Experimental and numerical optimization on chilled water configuration for improving temperature performance and economic viability of a centralized chiller plant

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Abstract. This research aimed to simulate and optimize chilled water circuit system of a central air conditioning plant applied for a five-star hotel in Bali Island, Indonesia. This optimization was performed because temperature of chilled water distributed to the loading system could not satisfy room temperature requirement. Detailed measurements have been taken on the AC system included refrigeration system of the chillers, chilled water and cooling water distribution, cooling load, heat rejection and pumping systems. Thermodynamic analyses on the system energy performance were carried out. Numerical simulations were also established to evaluate chilled water flow rate in the primary and secondary circuits. The measurement and simulation results showed there was unbalanced chilled water flow rate between primary and secondary circuits and the chilled water flow rate did not comply with rule of flow 'primary flow must always be equal to or greater than secondary flow' resulted in the cooling load coils lost their cooling capability. Optimization on the chilled water flow rate between primary and secondary circuits by implementing variable speed pumping system incorporated balancing valve was estimated to be potentially improve temperature performance and economic viability of the chiller plant operation.

1. Introduction

Recently energy consumption in building sector has become an important issue because of its intensive energy use and because of the rapid growth of this sector. In developed countries, domestic and commercial sectors could consume as high as 20% up to 40% of energy usage [1]. In Indonesia, the commercial building sector consumed about 5.3% of the country total energy use with energy utilization growth of 5.68% per year [2]. In this building sector, energy consumption of air conditioning (AC) systems is remarkably substantial. It could reach 50% of the building energy use [1,3]. Further studies on other commercial buildings have shown that AC system was the single largest electricity consuming service reaching a value of 47-54% of the total consumption followed by lighting with energy use of 33-38% [4]. For most modern commercial buildings, AC system and electric lighting are very important because modern buildings tend to have much larger lighting load, higher businesses density and, accordingly, larger space cooling demand. The increase of energy use is one of the main issues that raised by worldwide use of AC systems, which could be a downside due to its growth in developing countries [5].

Studies on the energy use of AC systems for commercial buildings especially hotel and office buildings have been reported in [6,7]. AC systems consumed the highest electrical energy among the buildings' services. Therefore, implementation of energy saving strategy for AC system would meaningfully contribute to energy conservation in the buildings. Strategies in improving energy efficiency of the AC systems in commercial buildings have been reported in [8-11]. The strategies include monitoring heat exchanger approach temperatures, waste heat recovery systems, implementation of thermal energy storage, application of heat recovery system and integration of heat pump and water-cooled AC system. Another way to reduce energy use of water-cooled AC systems is by improving heat rejection in condenser and cooling tower. Optimization of heat rejection in the cooling towers would be one of key roles in improving energy use efficiency of chiller systems. Ghazani *et al*. [12] studied energy systems incorporated cooling towers. Energy-saving strategies could be enhanced by improving overall performance of energy systems including cooling towers. Optimization heat transfer on the cooling tower systems could significantly contribute to energy saving strategy [13].

Energy savings strategies on AC systems almost consider everything related to whole system including chilled-water distribution system. There are three basic arrangements of chilled-water distribution system currently in use comprising: constant primary flow (CPF), constant primary flow with variable secondary flow (CPF-VSF) and variable primary flow (VPF). It has been reported that the VPF arrangement provides the highest potential for energy savings due to the essential efficiencies of the use of variable-speed pumping. The arrangement also makes the best efficient practice of over pumping to mitigate low Delta-T syndrome [14,15]. Moreover, balance variable flow chilled water system could also affect energy use and energy cost. This has comprehensively been discussed in [16]. Many balancing options were compared in detail including the first costs and energy costs of the various balancing options.

This paper presents results of simulation and optimization on chilled water distribution systems of a central air conditioning (CAC) applied for a five-star hotel in Bali Island, Indonesia. This optimization was performed to evaluate potential implementation of energy conservation and to improve temperature performance of chilled water distribution system satisfying room temperature requirement. The analyses on the chilled water circuit and its optimization are performed. Results of thermodynamic analyses on the energy performance of CAC system are also discussed.

