Optimization Strategy & Energy Savings Of
Water-Cooled Chiller Plant System

by

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CERTIFICATION OF APPROVAL

OPTIMIZATION STRATEGY & ENERGY SAVINGS OF WATER-COOLED CHILLER PLANT SYSTEM

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Approved by:

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(Dr. Aklilu Tesfamichael Baheta)

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BANDAR SRI ISKANDAR, PERAK
January 2015

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

JOSH WONG SHAO ZHE

ABSTRACT

Water-cooled chiller cooling systems are commonly used for air conditioning purposes in tropical countries like Malaysia. Chillers need to be optimized to increase the performance of the system and achieve energy savings. Understanding the interaction between each sub systems and exploiting their strength is the key to system optimization. The significance of this study is to provide more insight on how to optimize the water-cooled chiller system. For that, a thermodynamic model is developed to evaluate the energy saving opportunities in the system. Using an iterative procedure on finding the thermodynamic properties and equations of the system, data on the performance and energy savings of the system can be calculated for every part load conditions. From the results of the paper, there is improvement in the performance of the chiller plant system when variable speed pumps are used. Using a cooling load profile of a typical office building, electricity can be saved up to 80% for pump power and 18% for the total plant energy usage. Application of variable speed pump in the system can save significant amount of energy that conventional chiller plant system used in excess nowadays.

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CHAPTER 1: INTRODUCTION

1.1 Background of Study

In Malaysia, where tropical and humid weather are common, air conditioning within a building is important so that occupants are comfortable. When we can resort to air-cooled units to cool small spaces, in larger infrastructures however, the cooling capacity may easily increase by 10 times (Tracey, 2010). The cost of air cooled unit for such an enormous cooling capacity cannot be met, thus, chillers comes in to meet larger cooling capacity at lower cost.

Chiller systems are an integral part of a HVAC system and are long being used to provide cooling purposes through centralised air conditioning. There are mainly 2 types of chillers, air-cooled and water-cooled. The difference between both types basically determines how heat is rejected from the system. It is proven that water-cooled chillers have better efficiency and longer operating live compared to its counterpart since it uses water to reject the heat out of the system rather than air which has a lower specific heat capacity. Therefore, we will be focusing mainly on water-cooled chiller that uses a cooling tower to reject heat from water.

The conventional water-cooled chillers mainly comprise of three cycles namely the refrigeration cycle, the cooling water loop and the chilled water loop. The whole system works together as one under the vapour compression cycle which uses refrigerant as the working fluid. (MNSU)

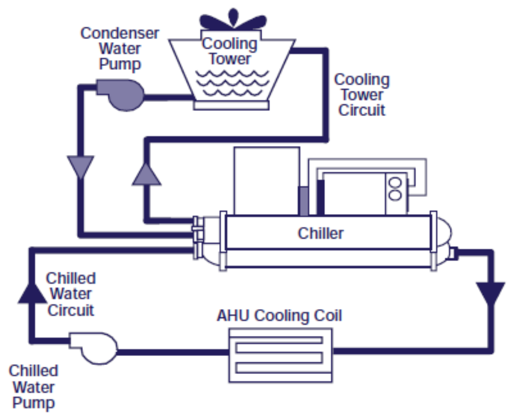

FIGURE 1.1: Simple schematic diagram of a water chiller system

Not only common in high rise office buildings, chillers are also used in the industry to control cooling of mechanisms, processes, products or machineries. Industrial chillers cover a wide range of industries from food and beverage processing, to manufacturing processes or even the semi-conductor industry. Therefore, the importance of optimizing a chiller system that can save energy and cost has to be stressed on to provide better solutions for air conditioning and industrial demands.

1.1.1 Working Principle of Water-Cooled Chiller

In a water-cooled chiller, the vapour compression refrigeration cycle is the core of the system. In the cycle, there are four primary components which consist of the compressor, evaporator, condenser and an expansion device.

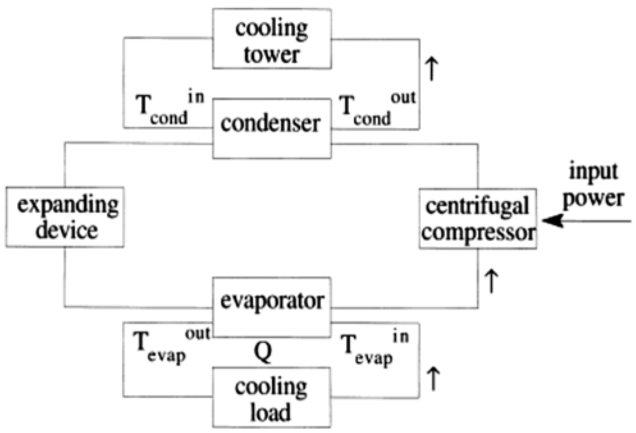

FIGURE 1.2: Breakdown of water chiller into 4 equipment and relation with other cycles.

The refrigerant first enters the compressor, which increase the pressure of the refrigerant through isentropic compression. Upon exiting the compressor, the refrigerant is superheated which passes on to the condenser. The condenser cools the incoming refrigerant, where heat is rejected from the refrigerant to the cooling water loop. The refrigerant is cooled to its saturated temperature upon exiting the condenser and moves on to the expansion valve. The expansion device controls the refrigerant flow which throttles the flow and causes drop in the pressure and evaporation of the refrigerant. At the evaporator, heat is drawn from the chilled water loop to the refrigerant. The low pressure refrigerant evaporates completely after the evaporator and continues on to the compressor to regain its pressure thus repeating the refrigeration cycle. (Cengel & Boles, 2008)

The chilled water loop transport water from the evaporator to the air handlers in the building. After releasing heat at the evaporator, the lower temperature chilled water is transported to air handling units where air is cooled by passing them through the chilled water coils. On the other hand, the cooling water loop connects the condenser to the cooling tower. The condenser water after receiving unwanted heat from the refrigerant is then transported to the cooling tower where water is cooled before circulating back to the condenser.

1.1.2 Energy Situation in Malaysia

From the study done by Saidur (2009) on the energy consumption in buildings in Malaysia, he stated that the energy consumed in recent years has increased dramatically and will continue its increasing trend due to high economic growth in the country. The demand of energy grows rapidly with the increase in higher standard living not only in developing countries like Malaysia, but also developed countries.

According to statistics from Electrical Supply in Malaysia, the total commercial energy consumption has reached a high of 29000GWh in the year 2006. (EC, 2007). Figure 1.3 shows the energy consumption in the commercial sector from the year 2003 to 2006. These increasing trend of energy usage have to be countered with efficient energy savings solutions so that energy consumptions can be reduced and following a further reduction of carbon emission to the atmosphere.

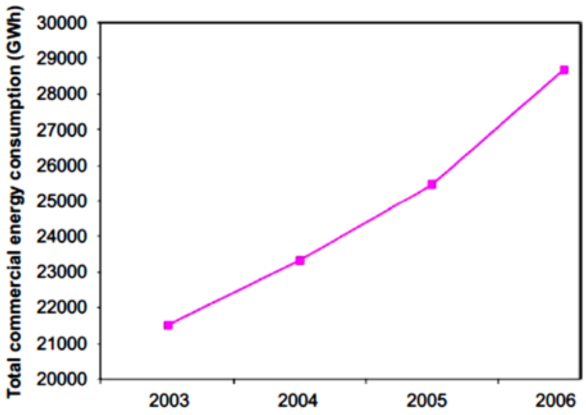

FIGURE 1.3: The total commercial energy consumption in Malaysia from 2003 to 2006 (EC, 2007)

1.2 Problem Statement

HVAC systems accounts for almost half of the total electrical energy consumption in a typical commercial building with chillers being the major energy consumer with approximately 40% of total energy consumption in a typical building. (Jayamaha, 2007) Companies have to bear high operation cost of conventional chiller systems due to its high electrical power demand.

Moreover, water chiller plant systems are designed to operate at full load and constant speed. However, it is rarely the case that they operate in full load conditions since the cooling load throughout a typical day is not always at the extreme side. Therefore, chiller systems are actually sapping unnecessary power most of the time when running at full load efficiency.

Thus, optimizing the performance and efficiency of a chiller system is essential to reduce energy consumption and operation cost.

1.3 Objectives and Scope of Study

The objective of the project is to suggest optimization strategy of a water-chiller plant system and to analyse the energy saving of the system with the optimization strategy.

