Experience of using a chilled water circuit design to expedite *in situ* chiller performance measurement

Yik F. W. H.,* DEng, PhD, CEng, FCIBSE, MIMechE, MHKIE
Lee S. H., BEng, MEng
Lai J. H. K., PhD, CEng, MCIBSE, MHKIE
Chan K. T., PhD, CEng, FCIBSE, MHKIE

Department of Building Services Engineering, The Hong Kong Polytechnic University, Hunghom, Kowloon, Hong Kong SAR, China

Abstract

A chilled water circuit design that can facilitate expeditious *in situ* chiller performance measurement has been adopted in a chiller plant and its effectiveness has been demonstrated. Before chiller tests commenced, the measurement accuracies of the sensors in the plant were checked by simple measurements and an analysis of the plant operation records. The analysis unveiled that the chilled water flow rate through the chillers was lower than the design flow rate and the chilled water return temperature often stayed below the design level due to excessive flow rate demand, which would hinder full load tests on the chillers. After the causes of the problems were diagnosed and the problems resolved as far as possible, measurement of the full- and part-load performance of the newly installed chillers was successfully conducted. This work demonstrated that the chilled water circuit design is effective in facilitating expeditious *in situ* chiller performance measurement but its use requires a properly functioning chilled water system.

Practical application

As the results presented in the paper show, the proposed chilled water circuit design is effective in reducing the time and effort required for measurement and verification of the full- and part-load performance of water chillers. This, in turn, allows more frequent chiller performance measurements to be made for detecting system or sensor faults and deterioration in chiller performance, which would help upkeep chiller energy efficiency and reliability. Described in the paper includes also several practical methods for verification of sensor accuracy and some common problems with chilled water systems where fan-coil units with on/off control are adopted.

Keywords: Chiller performance; *In situ* measurement; Chilled water circuit design; Pilot tests

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* Corresponding author; email: bewhyik@polyu.edu.hk

Introduction

Two chilled water circuit designs have been proposed,[1] one for a single-loop and the other for a two-loop system (Figure 1), to facilitate expeditious *in situ* measurement of chiller performance that can cover the entire output range of the chillers. In addition to the normal (differential pressure or de-coupler) bypass pipe at one side (the load side) of the group of chillers (pipe FC in Figure 1), the proposed chilled water circuit includes an alternative bypass pipe at the other side of the chillers (pipe HG in Figure 1). With this circuit design, the chiller plant may be operated in the normal operation mode or the measurement mode.

In the normal operation mode, the valve in the alternative bypass pipe (pipe HG) shall be tightly shut and the chiller plant will operate in exactly the same way as any plant with the conventional chilled water circuit design only. While operating in this mode, the running chillers will each have a share of the total cooling load on the plant at the same part-load ratio (actual-to-full load ratio) and any surplus chilled water will flow through the normal bypass pipe. However, this equal proportion load sharing characteristic limits the frequency of occurrence of full-load and near zero-load on the chillers, and thus measurement and verification (M&V) of their performance over their entire output range would need to span a long time period to allow sufficient data to be captured, which is a hindrance to M&V of chiller performance. The proposed circuit design is meant to address this hindrance inherent in the conventional chilled water circuit designs.[1]

For a single-loop system (Figure 1.a), the valve in the alternative bypass pipe (pipe HG) shall be a control valve identical to the differential pressure bypass control valve (DPV) in the normal bypass pipe (pipe FC). When switched to the measurement mode, the DPV in the alternative bypass pipe will take over the role of the DPV in the normal bypass pipe, i.e. to let surplus chilled water to flow through the alternative bypass pipe so as to maintain the differential pressure across the main supply and return pipes at the set point level. The DPV in the normal bypass pipe will serve another function, which is to let pass a required flow rate of the supply chilled water (at T_{ss}), which will mix with the return chilled water from the air-side systems (at T_{sr}) such that the chilled water entering the chillers will be kept at the rated temperature ($T_{r,rated}$). This function is needed whenever T_{sr} exceeds $T_{r,rated}$, to safeguard the chillers from being overloaded.

For a two-loop system (Figure 1.b), the valve in the alternative bypass pipe (pipe HG) shall be an isolation valve, which shall be fully open in the measurement mode while the isolation valve in the normal de-coupler bypass pipe (pipe FC) shall be tightly shut, such that the alternative bypass pipe will serve as the de-coupler bypass pipe. There is an additional bypass pipe (pipe JK) with a control valve in it to control the temperature of chilled water entering the chillers from overshooting $T_{r,rated}$. This additional bypass control valve will be operated only in the measurement mode, but shall be tightly shut in the normal operation mode.

In the measurement mode, a chiller will be loaded up to its full capacity before an additional chiller needs to be run to cope with a rising total load, as long as T_{sr} stays at or above $T_{r,rated}$. When multiple chillers are running, all but one of the running chiller(s) will be loaded steadily at their full capacity (or at a lower level if $T_{sr} < T_{r,rated}$). The exceptional chiller, referred to as the 'last' chiller, may be loaded at any level between 0 to 100% of its full capacity, depending on the total load on the plant. If no further provisions are made, the unit among the running chillers that is located the furthest away from the load side will become the last chiller. This is because mixing of the surplus chilled water from the alternative bypass pipe at T_{ss} and the return chilled water at either T_{sr} (if $T_{sr} < T_{r,rated}$) or $T_{r,rated}$ (if $T_{sr} > T_{r,rated}$) will take place only at the entry point to the 'last' chiller whereas all other running chiller(s) will be fed with chilled water at either T_{sr} or $T_{r,rated}$.

The circuit design includes auxiliary pipes with isolation valves to allow selection of any chiller in a plant to become the 'last' chiller (Figure 2).[1] With the circuit design and through the use of a building management system (BMS), sufficient full- and part-load performance data for each chiller can be recorded within a much shorter period of time than with just the conventional circuit design, which can greatly facilitate performance evaluation of chillers and for development of chiller models for fault detection and diagnostics.[2, 3]

The proposed chilled water circuit design has recently been installed in a chiller plant when the plant was retrofitted. This provided an opportunity to test the effectiveness of the proposed chilled water circuit design, including making observations on whether any problems could arise which may hinder its successful application.

