimum IPLV</td>
</tr>
<tr>
<td>Air-cooled</td>
<td>All Capacities</td>
<td>2.80</td>
<td>All Capacities</td>
<td>2.80</td>
<td>3.05</td>
</tr>
<tr>
<td>Scroll</td>
<td>All Capacities</td>
<td>2.80</td>
<td>All Capacities</td>
<td>2.80</td>
<td>3.05</td>
</tr>
<tr>
<td>Screw</td>
<td>All Capacities</td>
<td>2.80</td>
<td>All Capacities</td>
<td>2.80</td>
<td>3.05</td>
</tr>
<tr>
<td>Centrifugal</td>
<td>All Capacities</td>
<td>2.80</td>
<td>All Capacities</td>
<td>2.80</td>
<td>3.05</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Water-cooled</td>
<td>Reciprocating</td>
<td>4.20</td>
<td>All Capacities</td>
<td>4.20</td>
<td>5.05</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Scroll</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Screw</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Centrifugal</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

A-17
### B.1 BUILDING SITE LOCATION

#### Table B.1 Summary of Building Site Location

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Location</td>
<td>Hong Kong</td>
</tr>
<tr>
<td>Longitude</td>
<td>N 22° 17’</td>
</tr>
<tr>
<td>Latitude</td>
<td>E 144° 09’</td>
</tr>
<tr>
<td>Time Zone</td>
<td>2A</td>
</tr>
<tr>
<td>Nos. of Floor</td>
<td>GMT +8.0 Hours</td>
</tr>
<tr>
<td>Principle Energy Source</td>
<td>Electricity</td>
</tr>
<tr>
<td>Weather File</td>
<td>Hong Kong SAR 450070 (CityUHK), (Available at: <a href="http://apps1.eere.energy.gov/buildings/energyplus/cfm/weather_data3.cfm/region=2_asia_wmo_region_2/country=CHN/cname=China">http://apps1.eere.energy.gov/buildings/energyplus/cfm/weather_data3.cfm/region=2_asia_wmo_region_2/country=CHN/cname=China</a>)</td>
</tr>
<tr>
<td>Cooling Degree-Days</td>
<td>4782 Annual Cooling Degree-Days (10.0°C Baseline)</td>
</tr>
<tr>
<td>Heating Degree-Days</td>
<td>202 Annual Heating Degree-Days (18.0°C Baseline)</td>
</tr>
</tbody>
</table>
## B.2 BUILDING ENVELOPE COMPONENTS

### Table B.2 Summary of Building Envelope Components

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Roof</strong></td>
<td></td>
</tr>
<tr>
<td>1. Insulation Entirely Above Deck</td>
<td>U-Value = 0.368 W/m²/°C (Overall Heat Transfer Value)</td>
</tr>
<tr>
<td>2. 200 mm Light Weight Concrete</td>
<td>K Factor = 0.32 W/m²/°C (Thermal Conductivity)</td>
</tr>
<tr>
<td>3. 50 mm Concrete Thermal Insulation</td>
<td>K Factor = 0.026 W/m²/°C (Thermal Conductivity)</td>
</tr>
<tr>
<td>4. Roof Reflectivity</td>
<td>0.3 (with Green Roof)</td>
</tr>
<tr>
<td><strong>External Wall</strong></td>
<td></td>
</tr>
<tr>
<td>1. 150 mm Light Weight Concrete Wall</td>
<td>U-Value = 1.56 W/m²/°C (Overall Heat Transfer Value)</td>
</tr>
<tr>
<td>2. 50mm Concrete Thermal Insulation</td>
<td>K Factor = 0.32 W/m²/°C (Thermal Conductivity)</td>
</tr>
<tr>
<td>3. Metal Wall – for Semi-Retail and Podium Wall</td>
<td>U-Value = 2.0 W/m²/°C (Overall Heat Transfer Value)</td>
</tr>
<tr>
<td><strong>Window to Wall Ratio</strong></td>
<td></td>
</tr>
<tr>
<td>1. Office Tower</td>
<td>WWR = 0.55 (Wall Type I and 1A)</td>
</tr>
<tr>
<td>2. Semi-Retail</td>
<td>WWR = 0.45 (Wall Type 2)</td>
</tr>
<tr>
<td>3. Retail</td>
<td>WWR = 0.8 (Wall Type 4 for Sky Bridge) WWR = 0.8 (Wall Type 4A for Escalator Enclosure) WWR = 0.7 (Wall Type 6 for Podium Storefront)</td>
</tr>
<tr>
<td><strong>Glazing</strong></td>
<td></td>
</tr>
<tr>
<td>1. Vision Glazing</td>
<td>U-Value = 2.01 W/m²/°C, Shading Coefficient = 0.35, Visible Light Transmittance = 0.42 (Glazing Type 1 and 1A)</td>
</tr>
<tr>
<td>2. Daylighting Glazing</td>
<td>U-Value = 1.8 W/m²/°C, Shading Coefficient = 0.42, Visible Light Transmittance = 0.55 (Glazing Type 1 and 1A)</td>
</tr>
<tr>
<td>3. Semi-Retail</td>
<td>U-Value = 2.01 W/m²/°C, Shading Coefficient = 0.35, Visible Light Transmittance = 0.42 (Glazing Type 1 and 1A)</td>
</tr>
</tbody>
</table>

Note: Wall Type 3, 5, 7 and 8 are metal or concrete wall.
### Glazing

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glazing</td>
<td></td>
</tr>
<tr>
<td>4. Retail - Sky bridge</td>
<td>U-Value = 2.01 W/m²°C, Shading Coefficient = 0.35, Visible Light Transmittance = 0.42 (Wall Type 4)</td>
</tr>
<tr>
<td>5. Escalator Enclosure</td>
<td>U-Value = 5.5 W/m²°C, Shading Coefficient = 0.74, Visible Light Transmittance = 0.90 (Part A)</td>
</tr>
<tr>
<td></td>
<td>U-Value = 2.9 W/m²°C, Shading Coefficient = 0.70, Visible Light Transmittance = 0.84 (Part B)</td>
</tr>
</tbody>
</table>