The scope of study is to propose optimization strategies to save energy in water chiller plant system under the following conditions:

- Steady state flow in the chiller plant system
- Electrical powered chiller plant system
- Strategy to utilise pumps as energy saving equipment
- R134a as working fluid for vapour compression refrigeration cycle.
- Ideal working conditions

CHAPTER 2: LITERATURE REVIEW AND THEORY

2.1 Efficiency of Chiller Plant System

The power usage of chiller is dominant in the whole system. According to Neidlinger (2013), chillers consume over 50 percent of electrical power in a typical period of time of building use. He continues to add on that in the U.S, more than 120,000 chillers are using excess of 30% energy to operate inefficiently. (Waste Reduction Partners Org, 2010)

The conventional way is to run chiller at full load since existing chillers with constant speed compressors normally reaches maximum performance at full load. However, implementing a full load operation does not aid in reducing the power consumption. Chillers are usually selected based on the design rated cooling capability, but they most rarely operate in this condition. (Energy Design Resources, 2010)

Based on statistics, the peak cooling capacity of a typical office building is only a few hours per year. Figure 2.1 shows that only light to moderate loads (50-75% load) are used ` year based on the cooling load profile of the building per year. (Energy Design Resources, 2010)

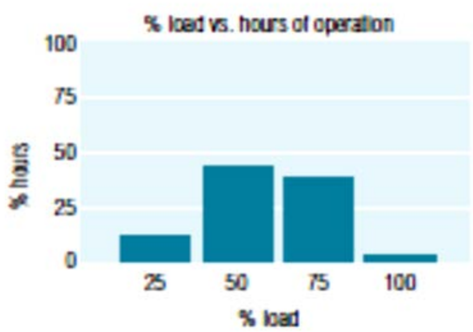

FIGURE 2.1: Typical office building cooling load profile (Energy Design Resource, 2010)

Since chillers operate most frequently at lower load conditions, the system running at full load will cause chiller efficiency to drop and more significantly when the outdoor temperature is low and the condenser temperature is set at a high level.

Therefore, to achieve building energy conservation, the key is to improve the part load efficiency of chillers. (Yu & Chan, 2009)

2.2 Optimization of Chiller System

Manufacturers of chiller have made great improvements to their system to improve efficiency. Performance of centrifugal chillers have improves from 0.75 kW/ton in the 1970s to a performance of less than 0.5 kW/ton today. (TRANE, 2000)

To capture every opportunity to save energy in the chiller system, variable speed control on the chiller system is slowly becoming an attractive alternative when conservation of electrical energy becomes a main concern. The benefit of variable speed control is it can match the required load requirement instantaneously by reducing the systems overall speed to save unnecessary energy consumption. (Saidur, 2008)

According to Qureshi and Tassoi (1996), chillers equipped with variable speed have better efficiency compared to conventional chillers with constant speed compressor. In addition, based on studies done by Bafnfleth and Peyer (2006), the total annual plant energy use is reduced by 2-5% and first cost by 4-8% by varying the primary flow for chillers.

Hartman (2001) proposed that variable speed chiller plants can improve plant energy performance significantly. In Hartman's study, he explained how condenser pump and tower tans should be operated during part load to achieve best performance. Yet he did not discuss on how to integrate properly the variable flow pumping system to reduce further pumping energy at part load operation.

Saidur et. al (2011) did a study on the chiller system in University of Malaya and compared the energy savings of the system when equipped with variable speed drive (VSD). It is found out that up to 2800 MWh can be saved after installing VSD in the chiller system and up to 11 million kg of carbon dioxide emission due to combustion of fossil fuels can be reduced when 60% of the motor speed is reduced using VSD.

2.3 Variable Flow Pumps to Optimize Chiller Performance

In a study by TRANE (2000), with reducing cooling loads, water pumps and cooling tower fans account for a larger percentage in the total energy consumption. Figure 2.2 shows the energy consumption of each chiller equipment with respect to the load capacity. At 25% load conditions, it is observed that the auxiliary equipment eg: cooling tower fan and pumps has a higher energy consumption compared to the chiller. (Figure 2.2)

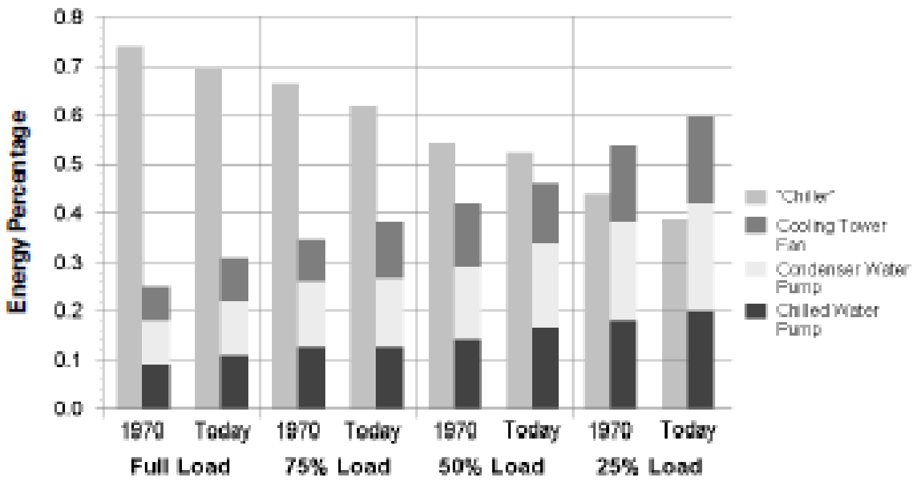

FIGURE 2.2: Energy percentage of different chiller plant equipment with different cooling load. (TRANE,2000)

A research study on variable primary flow pumping systems indicated that modern chillers are adaptable to flow rate modulation, provided that the flow rate is controlled between the minimum and maximum levels and does not exceed the tolerable range. (Bahnfleth & Peyer, 2004) Braun et al. (1989) also emphasized that using variable speed pump is a complement to optimize control of the chilled water systems.

Moses (2004) state that the most significant impact of variable primary flow is on chiller efficiency and it can prevent unnecessary chillers from being used prematurely. Gordon et al. (1999) also proved improvements in the energy performance of chiller system by varying chiller COP at different condenser flow rate. For true optimization, there has to be a balance dispersion of energy usage in the whole chiller system which accounts not only for the chiller itself but the auxiliary equipment as well.

2.4 Modelling of a Water-Cooled Chiller System

Chiller plant system has been developed in a lot of studies to evaluate the performance of the system. Most chiller systems were based on semi-empirical formulas where some of the information was taken from chiller manufacturer performance data to obtain results.

Gordan et.al (2000) formulated a semi-empirical model to investigate the variations of COP at different condenser flow rate. The model is based on derived mathematical equations together with actual performance data from chiller to obtain its result. The model is able to prove the benefits in varying condenser flow rate; however, no control regime was set on how water flow rate should be varied in response to various chiller loading to achieve optimized performance of the system.

Yu and Chan (2008) developed a thermodynamic model to optimize the water-cooled chiller system by using load-based speed control. The model uses common thermodynamic equations coupled with a few constant parameters from chiller performance data. The model clearly dissects the formulation of the thermodynamic system to give a clear view on how it is modelled. The heat exchangers in the models (evaporator and condenser) were modelled using the LMTD method to evaluate the accurate heat transfer coefficient from water to refrigerant and vice versa. For instance, the equations involved in the evaporator are:

$$Q = \dot{m}_w c_{pw}(T_{out} - T_{in})$$

$$Q = AU_{ev}LMTD_{ev}$$

where,

$$LMTD_{ev} = \frac{T_{chwr} - T_{chws}}{In(\frac{T_{chwr} - T_{ev}}{T_{chws} - T_{ev}})}$$

$$AU_{ev} = \frac{1}{c_1\dot{m}_{chw}^{-0.8} + c_2Q_L^{-0.745} + c_3}$$

As seen above, the thermodynamic equations were coupled with a few derived formulas (AUev) that required input from chiller performance data. The model is able to prove that optimal control on the flow rate and cooling tower can improve chiller performance and reduce operating cost of the chiller plant by 5.3%.

Equations used in this study will be explained in detailed in the sections below. Most of the equations will be based on thermodynamics equation with some reference from previous studies to obtain more accurate calculation that is able to represent the actual working condition of a chiller plant system.

CHAPTER 3: METHODOLOGY

3.1 Research Methodology

The first stage of the project will start off with the modelling of a conventional chiller plant system. The model is based on data taken from previous studies and assumptions are made during the process so that the model is feasible with the resources available.

Pumps will be the focus of this project to optimise the water-cooled chiller system, thus the next step will be determining the controlling factor in these equipment that can affect the power generated by the whole system. The power needed to power the equipment in different cooling loads or part loads is analysed.

A comparison is also done on the chiller performance between the optimised system and the existing model. Energy savings is done on both systems to see how much electricity is saved with the optimised solution. The results are analysed and discussed on how well the strategy proposed can save energy in the chiller plant system. Recommendations are proposed for future works and research.

Refer Summary Flow Chart (Figure 3.1) on the next page.

3.2 Project Activities

- Modelling of conventional chiller system based on previous studies
- Using excel spreadsheet to tabulate all the parameters for both conventional and optimised system
- Iterative procedure to find chiller performance at different part load conditions
- Plot graph of chiller performance against different part load ratios for both systems
- Calculate electricity savings based on obtained results and annual cooling load profile.