Characteristics of the chiller plant studied

Configuration of the retrofitted plant

The study was based on a chiller plant in a tertiary education building in Hong Kong. Since the five original air-cooled chillers in the plant were aging and the regulatory control that prohibits use of fresh water from city mains for air-conditioning purposes has been relaxed in Hong Kong,[4] the building owner recently decided to retrofit the chiller plant in the interest of energy saving. Three of the five old air-cooled chillers have been replaced by three new water-cooled chillers, each with a rated cooling capacity of 400 tons of refrigeration (TR; 1TR = 3.517kW), which can collectively cope with the peak cooling demand of the building. The remaining two air-cooled chillers are retained for standby purpose. The schematic diagrams of the chilled and condenser water systems, after the chiller replacement and pipe modification, are as shown in Figures 2 & 3.

The chiller plant is equipped with a two-loop chilled water distribution system (Figure 2). There are five constant speed primarily-loop chilled water pumps (PCWPs), each connected in series with the chiller it serves, and three variable speed secondary-loop chilled water pumps (SCWPs). Nearly all indoor spaces in the building are equipped with fan-coil units with two-way control valves. The three new water-cooled chillers (Chillers 2 to 4) are served by three cooling towers with matching heat rejection capacity and four condenser water pumps (CWPs), with one for standby (Figure 3). The condenser water circuit includes a bypass control system for ensuring the temperature of the condenser water entering the chillers will not fall below a minimum preset level. Table 1 summarizes the key performance characteristics of the major equipments in the plant.

Chiller sequencing control

Chiller operation is under the control of a BMS. According to the BMS operator, the chiller plant is run throughout the year, from 7:00 am till 11:00 pm every day. The chillers and chilled water pumps are started or stopped based on the cooling demand determined from measurements of the temperatures and flow rate of chilled water in the main supply and return pipes in the secondary-loop. One more group of chiller and primary-loop pump will be started if the cooling demand overshoots the total cooling capacity of the running chiller(s) and such condition has lasted continuously for 20 minutes. On the contrary, one chiller and one primary-loop pump will be stopped if the total cooling capacity of the running chiller(s) is greater than the cooling demand by 110% of the rated capacity of one running chiller and such condition has lasted continuously for 20 minutes.

Available plant operation records

The chiller plant is equipped with a variety of measuring instruments, which include discrete temperature sensors and flow meters installed at various locations in the chilled and condenser water circuits (Figures 2 & 3) as well as built-in sensors that came with the chillers. These sensors are connected to the BMS for control and performance monitoring purposes. Table 2 shows their measurement accuracy, as found from manufacturers' technical data. The chiller plant operation records, which cover the range of system variables as summarized in Table 3, were retrieved from the BMS and analyzed in the present study.

The chillers tested in the study

Since the building owner was convinced to adopt the proposed chilled water circuit design after the contract for chiller replacement has been awarded, permission was given to conduct tests on the newly installed water-cooled chillers (Chillers 2 to 4) only after the contractor has handed over the retrofitted plant to the owner, and the findings of the study would not affect in any way the contractual obligation of the contractor. Furthermore, as the standby air-cooled chillers (Chillers 1 & 5) had not been run throughout the study, no tests had been done on the two standby air-cooled chillers.

Verification of sensor accuracy and plant operating conditions

Water temperature sensors

Since neither auxiliary thermometers nor thermal wells were available in the piping system, simple *in situ* measurements, which involved releasing chilled water from the pipe and measuring the temperature of the running water using a mercury-in-glass thermometer, had been conducted at locations where release taps are available and are close to the water temperature sensors. Comparison of the measured chilled water temperatures and the corresponding readings obtained through the BMS unveiled that the chilled water supply and return temperature sensors in the secondary-loop and inside individual chillers were all reasonably accurate; the deviations between the sensor output readings and the *in situ* measurements were within ±0.3°C, as shown in Table 4.

Besides the simple temperature measurements, further analysis was conducted to verify if the measurements of the installed sensors would remain reasonably accurate at other values of the measured variables within the feasible operating range. The analysis, as will be presented below, was limited to relative comparisons based on about 200 sets of records retrieved from the BMS, which were recorded at 30 minutes intervals in September 2008.

The measurements of the main chilled water supply temperature sensor (T_{ss}) were taken as the reference for comparison with the measurements of the built-in supply chilled water temperature sensors in individual chillers (T_{si}; i = 2, 3, 4). Since the three chillers are identical, the measured values of T_{s2} to T_{s4} should be close to each other and to the measured values of T_{ss}; any large deviations would point to presence of faulty sensors. The results of a statistical analysis on the deviations of T_{s2} to T_{s4} from T_{ss}, as summarized in Table 5, show that the mean bias error and the root mean square error of T_{s2} to T_{s4} from T_{ss} are small enough to support the assumption that the sensors were in normal working order.

With the plant operating in the normal mode, the return chilled water temperature at the chillers (T_{ri}; i = 2, 3, 4) should all be equal to the mixed temperature (T_r) of the chilled water returning from the secondary-loop (at T_{sr}) and the bypass chilled water (at T_{ss}) whenever there is surplus flow through the bypass pipe (i.e. $m_{by} \geq 0$). If there is deficit flow (i.e. $m_{by} < 0$) instead, T_r should be equal to T_{sr}, which should occur only momentarily and under that condition, at least one more chiller unit should be started to avoid prolonged occurrence of deficit flow. On this basis, T_r can be estimated from T_{ss} & T_{sr} and the supply and bypass flow rates (m_{sec} & m_{by}; assuming both can be accurately measured), as depicted by equation (1). T_r so determined was taken as the reference for verifying whether the sensor for measuring the common chilled water return temperature to the chillers (denoted as T_{rch}) and the return chilled water temperature sensors inside individual chillers (T_{ri}) had significant measurement errors.

$$T_r = \begin{cases} \dfrac{m_{by}T_{ss} + m_{\sec}T_{sr}}{m_{by} + m_{\sec}}; m_{by} \geq 0 \\ \\ T_{sr}; m_{by} < 0 \end{cases} \tag{1}$$

Results of the statistical analysis, as shown in Table 5, show that the mean bias error and root mean square error of T_{rch} and of T_{r2} to T_{r4} from T_r were all within ±0.3°C. Hence, these sensors may be regarded as in normal working order.