### Escalator Enclosure

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Escalator Enclosure</td>
<td>U-Value = 5.5 W/m²°C, Shading Coefficient = 0.74, Visible Light Transmittance = 0.90 (Part A)</td>
</tr>
<tr>
<td></td>
<td>U-Value = 2.9 W/m²°C, Shading Coefficient = 0.70, Visible Light Transmittance = 0.84 (Part B)</td>
</tr>
</tbody>
</table>

### Podium Storefront

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Podium Storefront</td>
<td>U-Value = 2.9 W/m²°C, Shading Coefficient = 0.94, Visible Light Transmittance = 0.84</td>
</tr>
</tbody>
</table>

### Skylight

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Skylight</td>
<td></td>
</tr>
<tr>
<td>6. Skylight</td>
<td>U-Value = 2.62 W/m²°C, Shading Coefficient = 0.75, Visible Light Transmittance = 0.77 (11/F)</td>
</tr>
<tr>
<td></td>
<td>U-Value = 5.04 W/m²/°C, Shading Coefficient = 0.50, Visible Light Transmittance = 0.37 (16/F)</td>
</tr>
</tbody>
</table>

### Shading Device

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shading Device</td>
<td></td>
</tr>
<tr>
<td>1. Office Tower (for South and East Orientation)</td>
<td>Overhang with 400 mm</td>
</tr>
<tr>
<td>2. Office Tower (for North Orientation)</td>
<td>Overhang with 100 mm</td>
</tr>
<tr>
<td>3. Office Tower (for West Orientation)</td>
<td>Side Fins 300 mm</td>
</tr>
</tbody>
</table>

### Infiltration Rate

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Infiltration Rate</td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>2 L/s/m² of Glazing Area</td>
</tr>
</tbody>
</table>
## B.3 INDOOR DESIGN CONDITION

### Table B.3 Summary of Indoor Design Condition

<table>
<thead>
<tr>
<th>Zone/Space</th>
<th>Input Type</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Office</td>
<td>Indoor Air Dry-Bulb Temperature</td>
<td>23°C</td>
</tr>
<tr>
<td></td>
<td>Indoor Air Relative Humidity</td>
<td>60%</td>
</tr>
<tr>
<td></td>
<td>Indoor Occupancy Density</td>
<td>9 m² per person</td>
</tr>
<tr>
<td>Arcade</td>
<td>Indoor Air Dry-Bulb Temperature</td>
<td>22°C</td>
</tr>
<tr>
<td></td>
<td>Indoor Air Relative Humidity</td>
<td>60%</td>
</tr>
<tr>
<td></td>
<td>Indoor Occupancy Density</td>
<td>3 m² per person (B2/F and B1/F) or 4.5 m² per person</td>
</tr>
<tr>
<td>Retails</td>
<td>Indoor Air Dry-Bulb Temperature</td>
<td>22°C</td>
</tr>
<tr>
<td></td>
<td>Indoor Air Relative Humidity</td>
<td>60%</td>
</tr>
<tr>
<td></td>
<td>Indoor Occupancy Density</td>
<td>3 m² per person (B2/F and B1/F) or 4.5 m² per person</td>
</tr>
<tr>
<td>Food and Beverage</td>
<td>Indoor Air Dry-Bulb Temperature</td>
<td>22°C</td>
</tr>
<tr>
<td></td>
<td>Indoor Air Relative Humidity</td>
<td>60%</td>
</tr>
<tr>
<td></td>
<td>Indoor Occupancy Density</td>
<td>9 m² per person</td>
</tr>
<tr>
<td>Lobby</td>
<td>Indoor Air Dry-Bulb Temperature</td>
<td>23°C</td>
</tr>
<tr>
<td></td>
<td>Indoor Air Relative Humidity</td>
<td>60%</td>
</tr>
<tr>
<td></td>
<td>Indoor Occupancy Density</td>
<td>9 m² per person</td>
</tr>
<tr>
<td>Back of House (Office)</td>
<td>Indoor Air Dry-Bulb Temperature</td>
<td>24°C</td>
</tr>
<tr>
<td></td>
<td>Indoor Air Relative Humidity</td>
<td>60%</td>
</tr>
<tr>
<td></td>
<td>Indoor Occupancy Density</td>
<td>9 m² per person</td>
</tr>
</tbody>
</table>
## B.4 HVAC SYSTEM

### Table B.4 Summary of HVAC System

<table>
<thead>
<tr>
<th>Type</th>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air-Side System</td>
<td>Air-Conditioning System Type</td>
<td>1. <strong>Arcade</strong>&lt;br&gt;Constant Air Volume (CAV) with Variable Speed Drive (VSD) for Air-Handling Units (AHUs)&lt;br&gt;2. <strong>Office</strong>&lt;br&gt;VAV for AHUs (with electrical heating in perimeter zones of office floors)&lt;br&gt;3. <strong>Retail</strong>&lt;br&gt;Variable Air Volume (VAV) with VSD for Primary Air-Handling Unit (PAUs) + Fan Coil Units</td>
</tr>
<tr>
<td></td>
<td>Supply Air Fan Flow Rate</td>
<td>335 m³/s</td>
</tr>
<tr>
<td></td>
<td>Exhaust Air Fan Flow Rate</td>
<td>81 m³/s</td>
</tr>
<tr>
<td></td>
<td>Economizer Control</td>
<td>60% of Free Cooling under 18°C (for Office Floors)</td>
</tr>
<tr>
<td></td>
<td>Heat Recovery System</td>
<td>Total Enthalpy Wheel Effectiveness = 70% (for Office Floors)</td>
</tr>
</tbody>
</table>

Note: Demand Control Ventilation (DCV) is applied in Office and Arcade.