3.3 Work Process Flow for Methodology

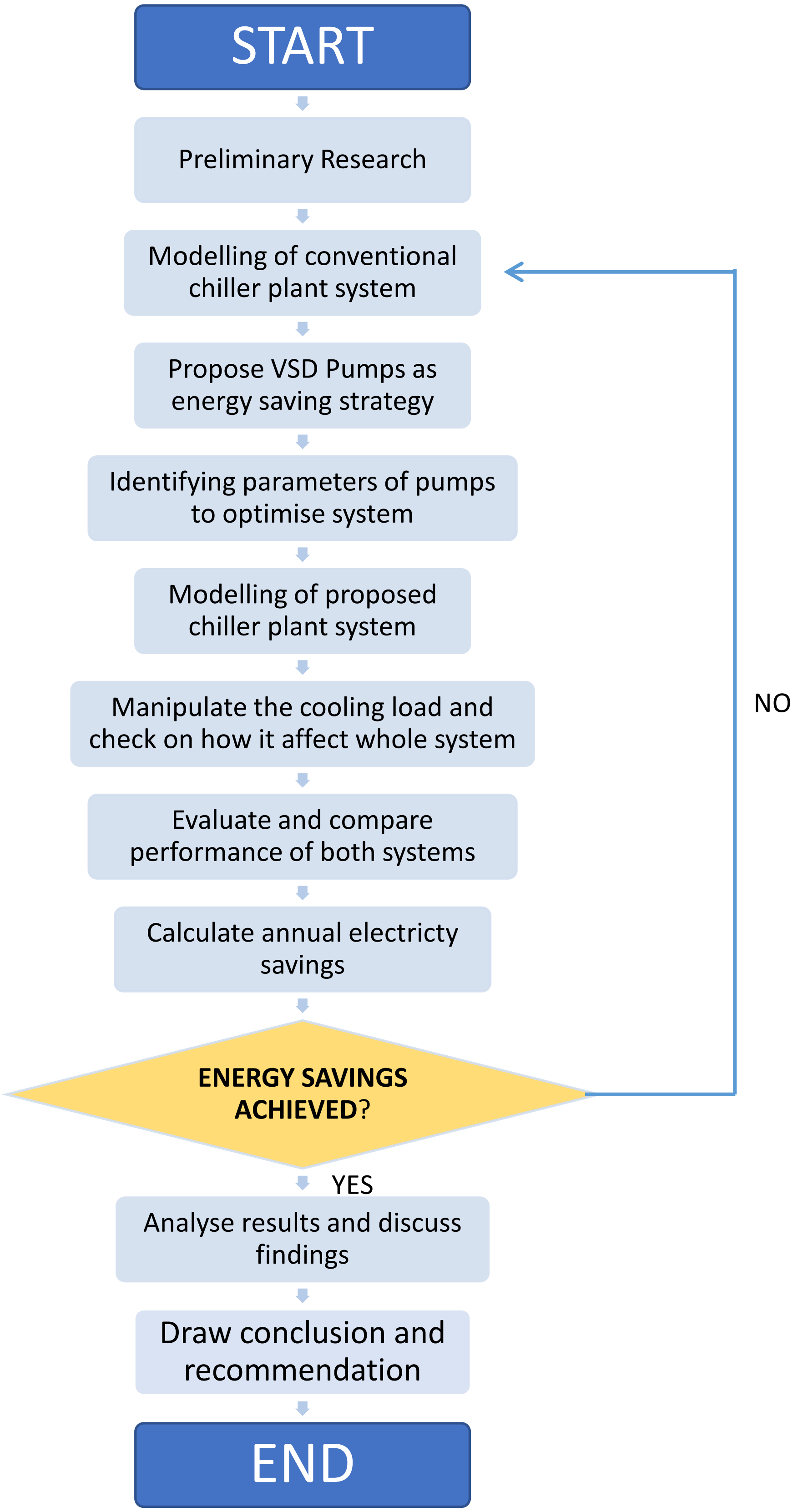

FIGURE 3.1: Work Process Flow for Methodology

3.4 Key Milestone

TABLE 3.1: Key Milestone of FYP1 and FYP2

FYP 1	
MILESTONE	**EVENT**
Week 7	Extended Proposal
Week 8	Model of Conventional System
Week 10	Confirm pumps as optimization strategy to save energy
Week 12	Preliminary Results
FYP 2	
MILESTONE	**EVENT**
Week 7	Confirmed model for conventional and proposed chiller system
Week 8	Progress Report
Week 9	Results on performance of VSD Pumps
Week 10	Annual Electricity Savings on Chiller System
Week 11-14	Poster Presentation, VIVA, Technical Report, Dissertation

3.5 Gantt Chart

TABLE 3.2: Gantt chart for FYP 1 and FYP 2

NO	TASK	FYP 1														FINAL EXAM & SEM BREAK	FYP 2													
		1	2	3	4	5	6	7	8	9	10	11	12	13	14		1	2	3	4	5	6	7	8	9	10	11	12	13	14
1.0	**CONFIRMATION OF FYP TITLE AND SUPERVISOR**																													
2.0	**PRELIMINARY RESEARCH**																													
	2.1 Literature Review																													
	2.2 Problem Analysis																													
3.0	**PROJECT INITIALIZATION**																													
	3.1 Forming correct methodology to kickstart project																													
	3.2 Setting assumptions and parameters																													
	3.3 Identifying potential energy saving equipment																													
4.0	**PROJECT EXECUTION**																													
	4.1 Modelling of Conventional Water-cooled Chiller System																													
	4.2 Investigate the effect of variable pump on chiller system																													
	4.3 Modelling of Proposed Water-cooled chiller system																													
5.0	**PROJECT ANALYSIS**																													
	5.1 Plotting of Chiller Performance with part load ratio																													
	5.2 Compare results between both system																													
	5.3 Calcualte annual electricty savings																													
6.0	**REPORT SUBMISSION**																													
	6.1 Extended Proposal																													
	6.2 Interim Report																													
	6.3 Progress Report																													
	6.4 Dissertation																													
7.0	**PRESENTATION**																													
	7.1 Proposal Defence																													
	7.2 VIVA																													

CHAPTER 4: RESULTS AND DISCUSSION

4.1 Modelling of Water-Cooled Chiller Plant System

The water-cooled chiller plant model is driven by a vapour compression refrigeration cycle with R-134a selected as the working fluid. Some of the design parameters of the chiller system are as follow:

TABLE 4.1: Design Operating Conditions of Water Chiller Plant

Design Specifications	Parameters	Unit
For Chiller		
Nominal Cooling Capacity	1600	kW
Nominal Compressor Power	280.7	kW
Refrigerant Type	R134a	
Design Chiller Water Supply Temperature	7	deg C
Design Chiller Water Return Temperature	12.5	deg C
Design Condenser Water Entering Temperature	33	deg C
Design Condenser Water Leaving Temperature	38	deg C
For Pumps		
Rated power for each chilled water pump	47	kW
Rated power for each cooling water pump	21.6	kW
Design Chilled Water Flow Rate	72	l/s
Design Condenser Water Flow Rate	87	l/s

All of the parameters are at design operating conditions and full load condition. The graphical modelling of the chiller system consist of 3 loops, the cooling water loop, the chilled water loop and the main vapour compression refrigeration cycle as shown in FIGURE 4.1:

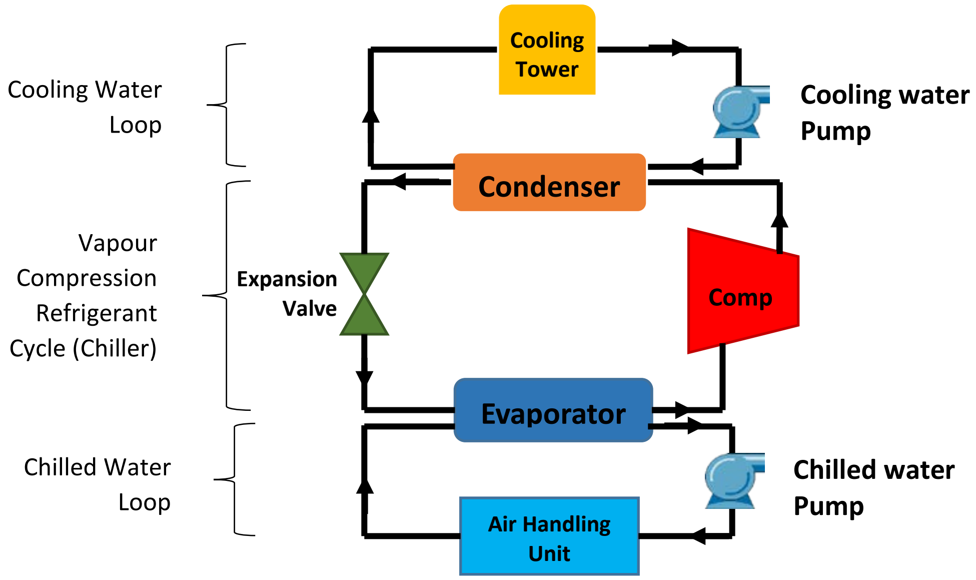

FIGURE 4.1: Graphical modelling of Water Chiller Plant System

By referring to the P-h diagram of the vapour compression refrigeration cycle (FIGURE 4.2), a few assumptions can be set to ease the hassle for calculations.