Chilled and condenser water flow rates

Table 6 summarizes the results of a statistical analysis on the chilled and condenser water flow rate data measured by the built-in flow meters in the chillers, which were included in the abovementioned set of BMS records. In this analysis, the mean values of the measured chilled water flow rates through the three chillers were found to be in the range of 49 to 53*l/s*. Notwithstanding that these mean flow rates were deviating from each other by no more than 4*l/s*, they were significantly lower than the rated flow rate, which should be at 67*l/s*. In response to our enquiry, the plant operator confirmed that the chilled water pumps were under-sized (but this was a problem that the building owner should deal directly with the contractor and we were not in a position to interfere). The mean condenser water flow rates through Chillers 2 to 4, also shown in Table 6, were in the range of 87 to 89*l/s* which are reasonably close to the chillers' rated condenser water flow rate (83.8*l/s*).

As no extra flow meters that could provide accurate measurements for comparison were available, verification of the measurement accuracies of the chilled and condenser water flow meters was based on the water pressure drops across the chiller evaporators and condensers, and further on the pumping pressures and the characteristic curves of the primary-loop chilled water pumps and condenser water pumps, albeit such references could only provide a crude comparison.

According to the manufacturer's catalogue, the evaporator and condenser water pressure drops of the chillers would be 110kPa and 58kPa when the chilled and condenser water flow rates were at their respective rated values, i.e. 67*l/s* and 83.8*l/s*. Based on this information, the flow resistances (K) of the evaporator and condenser of the chillers were calculated using equation (2), to allow the water flow rates to be estimated based on measured evaporator and condenser pressure drops, using equation (3).

$$K = \Delta p / V^2 \tag{2}$$

$$V = \sqrt{\frac{\Delta p}{K}} \tag{3}$$

The pressure drops across the evaporator and the condenser of the chillers, as found from a site inspection, and the corresponding estimates of the chilled and condenser water flow rates, are summarized in Table 7. The deviations between the mean chilled and condenser flow rates determined from measurements of the built-in flow meters in the chillers (Table 6) from these estimated flow rates are within 7%.

Another set of flow rate estimates was made with reference to the pumping pressures of the primary-loop chilled water pumps and condenser water pumps found during the site inspection in conjunction with the characteristic curves of the respective pumps (Figure 4 and

Tables 8 & 9). The flow rates through the primary-loop chilled water pumps and the condenser water pumps so determined were also found to be close to the respective mean values of the flow rates across the chillers as measured by their built-in sensors.

Besides the built-in flow meters in the chillers, the chilled water system was equipped with flow meters for measuring the total chilled water flow rate in the secondary-loop (m_{sec}) and the flow rate through the de-coupler bypass pipe (m_{by}). The measurement accuracies of these flow meters were also checked using the following method.

Based on the principle of mass conservation, the total chilled water flow rate through the operating chillers (Σm_i) should be equal to the sum of the chilled water flow rate in the secondary-loop and the bypass flow rate (m_{sec}+m_{by}). The bypass flow rate (m_{by}) would be a positive value if it is a surplus flow but it would be taken as a negative value if it is a deficit flow (Figure 2). Accordingly, the residual value (R_m), depicted by equation (4), should be equal to zero; a residual value that differs significantly from zero would imply abnormality in the flow rate measurements.

$$R_m = m_{\text{sec}} + m_{by} - \sum m_i \quad (4)$$

For a variable (R) that is dependent on a range of other variables (V_i; i = 1, 2, …, n), as shown in equation (5), when the uncertainties (δV_i) in the measurements of the independent variables are known, the uncertainty in the value of R (δR) estimated from the measurements can be quantified using equation (6).[5, 6]

$$R = f\,(V_1, V_2, \ldots, V_n) \quad (5)$$

$$\delta R = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial R}{\partial V_i} \cdot \delta V_i \right)^2} \quad (6)$$

Where

R = an estimate made from measurements of the variables V_1 to V_n

δR = uncertainty in the estimate R

δV_1 to δV_n = uncertainties in the measurements of the variables V_1 to V_n

Equation (7), derived from equations (4) & (6), was used to evaluate the uncertainty in the values of R_m estimated using equation (4). Accordingly, the estimated values of R_m may stay with the upper and lower limits $R_m+\delta R_m$ and $R_m-\delta R_m$ respectively, as any residual flow rates of magnitudes within these uncertainty limits could arise simply because of the uncertainties in the variables used in their estimation. In the calculation, the measurement accuracies of the flow meters (Table 2) were taken as the uncertainties in the flow rate measurements, i.e. δm_i, δm_{by} and δm_{sec}.

$$\delta R_m = \sqrt{\sum(\delta m_i^{\ 2}) + \delta m_{\sec}^{\ 2} + \delta m_{by}^{\ 2}} \qquad (7)$$

An analysis of the residual flow rates calculated from the plant records (Figure 5) showed that 93% of the residual flow rates fell within the uncertainly limits, which indicates that all flow meters, including the chilled water flow meters in individual chillers, the bypass flow meter and the secondary-loop chilled water flow meter, may be regarded as reasonably accurate.

Power measurements

In situ, snap-shot electrical measurements had been performed using a portable power analyzer (model: Fluke 41b) to verify the accuracy of the measurements of the electric power meter in each chiller on 11, November 2008, which indicated that all power measurements were within ±3kW from the corresponding readings of the portable power analyzer. Continued measurement using the power analyzer was also conducted to verify the readings of the power meter in Chiller 4. As shown in Figure 6, there is a linear correlation between the power demands of the chiller measured by the power analyser and those measured by the built-in power meter of the chiller. The mean bias error and the root mean square error were −0.89kW and 7.38kW, equivalent respectively to −0.35% and 3% of the rated power demand of the chiller (254kW). Despite the significant root mean square error, the power meters may be considered to be capable of providing reasonably accurate power demand measurements for the purpose of the present study.