<table>
<thead>
<tr>
<th>Water-Side System</th>
<th>Chilled Water Circuit</th>
<th>Primary Variable Flow System</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Chilled Water Pump</td>
<td>1. <strong>Group 1</strong>&lt;br&gt;Pump Head = 31 m, Pump Flow = 133 L/s, Pump Efficiency = 83% (VSD), Quantity = 6&lt;br&gt;2. <strong>Group 2</strong>&lt;br&gt;Pump Head = 28.1 m, Pump Flow = 42 L/s, Pump Efficiency = 75.2% (VSD), Quantity = 2&lt;br&gt;3. <strong>Group 3</strong>&lt;br&gt;Pump Head = 19.2 m, Pump Flow = 63 L/s, Pump Efficiency = 83.6% (VSD), Quantity = 4</td>
</tr>
<tr>
<td></td>
<td>Condensing Water Pump</td>
<td>1. <strong>Group 1 and Group 2</strong>&lt;br&gt;Pump Head = 31 m, Pump Flow = 133 L/s, Pump Efficiency = 83% (VSD), Quantity = 12</td>
</tr>
</tbody>
</table>
### Chilled/Condenser Water Design

<table>
<thead>
<tr>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Chilled Water Temperature</td>
<td>Supply = 6.0°C, Return = 12.0°C</td>
</tr>
<tr>
<td>2. Condenser Water Temperature</td>
<td>Outlet = 37.0°C, Inlet = 32.0°C</td>
</tr>
</tbody>
</table>

Note: The supply air temperature after the cooling coil of the AHU is 13.0°C.

### Heat Rejection System

#### Cooling Tower

<table>
<thead>
<tr>
<th>Type</th>
<th>Input Item</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>VSD Control</td>
<td>Water-Cooled Centrifugal Chiller (with VSD)</td>
<td>Cooling Capacity = 3,340 kW, COP = 5.95, Quantity = 5</td>
</tr>
<tr>
<td></td>
<td>Oil-Free Water-Cooled Centrifugal Chiller (with VSD)</td>
<td>Cooling Capacity = 1055 kW, COP = 5.76, Quantity = 1</td>
</tr>
</tbody>
</table>
## B.5 LIGHTING SYSTEM

### Table B.5 Lighting System (Lighting Power Density)

<table>
<thead>
<tr>
<th>Type</th>
<th>Zone/Space</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Landlord Area</strong></td>
<td>Arcade</td>
<td>9.9 W/m²</td>
</tr>
<tr>
<td></td>
<td>Arcade Atrium</td>
<td>2.7 W/m²</td>
</tr>
<tr>
<td></td>
<td>Office Area</td>
<td>6.3 W/m²</td>
</tr>
<tr>
<td></td>
<td>Lobby (at B1/F)</td>
<td>11.8 W/m²</td>
</tr>
<tr>
<td></td>
<td>Lobby (at G/F)</td>
<td>12.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Lobby (at 1/F)</td>
<td>6.8 W/m²</td>
</tr>
<tr>
<td></td>
<td>Sky Lobby (at 9/F and 10/F)</td>
<td>11.3 W/m²</td>
</tr>
<tr>
<td></td>
<td>Lift Lobby (20/F to 38/F)</td>
<td>10.4 W/m²</td>
</tr>
<tr>
<td></td>
<td>Office Male Toilet</td>
<td>11.6 W/m²</td>
</tr>
<tr>
<td></td>
<td>Office Female Toilet</td>
<td>11.3 W/m²</td>
</tr>
<tr>
<td></td>
<td>Retail Toilet</td>
<td>13.1 W/m²</td>
</tr>
<tr>
<td><strong>Tenant Area</strong></td>
<td>Retail</td>
<td>18.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Food and Beverage</td>
<td>15.0 W/m²</td>
</tr>
<tr>
<td><strong>Public Area</strong></td>
<td>E&amp;M Plantroom</td>
<td>16.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Storage Room (Back of House)</td>
<td>9.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Refuge Room (Back of House)</td>
<td>9.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Stairs (Back of House)</td>
<td>6.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Corridor (Back of House)</td>
<td>5.0 W/m²</td>
</tr>
<tr>
<td><strong>Exterior Area</strong></td>
<td>Façade Light</td>
<td>41,378 W</td>
</tr>
<tr>
<td></td>
<td>Canopies and Overhangs</td>
<td>1,1680 W</td>
</tr>
<tr>
<td></td>
<td>Signage</td>
<td>27,950 W</td>
</tr>
</tbody>
</table>
B.6 EQUIPMENT LOADS

Table A.6 Equipment Loads (Power Density)

<table>
<thead>
<tr>
<th>Type</th>
<th>Zone/Space</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Small Power</td>
<td>Retail</td>
<td>15.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Office</td>
<td>20.0 W/m²</td>
</tr>
<tr>
<td></td>
<td>Other Landlord Area</td>
<td>10.0 W/m²</td>
</tr>
<tr>
<td>Equipment Power</td>
<td>Restaurant</td>
<td>30.0 W/m²</td>
</tr>
</tbody>
</table>