2. Methodology

Study and optimization on chilled water distribution system of a central air conditioning (CAC) system has been conducted at a five-star hotel in Bali Island-Indonesia. The simulation and optimization were performed because temperature of chilled water distributed to the loading system could not satisfy room temperature requirement. The CAC system comprises three water cooled chillers, distribution pumps and cooling tower (CT) systems. Distribution pump systems consist of 3 primary-chilled water (ChW) pumps, 8 secondary-chilled water pumps and 3 pumps for cooling water (CW) distribution system. The chillers and pump systems are located in the plant room. The cooling towers are situated on the rooftop of the hotel building. Schematic diagram of the CAC system is illustrated in Figure 1.

The water-cooled chillers absorb heat from the chilled water and reject the heat to the ambient air through the cooling tower (CT) systems. The water-cooled chillers incorporate hermetic centrifugal compressors with R-134a refrigerant. Each chiller has a cooling capacity of 400 tons of refrigeration (TR). It is common on many hotels that only two chillers can handle the total calculated load. One chiller is prepared as spare refrigeration chiller to make sure continuous services of 24 hours per day and 365 days in a year. The spare chiller is also planned to provide enough time for maintenance and reparation due to service from the manufacturer or other service agencies as well as spare parts are not readily available. The CAC system provides cooling to the service facilities of the hotel through two loops comprise primary and secondary loops. Primary loop consists of the water-cooled chillers, primary pumps, and interconnecting pipe and utilizing constant-flow pumping system. While secondary loop comprises AHU and FCU coils, secondary pumps, interconnecting pipe and utilizing variable-flow pumping. A common pipe (also known as de-coupler pipe) hydronically separates the two loops, allowing temperature exchange but individual pressures and flows. From Figure 1, it can also be seen that the chilled water distribution system applies a CPF-VSF (constant primary flow with variable secondary flow) configuration but using three-way valves for the AHUs (Air Handling Units) and two-way valves for the FCUs (Fan Coils Units) as its flow controller. The common CPF-VSF arrangement employs only two-way valves [14].