Low chilled water return temperature

Note that the proposed chilled water circuit design would be effective in keeping all but one of the running chillers steadily at the full-load condition provided that the rated chilled water flow rate is maintained, the supply chilled water temperatures from all chillers (T_{si}; i = 2, 3, 4), and thus also the supply chilled water temperature at the secondary-loop (T_{ss}), stay at the rated supply temperature ($T_{s,rated}$) and, at the same time, the temperatures of the return chilled water entering all but the last chiller (T_{si}) stay at the rated return chilled water temperature of the chillers ($T_{r,rated}$). For checking whether these operating conditions were achievable, the measured readings of T_{sr} and T_{ss} were retrieved from the plant operation records for September 2008. Inspection of the data showed that T_{ss} could rise substantially above the design set point (7°C) while T_{sr} often stayed well below the design temperature (12°C) during periods when the building cooling load was lower than 1,000kW (Figure 7). Although the former would be normal if there was deficit flow, the latter condition would limit the chances of keeping chillers at full-load condition.

Low return chilled water temperature (T_{sr}) may result from one or a combination of the following conditions:

1. The chilled water control valves in the air-side systems were leaky and thus allowed chilled water to flow through the air-handling equipment while the control valves were supposed to be fully closed;

2. The pumping pressure in the secondary-loop was excessive, thus forcing too much chilled water to flow through the cooling coils in the air-side equipment;

3. Since on/off control was used for the fan-coil units connected to the system, chilled water could flow through the fan-coil units at the full flow rate but returning at a low temperature during the on-cycles in periods of low cooling load, especially when the fan-coil units were oversized (which is common for fan-coil units); and

4. The indoor set-point temperature set by the occupants was significantly lower than the design value (25°C), causing the control valves at the fan-coil units to stay more often at fully open position and the return chilled water staying at a low temperature.

Condition 1 above was found to be non-existent or, at least, not serious because, as shown in Figure 8, which was based on the operating data for September 2008, the chilled water flow through the air-side equipment approached zero together with the building cooling load. Condition 2 in isolation would not give rise to the problem but would exacerbate the problem if Condition 1 or 3 existed. For verifying if Condition 2 existed, the differential pressure setting was lowered from 80kPa to 50kPa in a trial run in October. After adjustment of the differential pressure setting, it was found that the return chilled water temperature could be kept more often around 12°C, as shown in Figure 9.

As Figure 10 shows, there was a general tendency that the temperature difference between the supply and return chilled water in the secondary-loop ($T_{sr} - T_{ss}$) would drop with the total flow rate, which may be taken as evidence of the effect of on/off control with or without oversized fan-coil units (Condition 3) and/or low indoor temperature settings (Condition 4). Since the building owner can do little on the problems caused by Condition 3 in the short term, further investigation focused on verifying if Condition 4 prevailed.

A small-scale site survey was conducted during which a sling psychrometer was used to measure the room air temperatures in six offices, including 2 general offices and 4 staff offices in the building. It was found that most of the measured room air temperatures were lower than 23°C; only one staff office was having a room temperature slightly above 25°C but the room temperature in one of the staff offices was found to be even lower than 22°C. A temperature logger was subsequently placed near to the return air grille in that office to measure the return air temperature. Figure 11 shows the temperature profile of that room over a daily cycle. There was no temperature control from 11:00 pm to 7:00 am on the next day but the room temperature stayed consistently below 22°C, sometimes approaching 19°C, throughout the air-conditioned period. This finding confirmed that low indoor temperature set point was one of the main reasons that had given rise to the problem of low return chilled water temperature.

Full range chiller performance measurement

Having verified the measurement accuracies of the sensors and despite the abovementioned imperfections in the chilled water system, measurements were carried out in the measurement mode to demonstrate the effectiveness of the proposed circuit design in facilitating chiller performance measurement. The test was done in three stages, with a different chiller selected as the ‘last’ chiller in each stage.

Activation of the measurement mode

The chilled water plant was switched to the measurement mode by opening fully the isolation valve in the alternative bypass pipe (pipe HG in Figure 2) followed by closing-off the isolation valve in the normal de-coupler bypass pipe (FC). The control set-point for the return chilled water temperature control system was set at the design chilled water inlet temperature for the chillers (12ºC) and the control system was activated such that the bypass valve (in pipe JK) would permit, when required, some chilled water from the main supply pipe to flow toward the main return pipe and mix with the chilled water returning from the air-side equipment, so as to keep the temperature of the chilled water entering the chillers from overshooting the set-point level, thus avoiding overloading the running chiller(s).

Inspection of the operating conditions of the condenser water circuit unveiled that the minimum entering condenser water temperature control system was unable to precisely control the condenser water temperature. Consequently, the condenser water temperature could overshoot the set-point level (32ºC), which could trigger the over-current protection control in the chillers. Therefore, the set-point was adjusted to 31ºC during the chiller tests.

Procedures for the Stage 1 test

The remaining test procedures are described below for the Stage 1 test where Chiller 4 was selected as the ‘last’ chiller, by opening and closing the isolation valves a to j (Figure 2) as described in Table 10. The test procedures for the other two stages were similar, except that the open/closed status of the isolation valves had to be changed according to Table 10, to allow a different chiller to be selected as the ‘last’ chiller.

In the Stage 1 test, Chiller 4 was selected as the ‘last’ chiller for part load test and was kept running throughout this stage of test. When the building cooling load exceeded the cooling capacity of 1 chiller unit but was lower than the total cooling capacity of 2 chiller units, either Chiller 2 or Chiller 3 would be run. In this stage, preference was given to start Chiller 2 (referred to as the ‘preferred’ chiller), unless it had just been stopped for a short period of time in which case Chiller 3 would be started instead. When the building cooling load exceeded the cooling capacity of 2 chiller units, both Chiller 2 and Chiller 3 would be run. When the building cooling load dropped below the total cooling capacity of two chiller units, Chiller 3 would be stopped. Likewise, if the building cooling load dropped below the cooling capacity of one chiller unit, Chiller 2 (or Chiller 3 if Chiller 2 had been stopped) would be stopped, in which case only Chiller 4 would remain running. The plant was kept operated as described above for 2 days, with the data recorded by the BMS over this period, which were extracted afterward for analysis. The Stage 2 test and, thereafter, the Stage 3 test were conducted.