B.7 Operating Schedules

Table B.7 Operating Schedules

<table>
<thead>
<tr>
<th>Zone/Space</th>
<th>Business Hours/Operating Schedule</th>
<th>Input Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shopping Mall</td>
<td>Business Hours</td>
<td>08:00 – 23:00 (Mon to Sun)</td>
</tr>
<tr>
<td></td>
<td>HVAC System Operating Schedule</td>
<td>09:00 – 24:00 (Mon to Sun)</td>
</tr>
<tr>
<td>Shopping Mall Arcade</td>
<td>HVAC System Operating Schedule</td>
<td>08:00 – 23:00 (Mon to Sun)</td>
</tr>
<tr>
<td>Office</td>
<td>Business Hours</td>
<td>08:00 – 19:00 (Mon to Fri)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>08:00 – 14:00 (Sat)</td>
</tr>
<tr>
<td></td>
<td>HVAC System Operating Schedule</td>
<td>07:00 – 20:00 (Mon to Fri)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>07:00 – 14:00 (Sat)</td>
</tr>
</tbody>
</table>
C.1 DATA CONDITIONING (BEFORE AND AFTER PROCESSING) (CHILLER PLANT A)
Figure C.1.1 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-01)

Figure C.1.1 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-01)

Figure C.1.1 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-01)

Figure C.1.1 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-01)

Figure C.1.1 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-01)

Figure C.1.1 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-01)
Figure C.1.1 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-01)

Figure C.1.1 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-01)

Figure C.1.1 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-01)

Figure C.1.1 (j) Refrigerant Evaporating Profile after Data Conditioning (CH-01)

Figure C.1.1 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-01)

Figure C.1.1 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-01)
Figure C.1.1 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-01)

Figure C.1.1 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-01)

Figure C.1.1 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-01)

Figure C.1.1 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-01)

Figure C.1.1 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-01)

Figure C.1.1 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-01)
Figure C.1.1 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-01)

Figure C.1.1 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-01)

Figure C.1.1 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-01)

Figure C.1.1 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-01)

Figure C.1.1 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-01)

Figure C.1.1 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-01)
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Figure C.1.2 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-02)

Figure C.1.2 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-02)

Figure C.1.2 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-02)

Figure C.1.2 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-02)

Figure C.1.2 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-02)

Figure C.1.2 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-02)
(g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-02)

(h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-02)

(i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-02)

(j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-02)

(k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-02)

(l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-02)
Figure C.1.2 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-02)

Figure C.1.2 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-02)

Figure C.1.2 (o) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-02)

Figure C.1.2 (p) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-02)

Figure C.1.2 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-02)

Figure C.1.2 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-02)
Figure C.1.2 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-02)

Figure C.1.2 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-02)

Figure C.1.2 (u) Chilled Water Flow Rate Profile before Data Conditioning (CH-02)

Figure C.1.2 (v) Chilled Water Flow Rate Profile after Data Conditioning (CH-02)

Figure C.1.2 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-02)

Figure C.1.2 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-02)
Figure C.1.3 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-03)

Figure C.1.3 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-03)

Figure C.1.3 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-03)

Figure C.1.3 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-03)

Figure C.1.3 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-03)

Figure C.1.3 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-03)
Figure C.1.3 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-03)

Figure C.1.3 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-03)

Figure C.1.3 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-03)

Figure C.1.3 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-03)

Figure C.1.3 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-03)

Figure C.1.3 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-03)
Figure C.1.3 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-03)

Figure C.1.3 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-03)

Figure C.1.3 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-03)

Figure C.1.3 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-03)

Figure C.1.3 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-03)

Figure C.1.3 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-03)
Figure C.1.3 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-03)

Figure C.1.3 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-03)

Figure C.1.3 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-03)

Figure C.1.3 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-03)

Figure C.1.3 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-03)

Figure C.1.3 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-03)
Figure C.1.4 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-04)

Figure C.1.4 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-04)

Figure C.1.4 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-04)

Figure C.1.4 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-04)

Figure C.1.4 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-04)

Figure C.1.4 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-04)

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Figure C.1.4 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-04)

Figure C.1.4 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-04)

Figure C.1.4 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-04)

Figure C.1.4 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-04)

Figure C.1.4 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-04)

Figure C.1.4 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-04)
Figure C.1.4 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-04)

Figure C.1.4 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-04)

Figure C.1.4 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-04)

Figure C.1.4 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-04)

Figure C.1.4 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-04)

Figure C.1.4 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-04)
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Figure C.1.4 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-04)

Figure C.1.4 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-04)

Figure C.1.4 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-04)

Figure C.1.4 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-04)

Figure C.1.4 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-04)

Figure C.1.4 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-04)
Figure C.1.5 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-05)

Figure C.1.5 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-05)

Figure C.1.5 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-05)

Figure C.1.5 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-05)

Figure C.1.5 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-05)

Figure C.1.5 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-05)
Figure C.1.5 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-05)

Figure C.1.5 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-05)

Figure C.1.5 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-05)

Figure C.1.5 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-05)

Figure C.1.5 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-05)

Figure C.1.5 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-05)
Figure C.1.5 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-05)

Figure C.1.5 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-05)

Figure C.1.5 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-05)

Figure C.1.5 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-05)

Figure C.1.5 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-05)

Figure C.1.5 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-05)
Figure C.1.5 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-05)

Figure C.1.5 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-05)

Figure C.1.5 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-05)

Figure C.1.5 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-05)

Figure C.1.5 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-05)

Figure C.1.5 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-05)
C.2 DATA CONDITIONING (BEFORE AND AFTER PROCESSING) (CHILLER PLANT B)
Figure C.2.1 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-01)

Figure C.2.1 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-01)

Figure C.2.1 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-01)

Figure C.2.1 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-01)

Figure C.2.1 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-01)

Figure C.2.1 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-01)
Figure C.2.1 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-01)

Figure C.2.1 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-01)

Figure C.2.1 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-01)

Figure C.2.1 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-01)

Figure C.2.1 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-01)

Figure C.2.1 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-01)
Figure C.2.1 (m) Refrigerant Discharge Temperature
Profile before Data Conditioning (CH-01)

Figure C.2.1 (n) Refrigerant Condensing Temperature
Profile after Data Conditioning (CH-01)

Figure C.2.1 (o) Refrigerant Evaporating Pressure
Profile before Data Conditioning (CH-01)