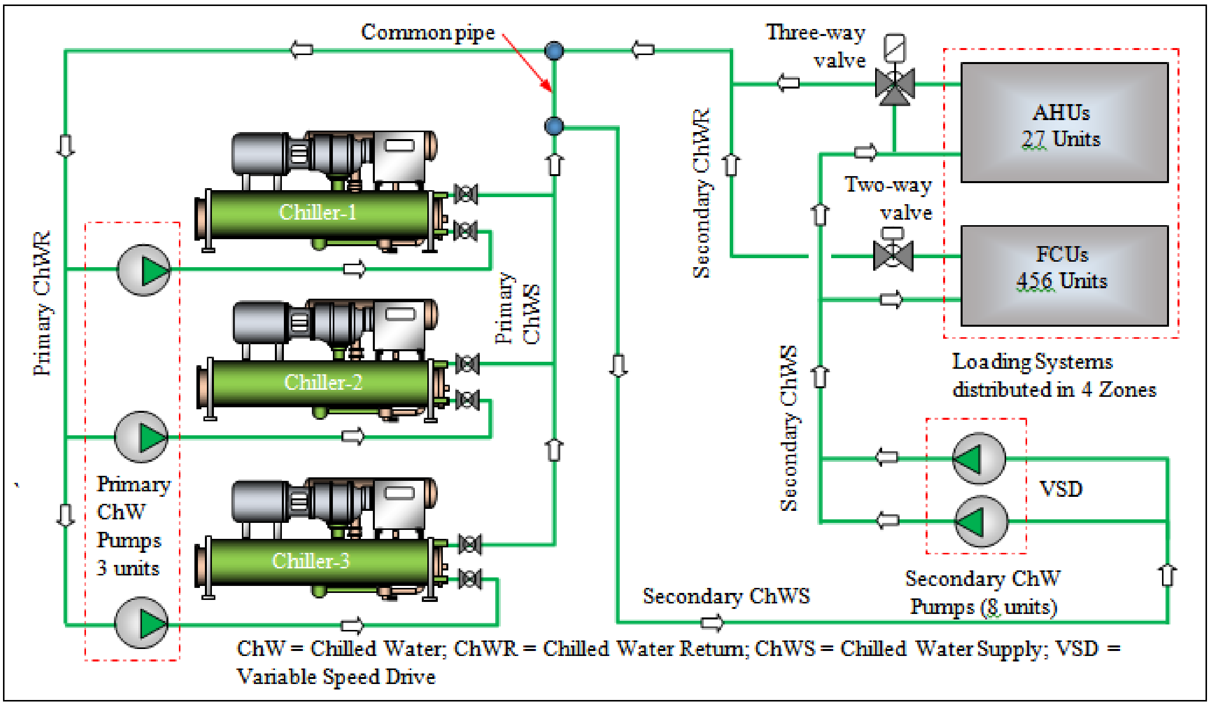


Figure 1. Schematic diagram of the investigated central air conditioning system.

Detailed measurements have been taken from the CAC system included refrigeration system of the chillers, chilled water and cooling water distribution systems, cooling load systems, heat rejection system and pumping systems. Some data were also retrieved from operational data logged-sheet. The data were processed and analyzed by using spreadsheet and EES (Engineering Equations Solver) programs. For investigation of the chillers performance include cooling capacity, COP (Coefficient of Performance) and kW/TR (Overall Efficiency), a simple refrigeration cycle models were developed. Input parameters include high and low pressures, temperatures of refrigerant at suction and discharge line of the compressor and at refrigerant liquid line, chilled and cooling water temperatures as well as cooling water flow rate. A numerical analysis has been established to simulate chilled water flow rate and temperature at primary and secondary loops.

COP of the chiller system is determined from Equation (1), where Q_{evap} = cooling capacity (kW) and W_{comp} = compressor power consumption (kW).

$$COP = \frac{Q_{evap}}{W_{comp}} \tag{1}$$

Cooling capacity (Q_{evap}) is calculated from Equation (2), where V_{ChW} = chilled water flow rate in m^3/s, EChWT = entering chilled water temperature (ºC), LChWT = leaving chilled water temperature (ºC), C_p = specific heat of chilled water (kJ/kgK), and ρ = density of chilled (kg/m^3).

$$Q_e = \rho . V_{ChW} . C_p . (EChWT - LChWT) \tag{2}$$

Overall efficiency (kW/TR) is calculated from Eq. (3). Where Q_{evap} is cooling capacity in TR (tons of refrigeration) and W_{comp} is power consumed by compressor in kW.

$$kW / TR = \frac{W_{comp}}{Q_{evap}} \tag{3}$$

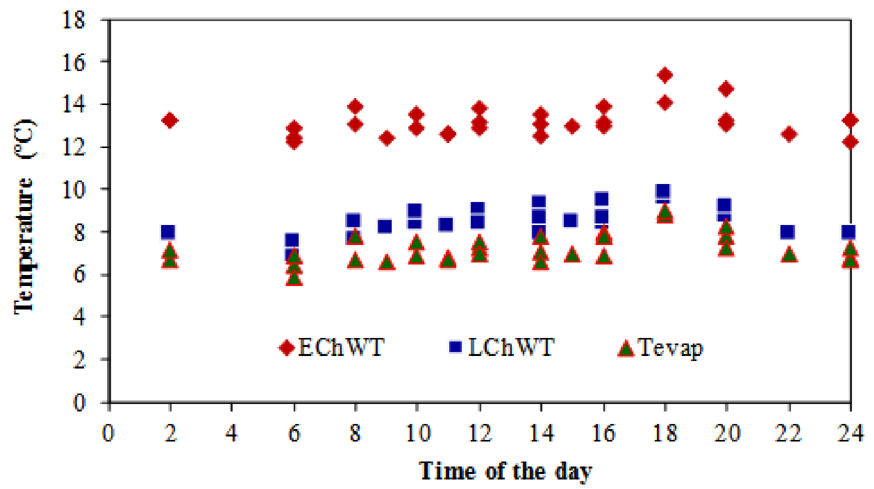
Temperature of the secondary chilled water supply (T_{SChWS}) with flowrate of V_{SChWS} is determined by using mixing flow concept between the flow from primary chilled water supply (V_{PChWS}) of temperature T_{PChWS} and the flow in the common pipe (V_{SChWS} - V_{PChWS}) with temperature equals to the secondary chilled water return (T_{SChWR}). Therefore, T_{SChWS} can be determined from Equation (4).