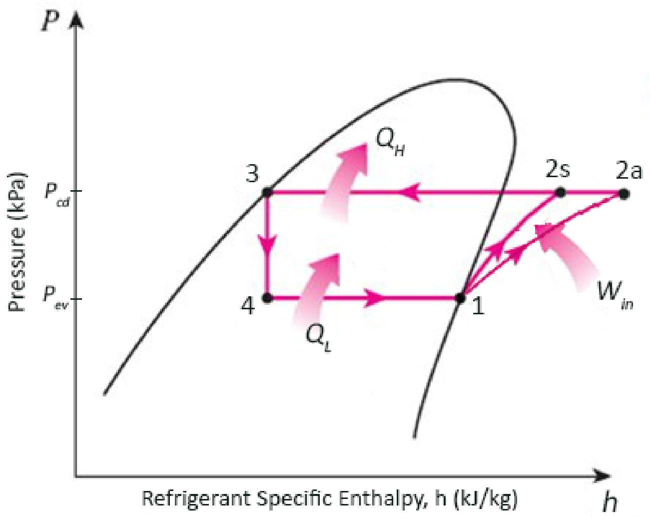

FIGURE 4.2: P-h diagram of vapour compression refrigeration cycle of the water-cooled chiller plant used in this study

At point 1, saturated vapour refrigerant leaves the evaporator to the compressor. Saturated vapour refrigerant is assumed to be compressed to point 2s through isentropic compression; a polytropic efficiency is factored in to mimic the actual compression of the compressor (point 2a). (will be elaborated further below). At the outlet of the compressor, the refrigerant is desuperheated in the condenser to its saturated gas state and continues on to condense to its saturated liquid state at point 3. From point 3, the refrigerant will be expanded in constant enthalpy before entering the evaporator back again.

At the heat exchangers (evaporator and condenser), no pressure drop is assumed in the refrigerant pipeline as changes in the pressure is considered negligible and does not caused any obvious variations in the performance of the chiller. Here are a few input parameters that need to be inserted for each operating conditions.

TABLE 4.2: Input parameters for chiller system model

No.	Parameters
1	Cooling Capacity, Q_{cl}
2	Chilled water flow rate, m_{chw}
3	Temperature of supply chilled water T_{chws}
4	Condenser water flow rate. m_{cdw}

Since pumps are the focal optimizing strategy of this study, the input of mass flow rate of chilled water pump and condenser water pump will be the controlling variable. The temperature of supply chilled water is set constant at 7 deg C in all operating conditions. Although increasing the supply temperature may be a way to increase efficiency, but we will not be looking into it in this study.

The model of the chiller system is governed by these thermodynamics and algebraic equations through an iterative procedure.

4.1.1 Evaporator Model

The cooling capacity, Q_L of the evaporator is related to the following equations:

$$Q_L = PLR * Q_{C.Rated}$$
$$Q_L = \dot{m}_{chw} c_{pw} (T_{chwr} - T_{chws})$$
$$Q_L = \dot{m}_{ref} (h_1 - h_4)$$
$$Q_L = AU_{ev} LMTD_{ev}$$

where,

$$LMTD_{ev} = \frac{T_{chwr} - T_{chws}}{In(\frac{T_{chwr} - T_{ev}}{T_{chws} - T_{ev}})}$$

$$AU_{ev} = \frac{1}{c_1 \dot{m}_{chw}^{-0.8} + c_2 Q_L^{-0.745} + c_3}$$ …..(Yu & Chan, 2008)

The evaporator heat transfer from the chilled water loop to the vapour compression loop is model using the log mean temperature difference method (LMTD). The process of calculating the overall heat transfer coefficient (AU_{ev}) is time consuming with limited known values, therefore, a relation done by Yu & Chan (2008) based on the performance data of the chiller at full load and part load conditions are used to save calculation time. AU_{ev} is a function $\dot{m}_{chw}$ and Q_L, where $c_1 = 0.1172, c_2 = 0.0593, c_3 = 0.0001$ of are constants used in the relation.

4.1.2 Compressor Model

The chiller is modelled to be a hermetic centrifugal compressor running at constant speed. However, the compressor is equipped with inlet guide vanes to regulate the cooling output during part load and full load capacity at the same time control the supply chilled water temperature at its constant point. Therefore, the refrigerant flow rate will vary with different input of cooling load.

To start off, the compressor is first assumed to undergo isentropic compression ($s_1 = s_2$). The mass flow rate of the refrigerant is calculated from Q_L, where

$$\dot{m}_{ref} = \frac{Q_L}{(h_1 - h_4)}$$

The isentropic compression yields the enthalpy at point 2 (h_{2s}). However, isentropic compression does not happen in actual compression. In order to get a closer value to the actual compression, a polytropic compression efficiency is factored in to find the actual enthalpy at point 2 (h_2). The polytropic compression expression, a function of PLR, is obtained from chiller performance data and mechanical work input of the compressor under part load conditions, (Yu & Chan, 2008)

$$\eta_{pol} = a_1 PLR^2 + a_2 PLR + a_3 \text{ …… (Yu \& Chan, 2008)}$$

$$\text{where, } a_1 = -0.8131, a_2 = 1.537, a_3 = 0.0057$$

The actual enthalpy at point 2, h_2 is calculated with different compressor efficiency at different part load ratio. Actual compressor work input can be obtained to calculate the performance of the chiller.

$$\eta_{pol} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$\dot{W}_{input} = \dot{m}_{ref}(h_2 - h_1)$$

4.1.3 Condenser Model

The condenser model works the same as the evaporator. The cooling capacity, Q_L is now replaced with the heat rejected to the cooling tower, Q_H.The equations for energy and mass balance of the condenser are as follow:

$$Q_H = \dot{m}_{ref}\,(h_2 - h_3)$$

$$Q_H = \dot{m}_{cdw} c_{pw}(T_{cdwl} - T_{cdwe})$$

$$Q_H = AU_{cd}\, LMTD_{cd}$$

where,

$$LMTD_{cd} = \frac{T_{cdwl} - T_{cdwe}}{In(\frac{T_{cd} - T_{cdwe}}{T_{cd} - T_{cdwl}})}$$

$$AU_{cd} = \frac{1}{c_4 \dot{m}_{cdw}^{-0.8} + c_5 Q_H^{-1/3} + c_6} \text{ .....(Yu \& Chan, 2008)}$$

The overall heat coefficient AU_{cd} is derived from chiller performance data by Yu & Chan (2008). The constant parameters are $c_4 = -0.0005$, $c_5 = 0.3443$, $c_6 = 0.0002$ respectively.

4.1.4 Cooling Tower Model

The cooling tower is not modelled in the study. The cooling fan is assumed to run in constant speed to provide a constant temperature difference in the condenser. The heat rejection may be affected by the air dry bulb and wet bulb temperature however it is not taken into consideration in this study. The cooling tower will be modelled if there is need to evaluate how weather affects the performance of the water-cooled chiller plant system.

4.2 VSD Pumps in Water Chiller System

Pumps control the flow rate of chilled water or cooling water in the heat rejection process. Conventional pumping system in water-cooled chillers run constantly at all conditions. VSD Pumps reduces flow rate of the pumps according to the instantaneous load conditions. This means that, the flow rate of water in the system reduced according to the required load at that moment. From the general equation below,

$$\dot{m}_{chilled\ water} = \frac{Q_L}{c_p \Delta T} \qquad \dot{m}_{cooling\ water} = \frac{Q_H}{c_p \Delta T}$$

Assuming $c_p \Delta T$ is the same, from the equation, $Q_L\ and\ Q_H$ is directly proportional to the flow rate of water. During lesser load conditions, lesser flow rate will be required to pump the water. Furthermore, a lesser flow rate required also will reduce the pump power required. According to the Pump Affinity Law, the relationship of flow rate and pump power can be described as follow,

$$\frac{\dot{W}_2}{\dot{W}_1} = \left(\frac{\dot{m}_2}{\dot{m}_1}\right)^3$$

From the equation it is observed that a reduced flow ratio has a reduced cubic effect on the power required thus the lower the flow rate, more pump power can be saved.