Figure C.2.1 (p) Refrigerant Evaporating Pressure
Profile after Data Conditioning (CH-01)

Figure C.2.1 (q) Refrigerant Condensing Pressure
Profile before Data Conditioning (CH-01)

Figure C.2.1 (r) Refrigerant Condensing Pressure
Profile after Data Conditioning (CH-01)
Figure C.2.1 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-01)

Figure C.2.1 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-01)

Figure C.2.1 (u) Chilled Water Flow Rate Profile before Data Conditioning (CH-01)

Figure C.2.1 (v) Chilled Water Flow Rate Profile after Data Conditioning (CH-01)

Figure C.2.1 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-01)

Figure C.2.1 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-01)
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Figure C.2.2 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-02)

Figure C.2.2 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-02)

Figure C.2.2 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-02)

Figure C.2.2 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-02)

Figure C.2.2 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-02)
Figure C.2.2 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-02)

Figure C.2.2 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-02)

Figure C.2.2 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-02)

Figure C.2.2 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-02)

Figure C.2.2 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-02)

Figure C.2.2 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-02)
Figure C.2.2 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-02)

Figure C.2.2 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-02)

Figure C.2.2 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-02)

Figure C.2.2 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-02)

Figure C.2.2 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-02)

Figure C.2.2 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-02)
Figure C.2.2 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-02)

Figure C.2.2 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-02)

Figure C.2.2 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-02)

Figure C.2.2 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-02)

Figure C.2.2 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-02)

Figure C.2.2 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-02)
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Figure C.2.3 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-03)

Figure C.2.3 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-03)

Figure C.2.3 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-03)

Figure C.2.3 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-03)

Figure C.2.3 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-03)

Figure C.2.3 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-03)
Figure C.2.3 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-03)

Figure C.2.3 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-03)

Figure C.2.3 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-03)

Figure C.2.3 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-03)

Figure C.2.3 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-03)

Figure C.2.3 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-03)
Figure C.2.3 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-03)

Figure C.2.3 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-03)

Figure C.2.3 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-03)

Figure C.2.3 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-03)

Figure C.2.3 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-03)

Figure C.2.3 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-03)
Figure C.2.4 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-04)

Figure C.2.4 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-04)

Figure C.2.4 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-04)

Figure C.2.4 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-04)

Figure C.2.4 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-04)

Figure C.2.4 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-04)
Figure C.2.4 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-04)

Figure C.2.4 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-04)

Figure C.2.4 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-04)

Figure C.2.4 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-04)

Figure C.2.4 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-04)

Figure C.2.4 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-04)
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Figure C.2.4 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-04)

Figure C.2.4 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-04)

Figure C.2.4 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-04)

Figure C.2.4 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-04)

Figure C.2.4 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-04)

Figure C.2.4 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-04)
Figure C.2.4 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-04)

Figure C.2.4 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-04)

Figure C.2.4 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-04)

Figure C.2.4 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-04)

Figure C.2.4 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-04)

Figure C.2.4 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-04)
Figure C.2.5 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-05)

Figure C.2.5 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-05)

Figure C.2.5 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-05)

Figure C.2.5 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-05)

Figure C.2.5 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-05)

Figure C.2.5 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-05)
Figure C.2.5 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-05)

Figure C.2.5 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-05)

Figure C.2.5 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-05)

Figure C.2.5 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-05)

Figure C.2.5 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-05)

Figure C.2.5 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-05)
Figure C.2.5 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-05)

Figure C.2.5 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-05)

Figure C.2.5 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-05)

Figure C.2.5 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-05)

Figure C.2.5 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-05)

Figure C.2.5 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-05)
Figure C.2.5 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-05)

Figure C.2.5 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-05)

Figure C.2.5 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-05)

Figure C.2.5 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-05)

Figure C.2.5 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-05)

Figure C.2.5 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-05)
Figure C.2.6 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-06)

Figure C.2.6 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-06)

Figure C.2.6 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-06)

Figure C.2.6 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-06)

Figure C.2.6 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-06)

Figure C.2.6 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-06)
Figure C.2.6 (g) Condensing Water Inlet Temperature Profile before Data Conditioning (CH-06)

Figure C.2.6 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-06)

Figure C.2.6 (i) Refrigerant Evaporating Temperature Profile before Data Conditioning (CH-06)

Figure C.2.6 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-06)

Figure C.2.6 (k) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-06)

Figure C.2.6 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-06)
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Figure C.2.6 (m) Refrigerant Condensing Temperature Profile before Data Conditioning (CH-06)

Figure C.2.6 (n) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-06)

Figure C.2.6 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-06)

Figure C.2.6 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-06)

Figure C.2.6 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-06)

Figure C.2.6 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-06)
Figure C.2.6 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-06)

Figure C.2.6 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-06)

Figure C.2.6 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-06)

Figure C.2.6 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-06)

Figure C.2.6 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-06)

Figure C.2.6 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-06)
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Figure C.2.7 (a) Chilled Water Supply Temperature Profile before Data Conditioning (CH-07)

Figure C.2.7 (b) Chilled Water Supply Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (c) Chilled Water Return Temperature Profile before Data Conditioning (CH-07)

Figure C.2.7 (d) Chilled Water Return Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (e) Condensing Water Outlet Temperature Profile before Data Conditioning (CH-07)

Figure C.2.7 (f) Condensing Water Outlet Temperature Profile after Data Conditioning (CH-07)
Figure C.2.7 (g) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (h) Condensing Water Inlet Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (i) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (j) Refrigerant Evaporating Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (k) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (l) Refrigerant Condensing Temperature Profile after Data Conditioning (CH-07)
Figure C.2.7 (m) Refrigerant Discharge Temperature Profile before Data Conditioning (CH-07)

Figure C.2.7 (n) Refrigerant Discharge Temperature Profile after Data Conditioning (CH-07)