$$T_{SChWS} = \frac{V_{PChWS} T_{PChWS} + (V_{SChWS} - V_{PChWS}) T_{SChWR}}{V_{SChWS}} \tag{4}$$

3. Results and discussion

3.1. Operational conditions of the investigated chillers

The operation of the water-cooled chillers is supported by cooling towers, AHUs and FCUs, cooling water pumps and chilled water pumps, and is centrally controlled from a control room. This type of control is certainly more responsive compared with the manual control system and could provide efficient operation. Operational temperature of heat absorption side of the chillers system is illustrated in Figure 2. Evaporation temperature of refrigerant (T_{evap}) can reach as low as 6 ºC and chilled water temperature leaving the evaporator (LChWT) of about 7 ºC which occurs at around 4 to 6 a.m. and gradually increases during the day. This indicated that the lowest cooling demand of the hotel occurs at around 4 a.m. to 6 a.m. While the lowest chilled water temperature entering the evaporator (EChWT) or returning from the loading system is about 12.5 ºC.



EChWT = Entering Chilled Water Temperature; LChWT = Leaving Chilled Water Temperature; T_{evap} = Evaporation temperature

Figure 2. Operational temperatures of heat absorption in the evaporator.

Evaporation temperature increases during the day when the cooling demand increases until about 6 p.m. and go down through the night when the demand decreases reaching the lowest temperature at about 6 a.m. This indicates that the chiller cannot satisfy the cooling demand when the load demand increases especially occurs from 2 p.m. to 8 p.m. with evaporation temperature ranging from 7 to 9 ºC and LChWT in the range between 8 and 10 ºC. The evaporation temperature is far too high from design temperature of 6 ºC.

The variation of condensing temperature (Tcond) and temperatures of cooling water entering and leaving condenser (ECWT and LCWT) are shown in Figure 3. Condensing temperature of the chiller is quite stable ranging from 36 to 39 ºC with average temperature of about 38 ºC. This shows that the capacity of cooling towers could be large enough to satisfy heat rejection required by the chiller. Ambient air temperature during the tests varies from 28 to 30.5 ºC.