Results of measurement

For demonstrating also the effectiveness of the circuit design in allowing selection of any chiller as the ‘last’ chiller, results for the Stage 2 test, during which the middle chiller, Chiller 3, was selected as the ‘last’ chiller, are presented here. During that test, the total cooling load was never high enough to require all the three chillers to be run simultaneously. The test results showed that:

1. The temperature of chilled water leaving both chillers (T_{s3} & T_{s4}) stayed rather steadily at the set point level of 7^{o}C (Figure 12);

2. Instead of being controlled to stay steadily at the set point level of 31^{o}C, the temperature of condenser water entering the chillers (T_{cde3} & T_{cde4}) varied slightly about the set point level (Figure 13);

3. The temperature of chilled water entering Chiller 4 (the 'preferred' chiller), T_{r4}, stayed rather steadily at 12°C (Figure 12), and the average part-load ratio (*PLR*) of the chiller over the test period was 0.74 (Figure 14), which matched with the ratio of the actual (51*l*/s; Table 6) to the rated (67*l*/s; Table 1) chilled water flow rate through the chiller; and

4. The temperature of the chilled water entering Chiller 3 (the 'last' chiller), T_{r3}, was variable and could drop to about 8.5°C (Figure 12), and its *PLR* varied within the range of 0.18 to 0.48 (Figure 14).

These observations are evidence that the circuit design was effective in keeping Chiller 4 loaded reasonably steadily at the highest achievable load level (it could have been fully loaded without a reduction in the chilled water flow rate) while the load on Chiller 3 (the 'last' chiller) could vary as a result of mixing of return and bypass chilled water, thereby maximizing the range of load under which the chiller operated.

Without the proposed chilled water circuit, all the running chillers would have evenly shared the total cooling load at all times. As a result, the range of operating conditions that could be captured would be significantly narrower. Figure 15 shows the frequency distribution of cooling load on the chillers over different load ranges under the measurement mode, together with the frequency distribution that would result had the plant been operated under the normal operation mode (same as with the conventional circuit design). It can be seen that the load range was expanded from over 40% to less than 90% for the latter case to over 10% to less than 90%.

The coefficient of performance (COP) of Chillers 2, 3 & 4 were calculated from the test results and are shown in Figures 16(a) to 16(c) together with the manufacturer's part load performance curve. These figures unveil that there are considerable scattering of the COP data. The uncertainties in the COP values incurred by the uncertainties in the measured data used for their estimation, which could be more than ±10% at full-load and more than ±20% at half-load,[5] should have been the major reason for this phenomenon.

Figure 16 shows also that the majority of the data points representing the COP of the chillers stay significantly below the manufacturer's curve in the *PLR* range above 0.5. In fact, plots of the COP of the chillers based on the BMS records collected before the chiller tests were conducted exhibited very similar patterns, including the slightly poorer performance of Chiller 2 (Figure 16(a)) as compared to Chillers 3 & 4 (Figures 16(a) & (b)). The reduction in chilled water flow rate (from the rated value of 67*l*/s down to 49 – 53*l*/s) is believed to be the main reason for the degradation in chiller performance.[7, 8]

As shown by the experimental results of Braun and Comstock for a water cooled centrifugal chiller with reduced evaporator water flow (Figure 4.10 in Ref. 8), this fault would have little impact on the kW/TR value (kW/TR = 3.517 / COP) of the chiller in the *PLR* range of 0.33 to 0.66 but would increase sharply with *PLR* beyond 0.66. The kW/TR value was increased by about 8% at the *PLR* value of 0.78, and by about 15% at the PLR value of 1.0, when the flow rate was reduced by 40%. For a chiller of a rated COP value of 5.5, which equals the rated COP of Chillers 2 to 4, this fault would cause the COP value to drop to 4.89 and further to 4.45 when the *PLR* value increased from 0.78 to 1.0. This pattern of COP variation matches roughly with those of the chillers being studied (Figure 16), which supports our belief.

Conclusion

The effectiveness of the proposed chilled water circuit design in facilitating *in situ* chiller performance tests has been empirically verified based on a pilot installation. Before the tests, the measurement accuracies of the available instruments were evaluated and the operating conditions of the chiller plant were analyzed. The most significant plant deficiencies found included reduced chilled water flow rate and frequent occurrence of low chilled water return temperature. With adjustment made to the differential pressure control setting, the latter problem was alleviated. Finally, *in situ* measurements were conducted and the results verified that the proposed circuit was capable of allowing the performance of chillers over a wide load range to be measured within a short period of time.

With just the conventional chilled water circuit design, full-load tests on chillers can be carried out only when the building cooling load is close to the total output capacity of the chiller plant, which will occur only within a limited period of time in a year and is thus a hindrance to chiller commissioning and performance measurement and verification. Not only this constraint can be overcome by adopting the proposed circuit design, with it, chiller full- and part-load tests can be accomplished within a short period of time (two days per chiller unit in the present study). Therefore, chiller tests can be carried out more frequently to allow close monitoring of their performance, making it possible to promptly detect any deterioration in their energy efficiency or in their ability to precisely control the supply chilled water temperature, which in turn will help minimize energy cost and ensure thermal comfort in air-conditioned spaces. The measured chiller performance may also be utilized to develop optimized sequencing control and fault detection and diagnosis strategies for chillers to further enhance energy efficiency and reliability of a chiller plant.

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References

1. Yik FWH. Chilled water circuit designs for *in situ* chiller performance measurement. *Building Services Engineering Research and Technology* 2008; 29(2): 107-118.
2. Katipamula S, Brambley MR. Methods for fault detection, diagnostics and prognostics for building Systems – A review, Part I. *International Journal of Heating, Ventilating, Air Conditioning and Refrigerating Research* 2005; 11(1): 3-25.
3. Katipamula S, Brambley MR. Methods for fault detection, diagnostics and prognostics for building Systems – A review, Part II. *International Journal of Heating, Ventilating, Air Conditioning and Refrigerating Research* 2005; 11(2): 169-187.
4. EMSD. *Scheme for Wider Use of Fresh Water in Evaporative Cooling Towers for Energy-efficient Air Conditioning Systems*. Electrical and Mechanical Services Department, the Hong Kong SAR Government 2008.
5. Yik FWH, Chiu TW. Measuring Instruments in Chiller plants and Uncertainties in Performance Evaluation, *HKIE Transactions* 1998; 5(3): 95-99.
6. Kirkup L, Frenkel RB. An introduction to uncertainty in measurement using the GUM (Guide to the expression of uncertainty in measurement). UK: Cambridge University Press 2006.
7. Cui J, Wang SW. A model-based online fault detection and diagnosis strategy for centrifugal chiller systems. *International Journal of Thermal Sciences* 2005; 44(10): 986-999.
8. Braun JE, Comstock MC. *Development of analysis tools for the evaluation of fault detection and diagnostics for chillers, HL 99-20, Report #4036-3, ASHRAE Research Project 1043*. American Society of Heating, Refrigerating and Air-conditioning Engineers, Inc., 1999.