Figure C.2.7 (o) Refrigerant Evaporating Pressure Profile before Data Conditioning (CH-07)

Figure C.2.7 (p) Refrigerant Evaporating Pressure Profile after Data Conditioning (CH-07)

Figure C.2.7 (q) Refrigerant Condensing Pressure Profile before Data Conditioning (CH-07)

Figure C.2.7 (r) Refrigerant Condensing Pressure Profile after Data Conditioning (CH-07)
Figure C.2.7 (s) Chilled Water Flow Rate Profile before Data Conditioning (CH-07)

Figure C.2.7 (t) Chilled Water Flow Rate Profile after Data Conditioning (CH-07)

Figure C.2.7 (u) Condensing Water Flow Rate Profile before Data Conditioning (CH-07)

Figure C.2.7 (v) Condensing Water Flow Rate Profile after Data Conditioning (CH-07)

Figure C.2.7 (w) Chiller Electricity Power Input Profile before Data Conditioning (CH-07)

Figure C.2.7 (x) Chiller Electricity Power Input Profile after Data Conditioning (CH-07)
D.1 RANGE VALIDITY CONDITIONING (CHILLER PLANT A)
APPENDIX D

Figure D.1.1 (a) Frequency Distribution Plot for Chilled Water Supply Temperature (CH-01)

Figure D.1.1 (b) Frequency Distribution Plot for Chilled Water Return Temperature (CH-01)

Figure D.1.1 (c) Frequency Distribution Plot for Condensing Water Outlet Temperature (CH-01)

Figure D.1.1 (d) Frequency Distribution Plot for Condensing Water Inlet Temperature (CH-01)

Figure D.1.1 (e) Frequency Distribution Plot for Refrigerant Evaporating Temperature (CH-01)

Figure D.1.1 (f) Frequency Distribution Plot for Refrigerant Condensing Temperature (CH-01)
Figure D.1.1 (g) Frequency Distribution Plot for Refrigerant Discharge Temperature (CH-01)

Figure D.1.1 (h) Frequency Distribution Plot for Refrigerant Evaporating Pressure (CH-01)

Figure D.1.1 (i) Frequency Distribution Plot for Refrigerant Condensing Pressure (CH-01)

Figure D.1.1 (j) Frequency Distribution Plot for Chilled Water Flow Rate (CH-01)

Figure D.1.1 (k) Frequency Distribution Plot for Condensing Water Flow Rate (CH-01)

Figure D.1.1 (l) Frequency Distribution Plot for Chiller Power Input (CH-01)
Figure D.1.2 (a) Frequency Distribution Plot for Chilled Water Supply Temperature (CH-02)

Figure D.1.2 (b) Frequency Distribution Plot for Chilled Water Return Temperature (CH-02)

Figure D.1.2 (c) Frequency Distribution Plot for Condensing Water Outlet Temperature (CH-02)

Figure D.1.2 (d) Frequency Distribution Plot for Condensing Water Inlet Temperature (CH-02)

Figure D.1.2 (e) Frequency Distribution Plot for Refrigerant Evaporating Temperature (CH-02)

Figure D.1.2 (f) Frequency Distribution Plot for Refrigerant Condensing Temperature (CH-02)
Figure D.1.2 (g) Frequency Distribution Plot for Refrigerant Discharge Temperature (CH-02)

Figure D.1.2 (h) Frequency Distribution Plot for Refrigerant Evaporating Pressure (CH-02)

Figure D.1.2 (i) Frequency Distribution Plot for Refrigerant Condensing Pressure (CH-02)

Figure D.1.2 (j) Frequency Distribution Plot for Chilled Water Flow Rate (CH-02)

Figure D.1.2 (k) Frequency Distribution Plot for Condensing Water Flow Rate (CH-02)

Figure D.1.2 (l) Frequency Distribution Plot for Chiller Power Input (CH-02)
Figure D.1.3 (a) Frequency Distribution Plot for Chilled Water Supply Temperature (CH-03)

Figure D.1.3 (b) Frequency Distribution Plot for Chilled Water Return Temperature (CH-03)

Figure D.1.3 (c) Frequency Distribution Plot for Condensing Water Outlet Temperature (CH-03)

Figure D.1.3 (d) Frequency Distribution Plot for Condensing Water Inlet Temperature (CH-03)

Figure D.1.3 (e) Frequency Distribution Plot for Refrigerant Evaporating Temperature (CH-03)

Figure D.1.3 (f) Frequency Distribution Plot for Refrigerant Condensing Temperature (CH-03)
Figure D.1.3 (g) Frequency Distribution Plot for Refrigerant Discharge Temperature (CH-03)

Figure D.1.3 (h) Frequency Distribution Plot for Refrigerant Evaporating Pressure (CH-03)

Figure D.1.3 (i) Frequency Distribution Plot for Refrigerant Condensing Pressure (CH-03)

Figure D.1.3 (j) Frequency Distribution Plot for Chilled Water Flow Rate (CH-03)

Figure D.1.3 (k) Frequency Distribution Plot for Condensing Water Flow Rate (CH-03)

Figure D.1.3 (l) Frequency Distribution Plot for Chiller Power Input (CH-03)
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Figure D.1.4 (b) Frequency Distribution Plot for Chilled Water Return Temperature (CH-04)

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Figure E.5.1 Time Series Plot of Compressor Polytropic Efficiency

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(CHILLER PLANT A AND CHILLER PLANT B)
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Chiller Plant B

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Figure E.11.1 (c) Chiller Efficiency Signature with Condensing Water Inlet Temperature Binned (CH-03)

Figure E.11.1 (d) Chiller Efficiency Signature with Condensing Water Inlet Temperature Binned (CH-04)

Figure E.11.1 (e) Chiller Efficiency Signature with Condensing Water Inlet Temperature Binned (CH-05)