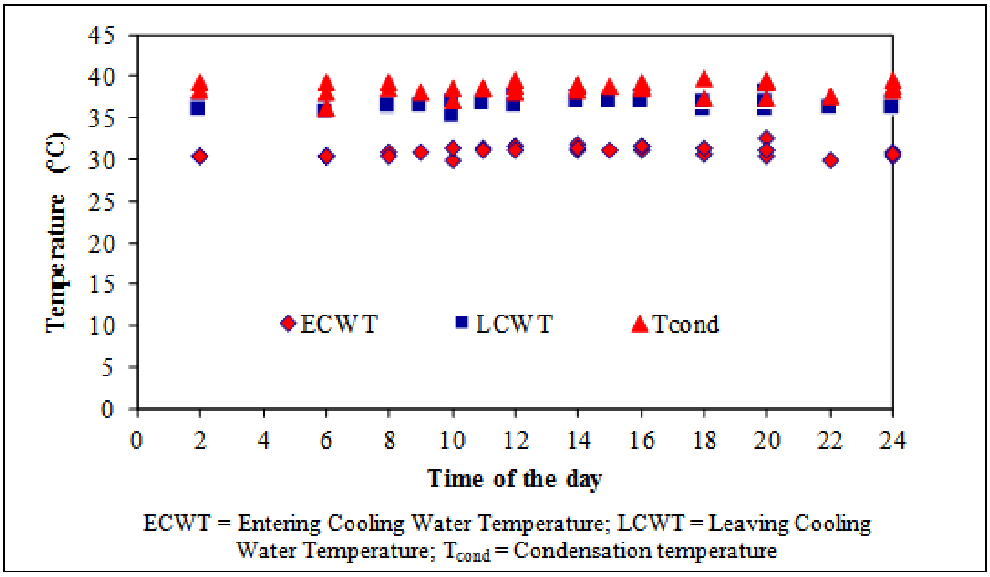


Figure 3. Operational temperatures of heat rejection process in condenser.

From Figures 2 and 3, it can also be identified temperature differences between refrigerant evaporation and leaving chilled water temperatures (LChWT) and between refrigerant condensation and leaving cooling water temperatures (LCWT). First temperature difference is commonly called evaporator approach temperature and the latter is known as condenser approach temperature. The evaporator approach temperature ranges from 0.6 to 1.6 K and condenser approach temperature varies from 0.5 to 3.6 K. The data were collected during the test to be combined with some data obtained from log sheet. In order to maintain a good chiller performance, evaporator and condenser pipes should be cleaned when approach temperatures reaching 2 K as recommended in [11].

3.2. Thermodynamic analysis on the chiller's energy performance

Thermodynamic analysis was established to evaluate energy performance of the chiller system incorporating cooling tower systems as heat rejection circuit. Performance parameters of the chiller system during investigation is shown in Figures 4 and 5.

Figure 4 illustrates cooling capacity of the chiller during one-day investigation. The cooling capacity ranges from 295 TR to 346 TR. The lowest cooling capacity of the chiller which also meant the lowest cooling demand of the hotel is consistently found at about 6 a.m. Where the COP and overall efficiency of the chiller system were 4.5 and 0.78 kW/TR respectively as shown in Figure 5. These results were achieved at evaporation temperature 6.4 ºC and condensing temperature 36.3 ºC. While chilled water temperatures: EChWT = 12.4 ºC and LChWT = 7.3 ºC and cooling water temperatures: ECWT = 30.6 ºC and LCWT = 35.8 ºC. The highest cooling capacity occurred in the middle of the day and at around 10 p.m. of cooling capacity 346 TR. The COP and overall efficiency were 5.1 and 0.68 kW/TR respectively. These results were achieved at evaporation temperature 7 ºC and condensing temperature 37.8 ºC. Chilled water temperatures EChWT and LChWT were 12.6 ºC and 7.9 ºC while cooling water temperatures ECWT and LCWT were 30.0 ºC 36.9 ºC respectively.

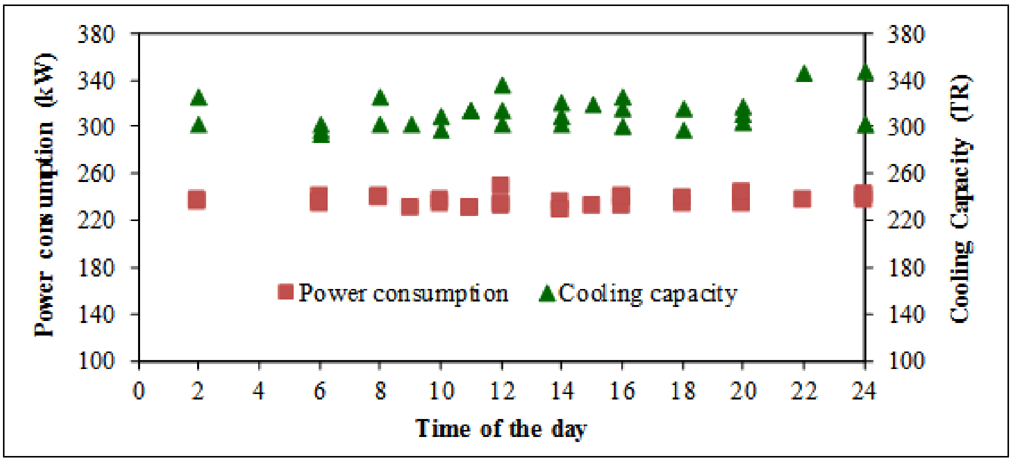


Figure 4. Chillers' cooling capacity and power consumption.