TABLE 4.3: The power savings of a pump with lesser flow ratio

Flow Ratio, $\frac{\dot{m}_2}{\dot{m}_1}$	1.0	0.9	0.8	0.7	0.6	0.5
Power Ratio, $\frac{\dot{W}_2}{\dot{W}_1}$	1.000	0.729	0.512	0.343	0.216	0.125
Power Saving (%)	0	27.1	48.8	65.7	78.4	87.5

Based on previous studies on the reduction in flow rate, the maximum allowable reduction in mass flow rate of water is between 30 to 60% at part load conditions. (Brasz & Tetu, 2008) This range is based on the industry recommendation of the minimum water velocity in the tubes which is at least 3 ft/s (approx.1 m/s). In another study, Yu & Chan (2008) also suggested that the minimum chilled water flow rate is to be set at 50% of the design flow rate for both condenser and chilled water pump. This is because below the limit, it might cause problems such as fouling and scaling at the cooling coils which reduced the performance of the system.

Therefore, it is important not to reduce the flow rate of the pumps under the limit set by manufacturers so that the performance of the system is not affected. Despite the attractive prospect to save pump power, reduction of flow rate has to be done carefully at suitable part load conditions without jeopardizing the performance of the system to achieve true optimization.

4.3 Procedure to Determine Variables in Chiller Model

The flow chart in Figure 4.3 shows the procedure and steps to on how all the operating variables are evaluated. We can see the relationship of each model and how they are linked together to find the performance of the chiller system.

The model starts off with 4 input variables: cooling capacity (Q_L), chilled water supply temperature (T_{chws}), mass flow rate of chilled water (m_{chw}) and cooling water (mcdw) respectively. The evaporator model calculates the PLR, T_{chwr}, AU_{ev} and $LMTD_{ev}$ using the input of the cooling capacity, chilled water flow rate and the temperature of supply chilled water. From there onwards, the evaporating temperature and pressure can be obtained together with the enthalpy and entropy at point 1.

The other part of the model involves the compressor and condenser model, combined in an iterative procedure. The condensing temperature is first assumed to be at 45 deg C. Using this temperature, the condensing pressure, P_{cd} and the refrigerant properties such as the enthalpy of point 3 and 4, mass flow rate of refrigerant and following by that the actual enthalpy at point 2 with addition of a polytropic efficiency factor. The work required to power the compressor will be obtained at the end of the compressor model.

Having the properties of the refrigerant at every point, the heat rejected out from the condenser, Qh can be calculated. Calculation of AU_{cd}, $LMTD_{cd}$ will lead to the calculated value of the condenser temperature at the end. This value will be compared with the initial iteration value of 45 deg C. If the error is more than 0.001, The calculated condenser temperature will replace the initial value and the iteration will continue on until it converges within 0.001.

The chiller performance in kW/RT can be determined next by dividing the work done of the system with respective part load conditions in RT. The based condition is that the pumps run at a constant speed where the pumps will be running at its rated power. When VSD is applied into the system, the work input of pumps will be reduced based on the cubic relationship between power and flow rate as stated in Affinity Law. The mass flow rate varies proportionally with the part load ratio (PLR) starting from 0.5 mass flow rate at PLR of 0.5 to the rated mass flow rate when PLR is 1.0.

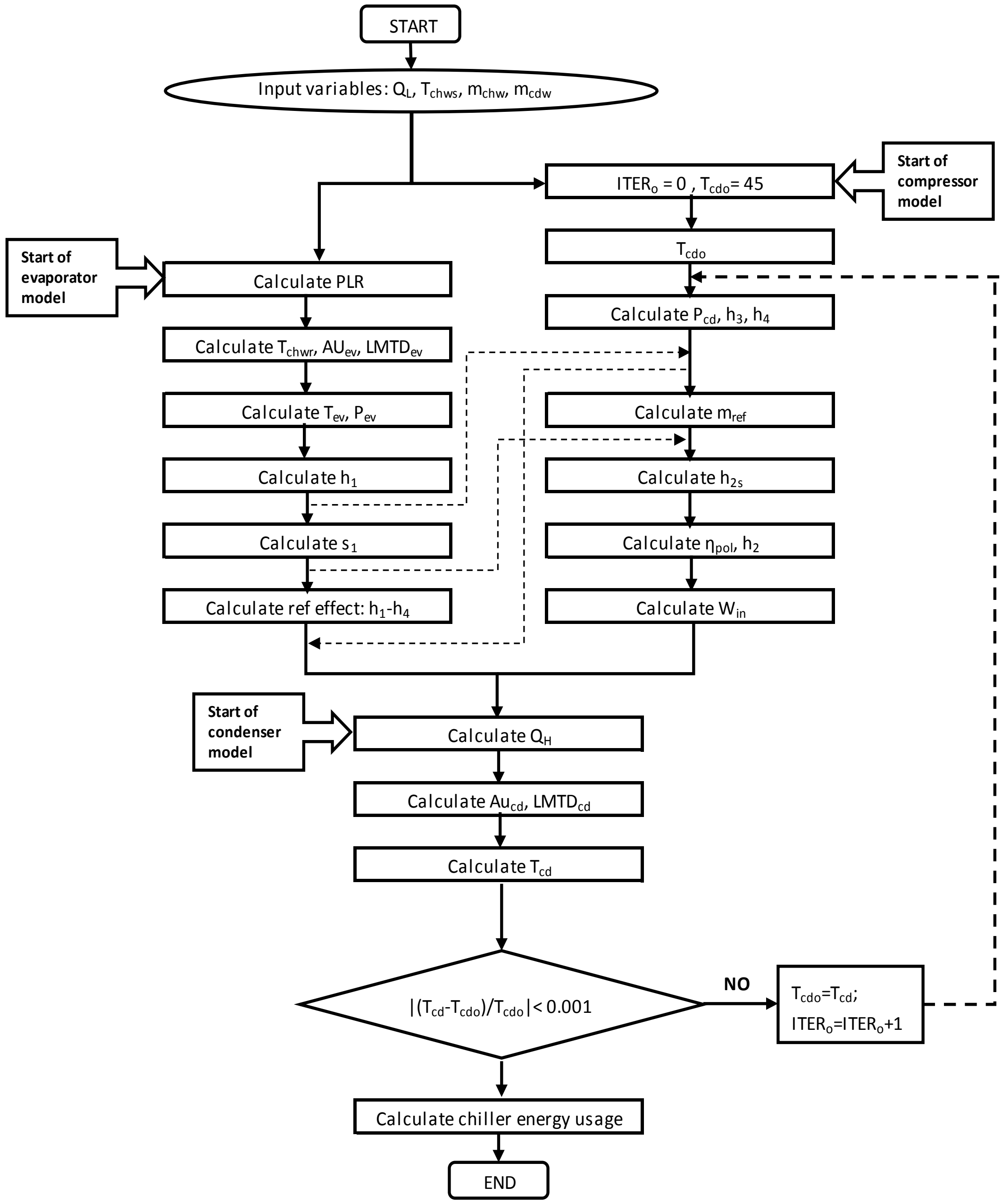

FIGURE 4.3: Flow chart to determine the operating variables of the chiller model through iterative procedure

4.4 Results and Discussion

A chiller running in design operating conditions is not the most energy efficient way as most of the time; chillers operate at lower cooling loads. Therefore, we have to look into the system in off-design operating conditions, or can also be known as part load conditions.

The temperature distribution changes throughout the whole day which directly affects the cooling load in a day. A change in the cooling capacity, Q_L can affect the refrigeration cycle and causes change to the pump power required. To replicate the varying cooling load throughout a typical day, Q_L will be varied accordingly from full load to part load conditions.

The calculation is done in Microsoft Excel to plot the chiller efficiency at different PLR. Table 4.4 shows the base case where the flow rate of chilled water and condenser water remain the same for every operating load conditions. It is observed that the when the mass flow rate remains constant, there is changes to the temperature difference of both heat exchangers. Looking at the relationship of flow rate and cooling load,

$$\Delta T_{ev} = \frac{Q_L}{c_p \dot{m}_{chw}} \qquad \Delta T_{cd} = \frac{Q_H}{c_p \dot{m}_{cdw}}$$

With reduced cooling load, the temperature difference in the heat exchangers will reduced with constant flow rate. From Table 4.4, it is observed that at the evaporator with the chilled water flow rate (m(dot)chw) is kept constant, and with chilled water supply temperature set at design temperature of 7 deg C, the chilled water return temperature (Tchwr) reduces at part load conditions, causing a drop in temperature difference at lower load The same drop in temperature is observed at the condenser with constant condenser flow rate of 87 kg/s.