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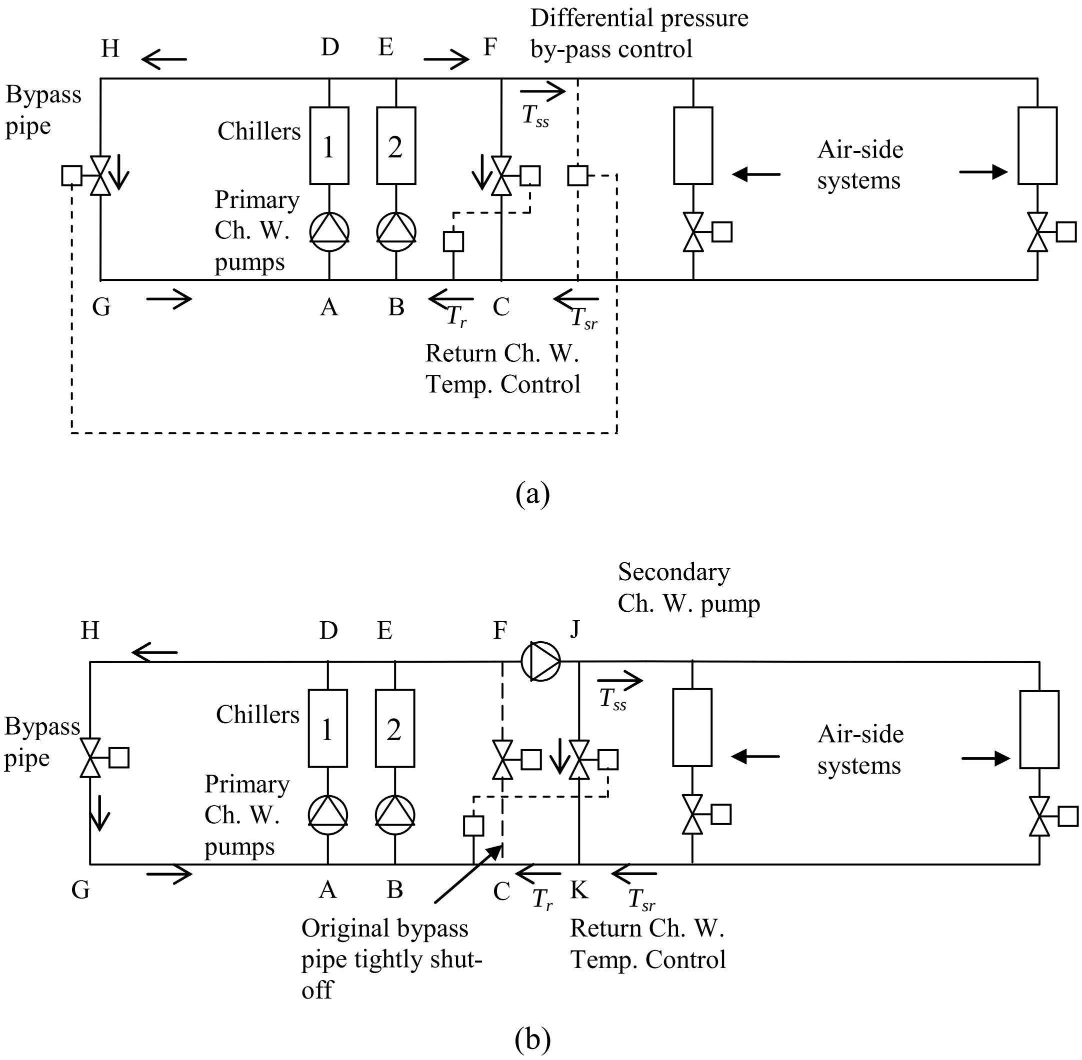


Figure 1 Proposed chilled water circuit designs: (a) single-loop pumping system; (b) two-loop pumping system (Source: Ref. 1)