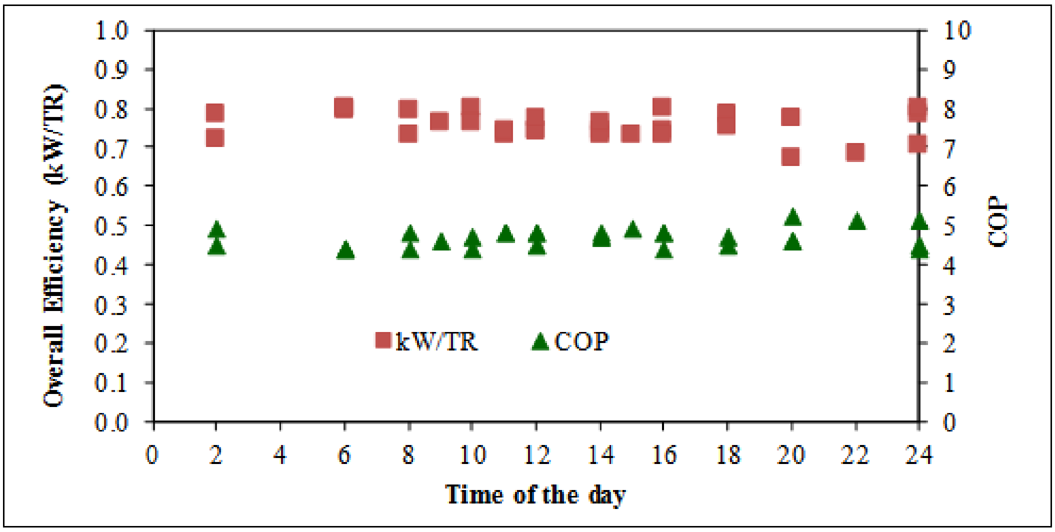


Figure 5. Energy performance COP and kW/TR of the chiller system in 24 hours' test.

3.3. Numerical simulation to optimize chillers' temperature performance

The investigation has found that supply temperature of chilled water at secondary loop to the loading system (AHUs and FCUs) is about 2 K higher than chilled water temperature supplied from the chiller in the primary loop. Although, heat losses along distribution pipes of chilled water supply and return could be negligible with a temperature drop measured from supply points of the secondary loop and inlet pipes of FCUs and AHUs was generally in an acceptable range from 0.1 to 0.2 ºC. The temperature difference has made chilled water temperature entering the loading systems (AHUs and FCUs) was about 2 K higher than chilled water temperature delivered from the chiller. This could cause AHUs and FCUs would be very difficult to achieve the designed room temperature.

In order to investigate temperature difference in the chilled water circuits, a numerical simulation has been established to optimize chilled water flow rate and temperature at primary and secondary loops. Results of the simulation are presented in Figure 6. The temperature difference between temperatures of primary and secondary chilled water supply is mainly caused by unbalanced chilled water flow rate at secondary and primary chilled water loops. When the flow rates of chilled water at both loops are the same, temperature difference is disappeared. It means when the flow rate in the secondary chilled water loop exceeds the flow rate in the primary loop, some of the chilled water discharged from the coils (AHUs and FCUs) is bypassing the chillers and flowing back to the secondary pumps through the common pipe, mixing with primary chilled water supply and increasing the temperature of water supplied to the secondary chilled water loop. If that continues to happen, the AHUs and FCUs can lose their cooling capability, and the control of the system can be lost [14]. The higher the bypassing flow rate through the common pipe is the bigger the temperature difference as shown in Figure 6.