TABLE 4.4: Calculation of chiller model at constant pump flow rate

PLR	Qcl	m(dot) chw	Tchwr	ΔTev	Qcd	m(dot) cdw	Tcdwe	ΔTcd
1.00	1600	72.0	12.309	5.309	2018.473	87.0	32.457	5.543
0.90	1440	72.0	11.778	4.778	1799.971	87.0	33.057	4.943
0.80	1280	72.0	11.247	4.247	1594.101	87.0	33.622	4.378
0.70	1120	72.0	10.717	3.717	1393.125	87.0	34.174	3.826
0.60	960	72.0	10.186	3.186	1201.698	87.0	34.700	3.300
0.50	800	72.0	9.655	2.655	1015.774	87.0	35.210	2.790
0.40	640	72.0	9.124	2.124	829.571	87.0	35.722	2.278
0.30	480	72.0	8.593	1.593	646.882	87.0	36.224	1.776
0.25	400	72.0	8.327	1.327	557.600	87.0	36.469	1.531
0.20	320	72.0	8.062	1.062	469.531	87.0	36.711	1.289
0.10	160	72.0	7.531	0.531	295.460	87.0	37.189	0.811

Moving on to the proposed chiller system to use variable flow pumps, when the flow rate of pump varies proportionally with reduced load, the temperature difference of the heat exchangers can be made constant as seen in the relationship between mass flow rate and operating load condition. However, as stated in previous study, the flow rate of water can only be reduced to a maximum of 0.5 PLR, therefore, for loadings lower than 0.5, the mass flow rate of water will maintain at 0.5 PLR which will cause a drop in temperature difference starting from 0.5 PLR onwards. From Table 4.5, it is seen that the temperature difference can be set constant at 5.5 deg C (Evaporator) and 5 deg C (Condenser) from 1.0 to 0.5 PLR. The temperature difference exhibit the same pattern of the base case on constant flow rate pumps previously after 0.5 PLR where temperature difference continues to drop thereafter.

$$\dot{m}_{chw} = \frac{Q_L}{c_p \Delta T} \qquad \dot{m}_{cdw} = \frac{Q_H}{c_p \Delta T}$$

TABLE 4.5: Calculation of chiller model at VSD pump flow rate

PLR	Qcl	m(dot) chw	Tchwr	ΔTev	Qcd	m dot cdw	Tcdwe	ΔTcd
1.00	1600	69.50	12.5	5.5	2022.35	96.64	33.00	5.00
0.90	1440	62.55	12.5	5.5	1803.48	86.18	33.00	5.00
0.80	1280	55.60	12.5	5.5	1597.19	76.32	33.00	5.00
0.70	1120	48.65	12.5	5.5	1399.99	66.90	33.00	5.00
0.60	960	41.70	12.5	5.5	1209.45	57.79	33.00	5.00
0.50	800	34.75	12.5	5.5	1024.06	48.93	33.00	5.00
0.40	640	34.75	11.4	4.4	835.75	48.93	33.92	4.08
0.30	480	34.75	10.3	3.3	651.06	48.93	34.82	3.18
0.25	400	34.75	9.75	2.75	560.91	48.93	35.26	2.74
0.20	320	34.75	9.2	2.2	472.05	48.93	35.70	2.30
0.10	160	34.75	8.1	1.1	296.58	48.93	36.55	1.45

With all the data obtained, the performance of the chiller system at difference part load can be plotted. Performance of the chiller system is in the unit of kW/RT which is the standard commonly used in the HVAC industry. It represents the power used per refrigeration ton (RT) of cooling load. The lower the value the better the energy saving capability at any particular part load conditions. Figure 4.4 shows the plot of both conventional and proposed optimised system. The performance started off the same at full load (1 PLR) approximately 1.1 kW/RT. Moving to lower part load ratios, it can be seen that the variable flow system performs better as compared to the conventional constant flow system. This means that the performance of variable flow pumps improved in part load conditions.

It is observed that both plots show a decrease in performance (higher kW/RT) with lower part load ratios. This is due to the same constant speed compressor used for both systems. The efficiency of the compressor reduces at part load conditions causing a higher kW/RT. The highest kW/RT for constant flow system is up to 4.5 kW/RT whereas variable flow system is able to conserve energy usage to a lower value of 3.2 kW/RT. Meaning a lower energy consumption and higher performance at part load conditions.

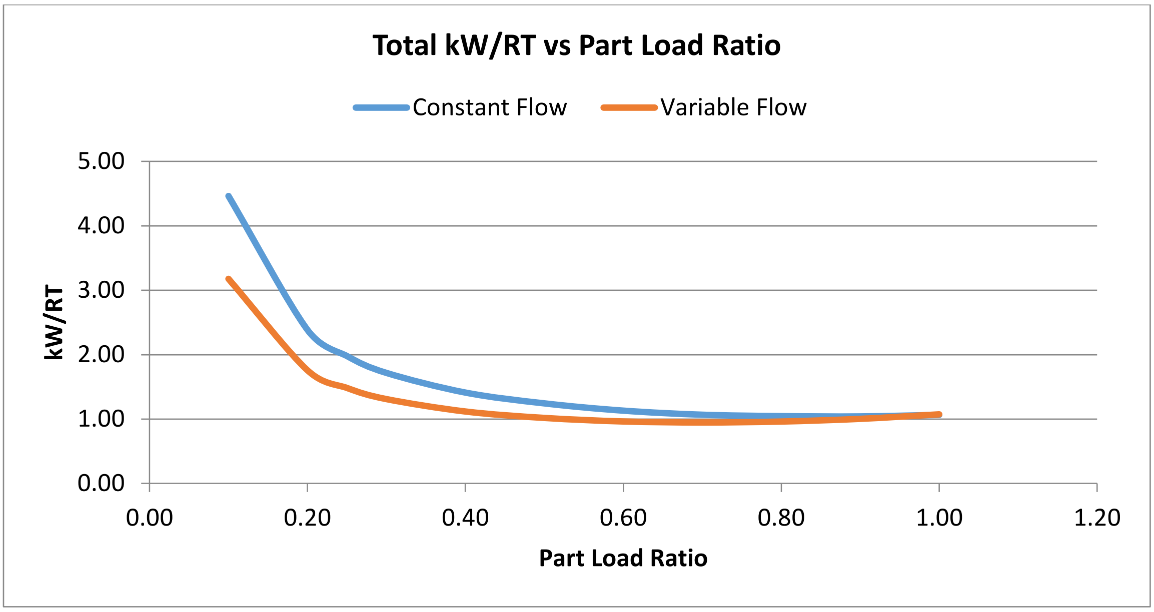

FIGURE 4.4: Performance of Chiller Plant System vs Part Load Ratio

4.5 Annual Electricity Savings Analysis

To calculate the electricity savings in a typical commercial building, the annual cooling load profile is used to evaluate the savings. Figure 4.5 shows the cooling load profile in a year of a typical building (Yu & Chan, 2009). The operating hours at full load is the lowest in the year (20 hours), whereas at most hours the chiller runs at part load conditions. From this information, the kWh of every cooling load conditions can be plotted to check the annual energy usage of the chiller plant.

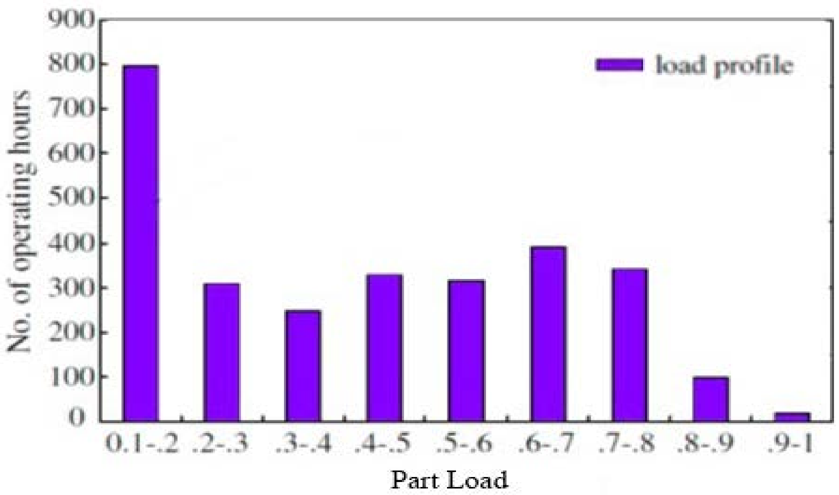

FIGURE 4.5: Cooling load profile for a typical building in a year

Figure 4.6 shows the results of the pump energy usage at different part load ratio. It can be seen that the constant flow chilled water pumps (CHW Pump) and cooling water pumps (CW Pump) has the overall higher energy usage. A significant drop can be seen with the usage of variable speed pumps (VSD). It can be observed that the energy used in full load condition for both constant and VSD pumps are relatively the same because both pumps are required to run at full load. However, energy reduction of VSD pumps reduces significantly at lower load conditions saving much pump power. For instance, at 0.15 PLR, chilled water pump energy is reduced from 35500 kWh to less than 5000 kWh. The savings are contributed from the cubic relationship between power and flow rate which reduces the power usage significantly at lower load conditions. The improvement in performance of the chiller system pairing up with the energy savings of pumps brings upon optimization to the system.