Chiller Plant B

Figure E.11.2 (a) Chiller Efficiency Signature with Condensing Water Inlet Temperature Binned (CH-01)

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Figure E.11.2 (c) Chiller Efficiency Signature with Condensing Water Inlet Temperature Binned (CH-03)

Figure E.11.2 (d) Chiller Efficiency Signature with Condensing Water Inlet Temperature Binned (CH-04)

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OVERALL HEAT TRANSFER VALUE BINNED

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Figure E.14.2 (b) Chiller Efficiency Signature with Evaporator Overall Heat Transfer Value Binned (CH-02)

Figure E.14.2 (c) Chiller Efficiency Signature with Evaporator Overall Heat Transfer Value Binned (CH-03)

Figure E.14.2 (d) Chiller Efficiency Signature with Evaporator Overall Heat Transfer Value Binned (CH-04)
Figure E.14.2 (e) Chiller Efficiency Signature with Evaporator Overall Heat Transfer Value Binned (CH-05)

Figure E.14.2 (f) Chiller Efficiency Signature with Evaporator Overall Heat Transfer Value Binned (CH-06)

Figure E.14.2 (g) Chiller Efficiency Signature with Evaporator Overall Heat Transfer Value Binned (CH-07)
E.15 CHILLER EFFICIENCY SIGNATURE – CONDENSER

OVERALL HEAT TRANSFER VALUE BINNED

(CHILLER PLANT A AND CHILLER PLANT B)
Chiller Plant A

Figure E.15.1 (a) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-01)

Figure E.15.1 (b) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-02)

Figure E.15.1 (c) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-03)

Figure E.15.1 (d) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-04)

Figure E.15.1 (e) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-05)
Chiller Plant B

Figure E.15.2 (a) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-01)

Figure E.15.2 (b) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-02)

Figure E.15.2 (c) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-03)

Figure E.15.2 (d) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-04)
Figure E.15.2 (e) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-05)

Figure E.15.2 (f) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-06)

Figure E.15.2 (g) Chiller Efficiency Signature with Condenser Overall Heat Transfer Value Binned (CH-07)
E.16  CHILLER EFFICIENCY SIGNATURE – COMPRESSOR POLYTROPIC EFFICIENCY BINNED (CHILLER PLANT A AND CHILLER PLANT B)


**Chiller Plant A**

![Figure E.16.1 (a) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-01)](image)

![Figure E.16.1 (b) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-02)](image)

![Figure E.16.1 (c) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-03)](image)

![Figure E.16.1 (d) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-04)](image)

![Figure E.16.1 (e) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-05)](image)
**Chiller Plant B**

Figure E.16.2 (a) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-01)

Figure E.16.2 (b) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-02)

Figure E.16.2 (c) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-03)

Figure E.16.2 (d) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-04)

Figure E.16.2 (e) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-05)

Figure E.16.2 (f) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-06)
Figure E.16.2 (g) Chiller Efficiency Signature with Compressor Polytropic Efficiency Binned (CH-07)
E.17 CHILLER EFFICIENCY SIGNATURE – ELECTRO-MECHANICAL LOSS EFFICIENCY BINNED (CHILLER PLANT A AND CHILLER PLANT B)
Chiller Plant A

Figure E.17.1 (a) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-01)

Figure E.17.1 (b) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-02)

Figure E.17.1 (c) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-03)

Figure E.17.1 (d) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-04)

Figure E.17.1 (e) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-05)
**Chiller Plant B**

- Figure E.17.2 (a) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-01)
- Figure E.17.2 (b) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-02)
- Figure E.17.2 (c) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-03)
- Figure E.17.2 (d) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-04)
- Figure E.17.2 (e) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-05)
- Figure E.17.2 (f) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-06)
Figure E.17.2 (g) Chiller Efficiency Signature with Electro-Mechanical Loss Efficiency Binned (CH-07)
E.18 CHILLER EFFICIENCY SIGNATURE – REFRIGERANT
MASS FLOW RATE BINNED (CHILLER PLANT A AND
CHILLER PLANT B)
**Chiller Plant A**

Figure E.18.1 (a) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-01)

Figure E.18.1 (b) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-02)

Figure E.18.1 (c) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-03)

Figure E.18.1 (d) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-04)

Figure E.18.1 (e) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-05)
**Chiller Plant B**

Figure E.18.2 (a) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-01)

Figure E.18.2 (b) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-02)

Figure E.18.2 (c) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-03)

Figure E.18.2 (d) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-04)

Figure E.18.2 (e) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-05)

Figure E.18.2 (f) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-06)
Figure E.18.2 (g) Chiller Efficiency Signature with Refrigerant Mass Flow Rate Binned (CH-07)
E.19  CHILLER EFFICIENCY SIGNATURE – EXPANSION

DEVICE DISCHARGE COEFFICIENT BINNED

(Chiller Plant A and Chiller Plant B)
Chiller Plant A

Figure E.19.1 (a) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-01)

Figure E.19.1 (b) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-02)

Figure E.19.1 (c) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-03)

Figure E.19.1 (d) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-04)

Figure E.19.1 (e) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-05)
**Chiller Plant B**

Figure E.19.2 (a) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-01)

Figure E.19.2 (b) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-02)

Figure E.19.2 (c) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-03)

Figure E.19.2 (d) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-04)
Figure E.19.2 (e) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-05)

Figure E.19.2 (f) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-06)

Figure E.19.2 (g) Chiller Efficiency Signature with Expansion Device Discharge Coefficient Binned (CH-07)
E.20 OPTIMIZATION MAPS AND PREVAILING DISTRIBUTION GRAPHS FOR EACH CONTROLLABLE VARIABLE (CHILLER PLANT A)
Figure E.20.1 (a) Chiller Optimization Map before Transformation (CH-01)

Figure E.20.1 (b) Chiller Optimization Map after Transformation (CH-01)