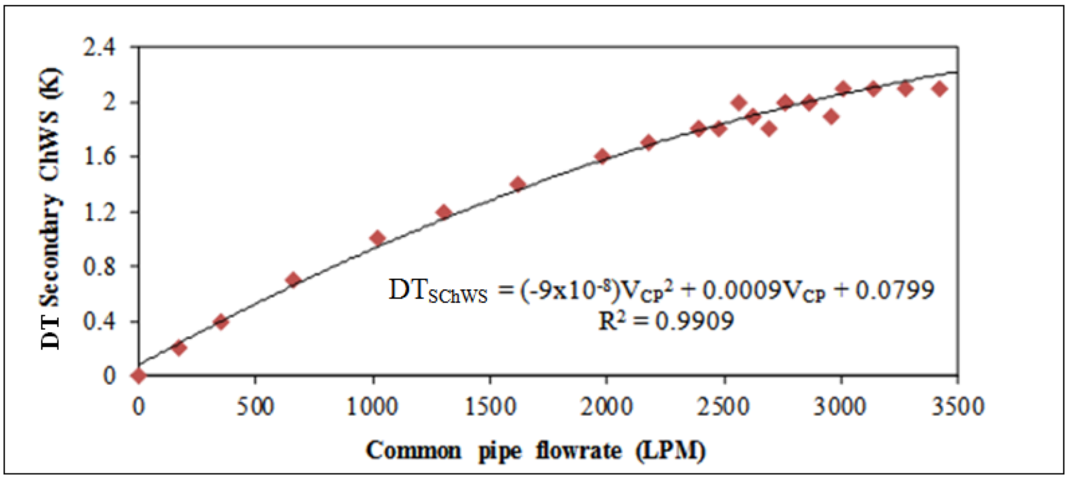


Figure 6. The increase of secondary chilled water supply temperature (DT_{SChWS}) over primary chilled water supply temperature (T_{PChWS}) at different common pipe flow rate (V_{CP}).

Polynomial regression of order 2 shows the relationship of the temperature increase of the secondary chilled water supply (DT_{SChWS}) and the flow rate in the common pipe (VCP) and have a very strong correlation with coefficient of determination $R^2 = 0.99$ (Figure 6). The analysis also shows that higher flow rate at secondary loop results in temperature difference across the loading system (AHUs and FCUs) decreases to 3 K which is much lower than designed temperature difference of 5.7 K. This is known as low Delta-T syndrome [14,15]. This can also cause energy pumping for chilled water at secondary loop increases. Based on the analyses results, this study proposes the flow rate of chilled water at secondary loop should be maintained the same or slightly lower than the flow rate in the primary loop. The installation of balancing valves (two-way valves) on the AHU systems as replacement of the existing three-way valves can provide advantages to the chiller system which include: chilled water flow rate between primary and secondary loops can be easily balanced; the occurrence of low Delta-T syndrome can be minimized; energy use for pumping systems can be reduced; and the installation of two-way valves is found to be fairly economic of payback period less than 1.5 years. This is in agreement with solution alternatives reported in [15,16].

4. Conclusions

Experimental and numerical optimization on constant flow primary and variable flow secondary (P/S) chilled water distribution of a centralized chiller system have been established. The experimental and simulation results showed there was unbalanced chilled water flow rate between primary and secondary circuits. The flow rate in the secondary chilled water loop exceeds the flow rate in the primary circuit and some of the chilled water discharged from the AHUs and FCUs bypasses the chillers through the common pipe which causes the temperature of water supplied to AHUs and FCUs increases and it can further cause low Delta-T syndrome. To optimize the chilled water distribution system, the study proposes installation of two-way valves on the AHU systems as replacement of the existing three-way valves. The proposed configuration can provide advantages to the chiller system include ease in regulating flow balance between the circuits, minimum occurrence of low Delta-T syndrome, reduction of energy consumption for pumping and more economical chiller operation.

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