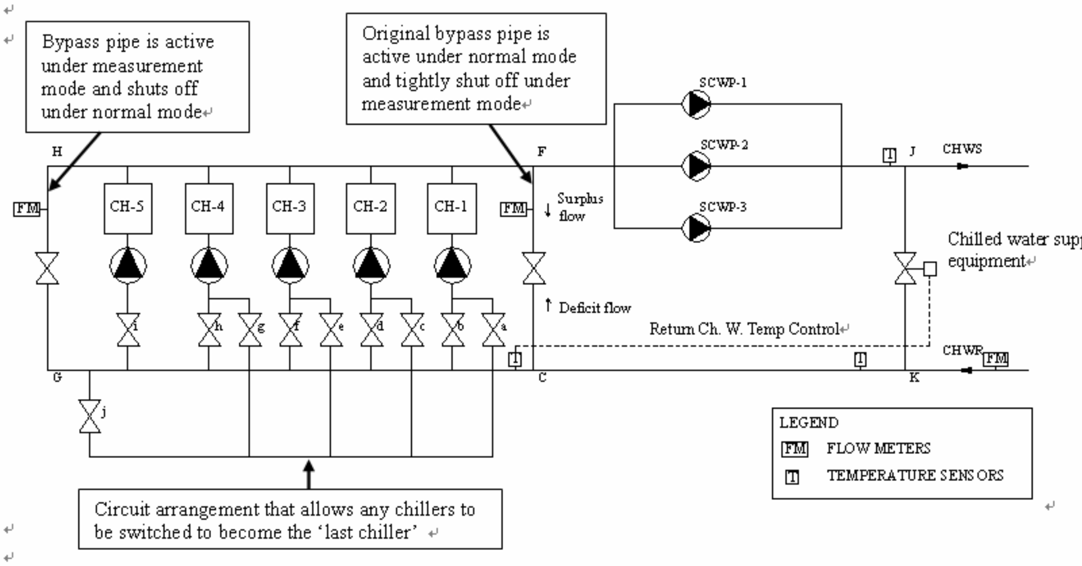


Figure 2 Schematic of the chilled water system

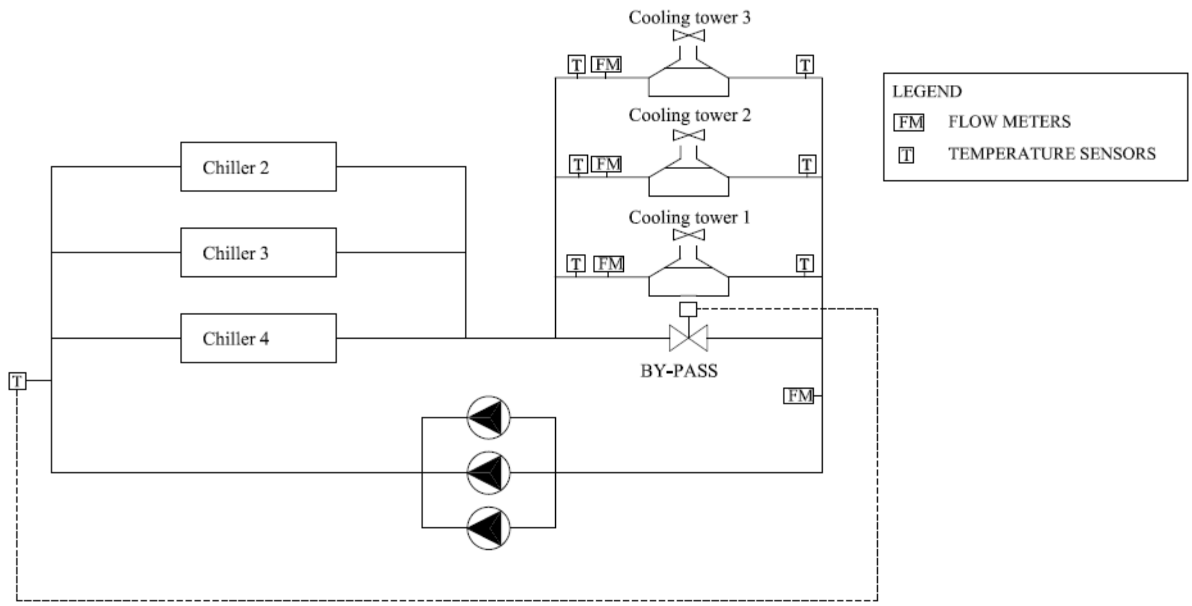


Figure 3 Schematic of the condenser water system

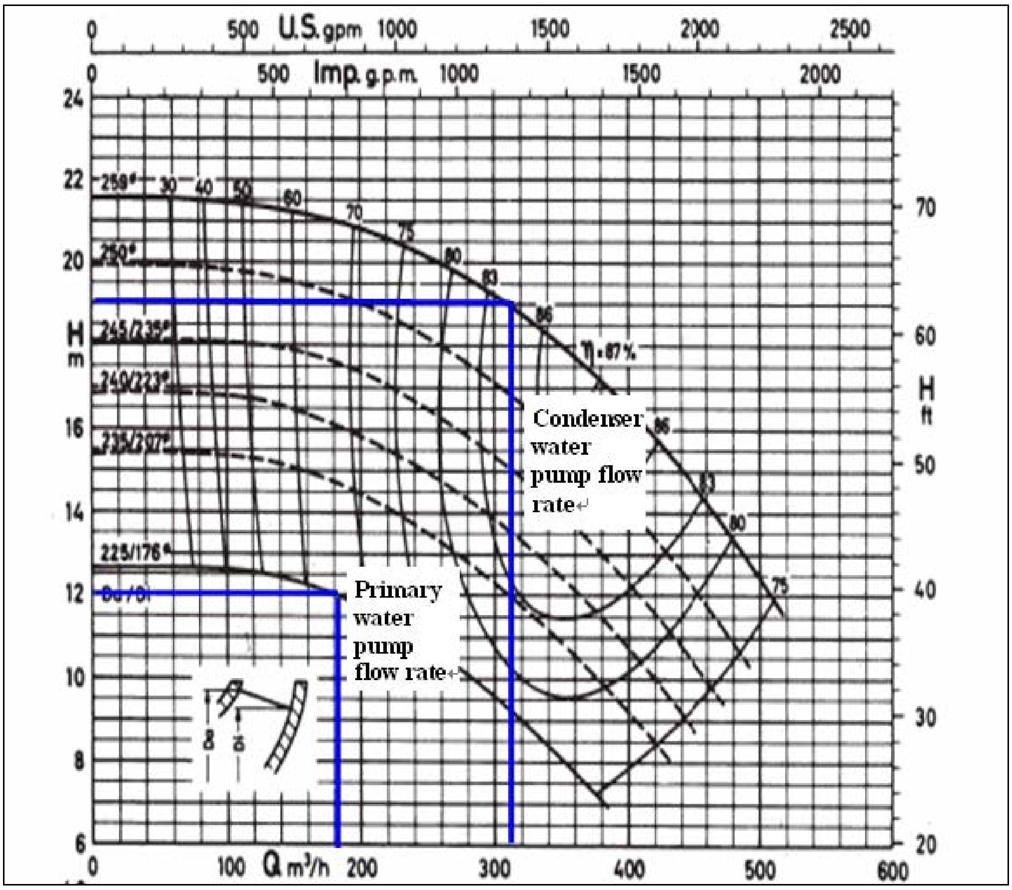


Figure 4 Chilled water flow rate estimation from the pressure measurement and pump curve