Figure E.20.1 (c) Chilled Water Supply Temperature under Different Chiller Optimization Zones (CH-01)

Figure E.20.1 (d) Condensing Water Inlet Temperature under Different Chiller Optimization Zones (CH-01)

Figure E.20.1 (e) Part Load Ratio under Different Chiller Optimization Zones (CH-01)
Figure E.20.2 (a) Chiller Optimization Map before Transformation (CH-02)

Figure E.20.2 (b) Chiller Optimization Map after Transformation (CH-02)

Figure E.20.2 (c) Chilled Water Supply Temperature under Different Chiller Optimization Zones (CH-02)

Figure E.20.2 (d) Condensing Water Inlet Temperature under Different Chiller Optimization Zones (CH-02)

Figure E.20.2 (e) Part Load Ratio under Different Chiller Optimization Zones (CH-02)
Figure E.20.3 (a) Chiller Optimization Map before Transformation (CH-03)

Figure E.20.3 (b) Chiller Optimization Map after Transformation (CH-03)

Figure E.20.3 (c) Chilled Water Supply Temperature under Different Chiller Optimization Zones (CH-03)

Figure E.20.3 (d) Condensing Water Inlet Temperature under Different Chiller Optimization Zones (CH-03)

Figure E.20.3 (e) Part Load Ratio under Different Chiller Optimization Zones (CH-03)
Figure E.20.4 (a) Chiller Optimization Map before Transformation (CH-04)

Figure E.20.4 (b) Chiller Optimization Map after Transformation (CH-04)

Figure E.20.4 (c) Chilled Water Supply Temperature under Different Chiller Optimization Zones (CH-04)

Figure E.20.4 (d) Condensing Water Inlet Temperature under Different Chiller Optimization Zones (CH-04)

Figure E.20.4 (e) Part Load Ratio under Different Chiller Optimization Zones (CH-04)
Figure E.20.5 (a) Chiller Optimization Map before Transformation (CH-05)

Figure E.20.5 (b) Chiller Optimization Map after Transformation (CH-05)

Figure E.20.5 (c) Chilled Water Supply Temperature under Different Chiller Optimization Zones (CH-05)

Figure E.20.5 (d) Condensing Water Inlet Temperature under Different Chiller Optimization Zones (CH-05)

Figure E.20.5 (e) Part Load Ratio under Different Chiller Optimization Zones (CH-05)
E.21 OPTIMIZATION MAPS AND PREVAILING DISTRIBUTION GRAPHS FOR EACH CONTROLLABLE VARIABLE (CHILLER PLANT B)
Figure E.21.1 (a) Chiller Optimization Map before Transformation (CH-01)

Figure E.21.1 (b) Chiller Optimization Map after Transformation (CH-01)

Figure E.21.1 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-01)

Figure E.21.1 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-01)

Figure E.21.1 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-01)

Figure E.21.1 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-01)
Figure E.21.1 (g) Part Load Ratio under Different Chiller Optimization Zones (CH-01)
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Figure E.21.2 (a) Chiller Efficacy Zones Identification before Transformation (CH-02)

Figure E.21.2 (b) Chiller Efficacy Zones Identification after Transformation (CH-02)

Figure E.21.2 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-02)

Figure E.21.2 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-02)

Figure E.21.2 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-02)

Figure E.21.2 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-02)
Figure E.21.2 (g) Part Load Ratio under Different Chiller Efficacy Zones (CH-02)
Figure E.21.3 (a) Chiller Efficacy Zones Identification before Transformation (CH-03)

Figure E.21.3 (b) Chiller Efficacy Zones Identification after Transformation (CH-03)

Figure E.21.3 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-03)

Figure E.21.3 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-03)

Figure E.21.3 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-03)

Figure E.21.3 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-03)
Figure E.21.3 (g) Part Load Ratio under Different Chiller Efficacy Zones (CH-03)
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Figure E.21.4 (a) Chiller Efficacy Zones Identification before Transformation (CH-04)

Figure E.21.4 (b) Chiller Efficacy Zones Identification after Transformation (CH-04)

Figure E.21.4 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-04)

Figure E.21.4 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-04)

Figure E.21.4 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-04)

Figure E.21.4 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-04)
Figure E.21.4 (g) Part Load Ratio under Different Chiller Efficacy Zones (CH-04)
Figure E.21.5 (a) Chiller Efficacy Zones Identification before Transformation (CH-05)

Figure E.21.5 (b) Chiller Efficacy Zones Identification after Transformation (CH-05)

Figure E.21.5 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-05)

Figure E.21.5 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-05)

Figure E.21.5 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-05)

Figure E.21.5 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-05)
Figure E.21.5 (g) Part Load Ratio under Different Chiller Efficacy Zones (CH-05)
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Figure E.21.6 (b) Chiller Efficacy Zones Identification after Transformation (CH-06)

Figure E.21.6 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-06)

Figure E.21.6 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-06)

Figure E.21.6 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-06)

Figure E.21.6 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-06)
Figure E.21.6 (g) Part Load Ratio under Different Chiller Efficacy Zones (CH-06)
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Figure E.21.7 (a) Chiller Efficacy Zones Identification before Transformation (CH-07)

Figure E.21.7 (b) Chiller Efficacy Zones Identification after Transformation (CH-07)

Figure E.21.7 (c) Chilled Water Supply Temperature under Different Chiller Efficacy Zones (CH-07)

Figure E.21.7 (d) Condensing Water Inlet Temperature under Different Chiller Efficacy Zones (CH-07)

Figure E.21.7 (e) Chilled Water Flow Rate under Different Chiller Efficacy Zones (CH-07)

Figure E.21.7 (f) Condensing Water Flow Rate under Different Chiller Efficacy Zones (CH-07)
Figure E.21.7 (g) Part Load Ratio under Different Chiller Efficacy Zones (CH-07)
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