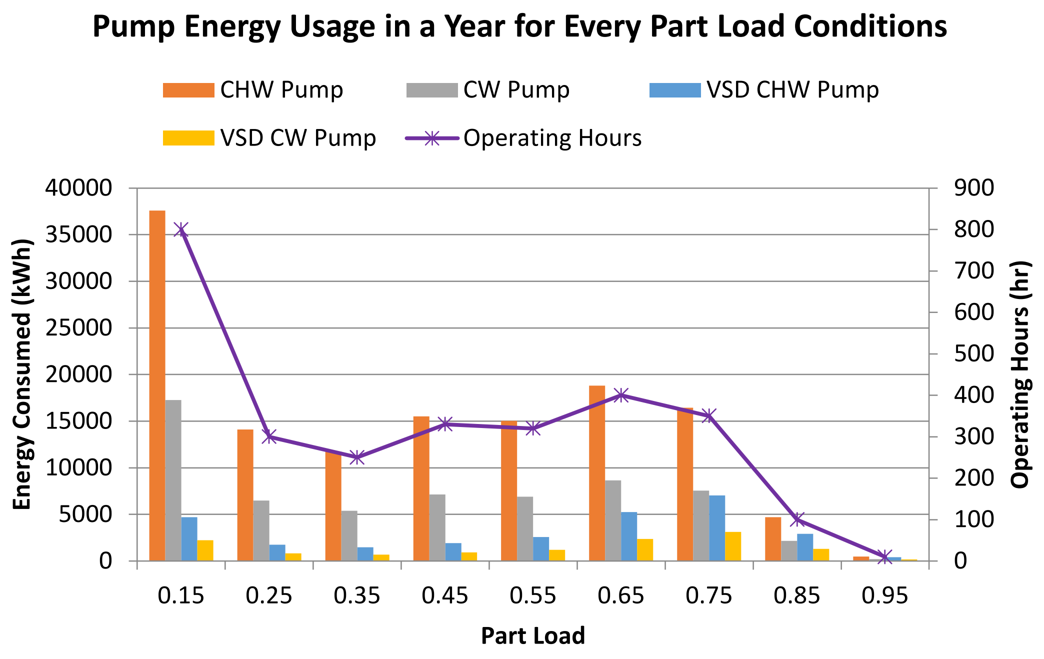

FIGURE 4.6: Pump energy consumed based on annual cooling load profile for every part load condition

Looking at the energy savings of the whole chiller plant system, Figure 4.7 shows us the comparison of both plants energy usage. Savings of plant energy usage is also consistent for the variable flow pump used in the optimised chiller plant. Reduction of energy consumption can be seen clearly especially at part load conditions. However, the savings are not as significant as the pump energy usage. This is due to the same compressor used in both cases. The work done of the compressor is relatively same for both cases (constant flow and variable flow) causing a reduced in savings when calculating the total plant energy usage. The benefit of variable flow in chiller plant system still stands and is a strong candidate for energy saving opportunities in a chiller plant system.

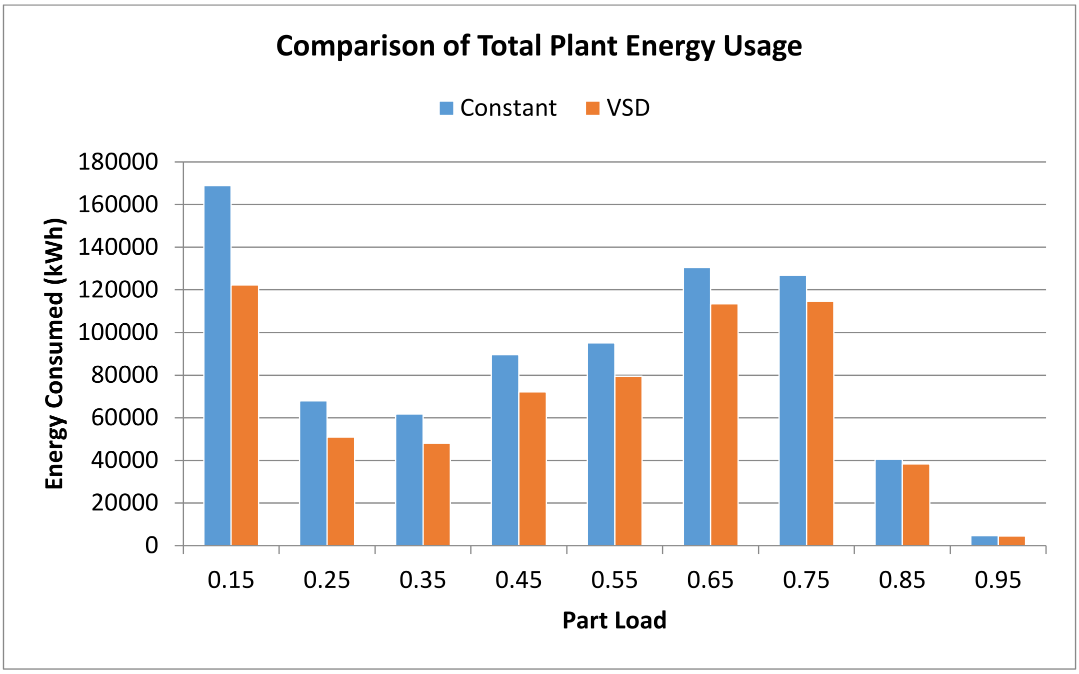

FIGURE 4.7: Total Plant Energy Saving Comparison

TABLE 4.6: Energy Saving Percentage and Electricity Savings of Chiller Plant

Description	Base Case (kWh)	Variable Flow (kWh)	Energy Saved (kWh)	Perc. (%)
CHW Pump Energy	134420	28044	106376	79.14
CW Pump Energy	61776	12855	48921	79.19
Total Pump Energy	196196	40899	155297	79.15
Total Plant Energy	785435	643873	141562	18.02
Electricity Saving (RM)	RM 377009	RM 309059	RM 67950	-

Table 4.6 shows the saving percentage of the water-cooled chiller system. It is observed that variable speed pumps can save up to 80% of pump power when compared to the conventional chiller system. Calculating the total chiller plant

energy saved, 18% of plant energy can be saved. Using the standard electricity tariff rate in Malaysia, the electrical savings is up to RM68k per annum.

Although only one cooling load profile was used to determine the savings of the chiller plant system, it is safe to assume that the nominal cooling load of most buildings falls more to part load conditions rather than full load. Thus, energy savings through variable speed pumps can be implemented in most of the chiller plant system unless the cooling load of the building needs to be at full load most of the time throughout the year.

CHAPTER 5: CONCLUSIONS AND RECOMMENDATION

Water-cooled chiller plant system is one of the major consumers of energy in most of the buildings that has the plant installed. The high energy consumption of the chiller plant raised concerns on how to effectively reduce the energy usage and at the same time improve the performance of the chiller plant.

This paper presents a thermodynamic chiller model to investigate the energy saving opportunities in a water-cooled chiller plant. The optimization strategy of using variable speed pumps for both chilled water and cooling water pumps is proven to be able to increase chiller plant performance and also save significant amount of energy as compared to conventional chiller plant systems. At part load conditions, performance of the chiller system with variable flow is superior over the conventional chiller plant system that uses pumps with constant flow rate. Using a cooling load profile of a typical building in a year, the proposed chiller system is able to save up to 80% of pump power and 18% of total plant energy usage. The objective of this paper is achieved. VSD pumps can consider as an optimization strategy for chiller plant system to save great amount of energy.

Further studies should be done on applying variable speed drive on other equipment in the chiller system such as the compressor and cooling tower to evaluate further possibilities of energy saving opportunities of water-cooled chiller system. The findings of this paper highlights the importance of the application of variable speed drive on water-cooled chiller systems serving air-conditioned building in order to enhance sustainability and promote energy conservation.

REFERENCES

ASHRAE. (2010). *ASHRAE Standard 90.1-2010.* Trane.

Brasz, J., & Tetu, L. (2008). Variable -Speed Centrifugal Chiller Control for Variable Primary Flow (VRP) Applications. *International Compressor Engineering Conference* (p. 1922). Purdue University.

Cengel, Y. A., & Boles, M. A. (2008). *Thermodynamics: An Engineering Approach (6th Edition).* McGraw-Hill.

Chang, Y.-C. (2007). Optimal Chiller Loading by Evolution Strategy for Saving Energy. *Energy and Buildings*, 437-444.

Crowther, H., & J. Furlong. (2004). Optimizing chillers and towers. *ASHRAE J 46 (7)*, 34-40.

EC. (2007). *Statistics of Electrical Supply in Malaysia.* Malaysia: Energy Commission.

Energy Design Resources. (2010, June 13). *Design brief: Chiller Plant Efficiency.* Retrieved from Energy Design Resources: http://energydesignresources.com/resources/publications/design-briefs/design-brief-chiller-plant-efficiency.aspx

F.W., Y., & Chan, K. (2008). Optimization of water-cooled chiller system with load-based speed control. *Applied Energy 85*, 931.

FPL. (2013). *Florida Power and Light Company.* Retrieved from Water-Cooled Chillers: http://www.fpl.com/business/energy_saving/pdf/br_water_cooled_primer.pdf

Furlong, J. W., & Morrison, F. T. (2006). Optimization of Water-Cooled Chiller - Cooling Tower Combinations. *CTI Journal*, 1-8.

HESS, H. (2013, December 6). *FactSheet Chiller Efficiency*. Retrieved from Australian Government, Department of Industry: http://www.industry.gov.au/Energy/EnergyEfficiency/Non-residentialBuildings/HVAC/FactSheets/Documents/HVACFSChillerEfficiency.pdf

Hydeman, M., & Zhou, G. (2007). Optimizing Chilled Water Plant Control. *ASHRAE Journal*.

J.M., G., Ng, K., Chua, H., & Lim, C. (2000). How varying condenser coolant flow rate affects chiller performance: Thermodynamic modelling and experimental confirmation. *Applied Thermo Eng. 20*, 1149-1159.

Jayamaha, L. (2007). *Energy efficiency building systems: green strategies for operation.* Mc Graw-Hill.

MNSU. (n.d.). *Chiller*. Retrieved from Engaged in Thermodynamics: http://cset.mnsu.edu/engagethermo/components_main.html

Progress Energy. (2012). *Energy Savers.* Retrieved from Chiller Optimization and Energy Efficient Chillers: www.progress-energy.com/assets/www/docs/business/chiller-fact-sheet-052005.pdf

Qureshi, T., & S.A.Tassou. (1996). Variable speed control in refrigeration system. *Applied Thermodynamics Eng. 16 (2)*, 103-113.

Reindl, D. T. (2009, January). The Vapor Compression Cycle.

Saidur, R. (2009). Energy Consumption, energy savings, and emission analysis in Malaysian office buildings. *Energy Policy 37*, 4104-4113.

Saidur, R., Hasanuzzaman, M., Mahlia, T., Rahim, N., & Mohammed, H. (2011). Chillers energy consumption, energy savings and emission analysis in an institutional building. *Energy 36*, 5233.

T.Hartman. (2001). All variable speed centrifugal chiller plants. *ASHRAE Journal 43 (9)*, 43-52.

Taylor, S. T. (2012). Optimizing Design & Control of Chilled Water Plants. *ASHRAE Journal*, 56-75.

Tracey. (2010, November 29). *How Do Water-Cooled Chillers Work, And When Can I Use Them*. Retrieved from Global Test Supply : http://blog.globaltestsupply.com/technical-articles/how-do-water-cooled-chillers-work-and-when-can-i-use-them/

TRANE. (2000, June). *Chilled Water Deisgn and Operation.* Retrieved from Cost Reducing Optimization Strategies: http://www.dalkia.com.au/literature/System%20Optimization/ctvslb005en.pdf

W.P.Bahnfleth, & E.B.Peyer. (2006). Energy use and comparison of chilled water pmping system alternatives. *ASHRAE Trans. 112 (2)*, 198-208.

Waste Reduction Partners Org. (2010, May). Retrieved from Chinners Energy Saving Fact Sheet: http://wastereductionpartners.org/phocadownload/userupload/Resources/Energy_Saving_Fact_Sheet_Chillers.pdf

Wei, X., Xu, G., & Kusiak, A. (2014). Modelling and optimization of a chiller plant. *Energy Volume 73*, 898-907.

Yu, F. W., & Chan, K. T. (2007). Part load performance of air-cooled centrifugal chillers with variable speed condenser fan control. *Building and Environment*, 3816.

Yu, F. W., & Chan, K. T. (2009). Environmental performance and economic analysis of all varaible speed chillers systems with load based speed control. *Applied Thermal Engineering* , 1721-1729.

APPENDICES

Excel screenshot on calculation of the system.

CONSTANT FLOW PUMPS										
PLR	Qcl	m(dot) chw	Tchwr	ΔTev	Win	Qcd	m(dot) cdw	Tcdwe	ΔTcd	kW/RT
1.00	1600	72.0	12.309	5.309	418.47	2018.473	87.0	32.457	5.543	1.07
0.90	1440	72.0	11.778	4.778	359.97	1799.971	87.0	33.057	4.943	1.04
0.80	1280	72.0	11.247	4.247	314.10	1594.101	87.0	33.622	4.378	1.05
0.70	1120	72.0	10.717	3.717	273.13	1393.125	87.0	34.174	3.826	1.07
0.60	960	72.0	10.186	3.186	241.70	1201.698	87.0	34.700	3.300	1.13
0.50	800	72.0	9.655	2.655	215.77	1015.774	87.0	35.210	2.790	1.24
0.40	640	72.0	9.124	2.124	189.57	829.571	87.0	35.722	2.278	1.41
0.30	480	72.0	8.593	1.593	166.88	646.882	87.0	36.224	1.776	1.72
0.25	400	72.0	8.327	1.327	157.60	557.600	87.0	36.469	1.531	1.98
0.20	320	72.0	8.062	1.062	149.53	469.531	87.0	36.711	1.289	2.39
0.10	160	72.0	7.531	0.531	135.46	295.460	87.0	37.189	0.811	4.46

VARIABLE FLOW PUMPS										
PLR	Qcl	m(dot) chw	Tchwr	ΔTev	Win	Qcd	m dot cdw	Tcdwe	ΔTcd	kW/RT
1.00	1600	69.50	12.5	5.5	422.35	2022.35	96.64	33.00	5.00	1.07
0.90	1440	62.55	12.5	5.5	363.48	1803.48	86.18	33.00	5.00	1.00
0.80	1280	55.60	12.5	5.5	317.19	1597.19	76.32	33.00	5.00	0.96
0.70	1120	48.65	12.5	5.5	279.99	1399.99	66.90	33.00	5.00	0.95
0.60	960	41.70	12.5	5.5	249.45	1209.45	57.79	33.00	5.00	0.96
0.50	800	34.75	12.5	5.5	224.06	1024.06	48.93	33.00	5.00	1.02
0.40	640	34.75	11.4	4.4	195.75	835.75	48.93	33.92	4.08	1.12
0.30	480	34.75	10.3	3.3	171.06	651.06	48.93	34.82	3.18	1.31
0.25	400	34.75	9.75	2.75	160.91	560.91	48.93	35.26	2.74	1.48
0.20	320	34.75	9.2	2.2	152.05	472.05	48.93	35.70	2.30	1.76
0.10	160	34.75	8.1	1.1	136.58	296.58	48.93	36.55	1.45	3.18

Calculate Energy Savings in Pumps and Total Plant System

			Chiller	Comp	CHW Pump	CW Pump		
PLR	**Cooling Load**	**Op. Hour**	**kWh**	**kWh**	**kWh**	**kWh**	**Total kWh**	**RM**
0.15	240	800	192000	114000	37600	17280	168880	81062.40
0.25	400	300	120000	47280	14100	6480	67860	32572.80
0.35	560	250	140000	44557.5	11750	5400	61707.5	29619.60
0.45	720	330	237600	66881.1	15510	7128	89519.1	42969.17
0.55	880	320	281600	73196.8	15040	6912	95148.8	45671.42
0.65	1040	400	416000	102964	18800	8640	130404	62593.92
0.75	1200	350	420000	102763.5	16450	7560	126773.5	60851.28
0.85	1360	100	136000	33704	4700	2160	40564	19470.72
0.95	1520	10	15200	3892.2	470	216	4578.2	2197.54
			1958400	**589239.1**	**134420**	**61776**	**785435.1**	**377008.85**

			Chiller	Comp	CHW Pumps	CW Pumps		
PLR	**Cooling Load**	**Op. Hour**	**kWh**	**kWh**	**kWh**	**kWh**	**Total kWh**	**RM**
0.15	240	800	192000	115448	4700	2243.6006	122391.6	58747.968
0.25	400	300	120000	48273	1762.5	841.35024	50876.85	24420.888
0.35	560	250	140000	45852.5	1468.75	701.1252	48022.375	23050.74
0.45	720	330	237600	69270.3	1938.75	925.48526	72134.535	34624.577
0.55	880	320	281600	75760	2564.32	1187.926	79512.246	38165.878
0.65	1040	400	416000	105900	5254.6	2357.1451	113511.75	54485.638
0.75	1200	350	420000	104506.5	7032.375	3116.0496	114654.92	55034.364
0.85	1360	100	136000	34034	2916.35	1297.944	38248.294	18359.181
0.95	1520	10	15200	3929.2	406.315	184.59286	4520.1079	2169.6518
			1958400	**602973.5**	**28043.96**	**12855.219**	**643872.68**	**309058.89**