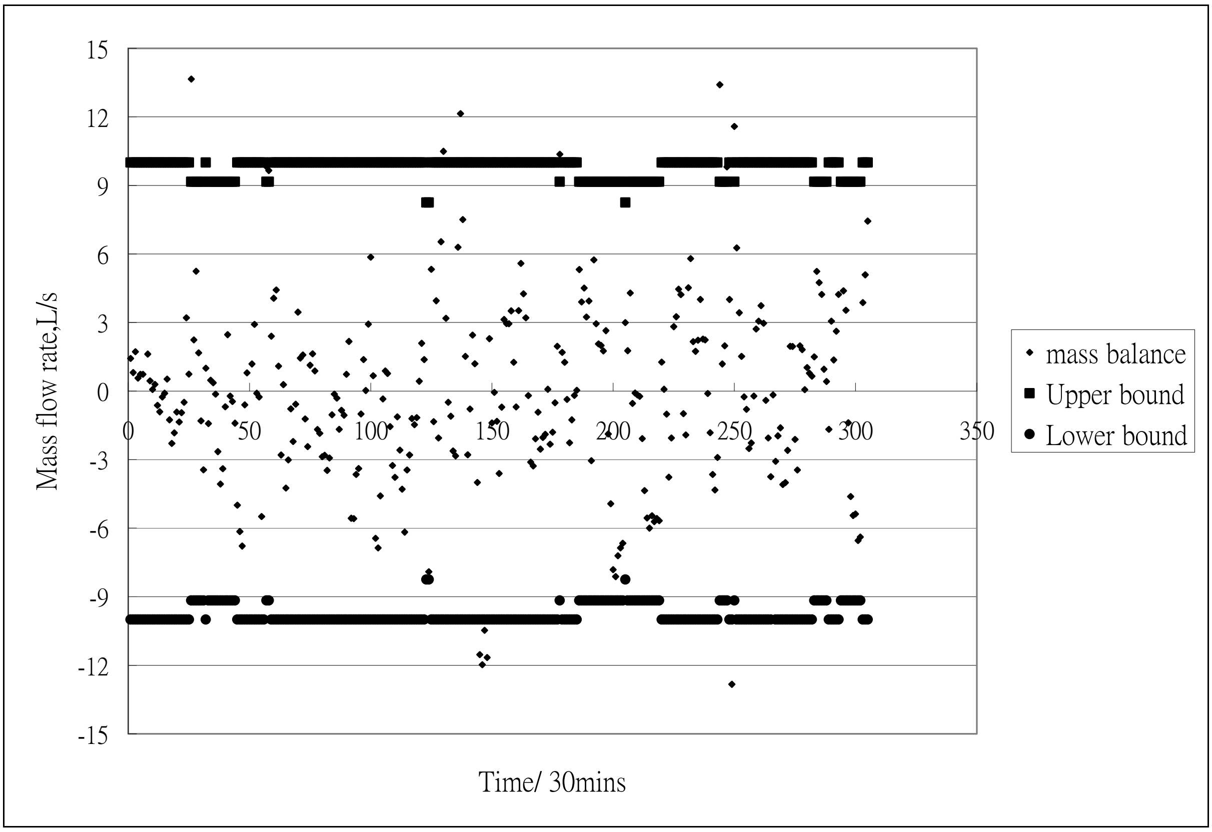


Figure 5 Residual mass flow rates

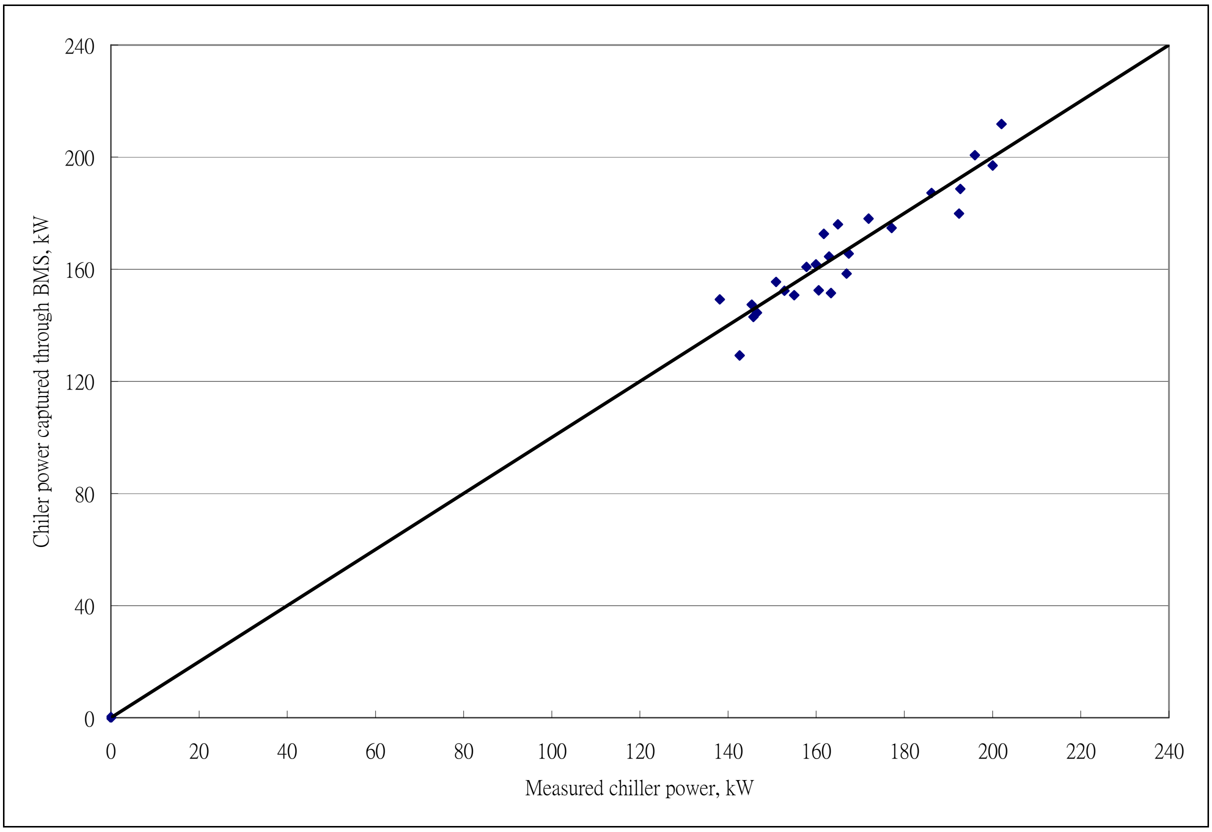


Figure 6 Power demand of Chiller 4 retrieved from BMS and measured by portable power analyzer

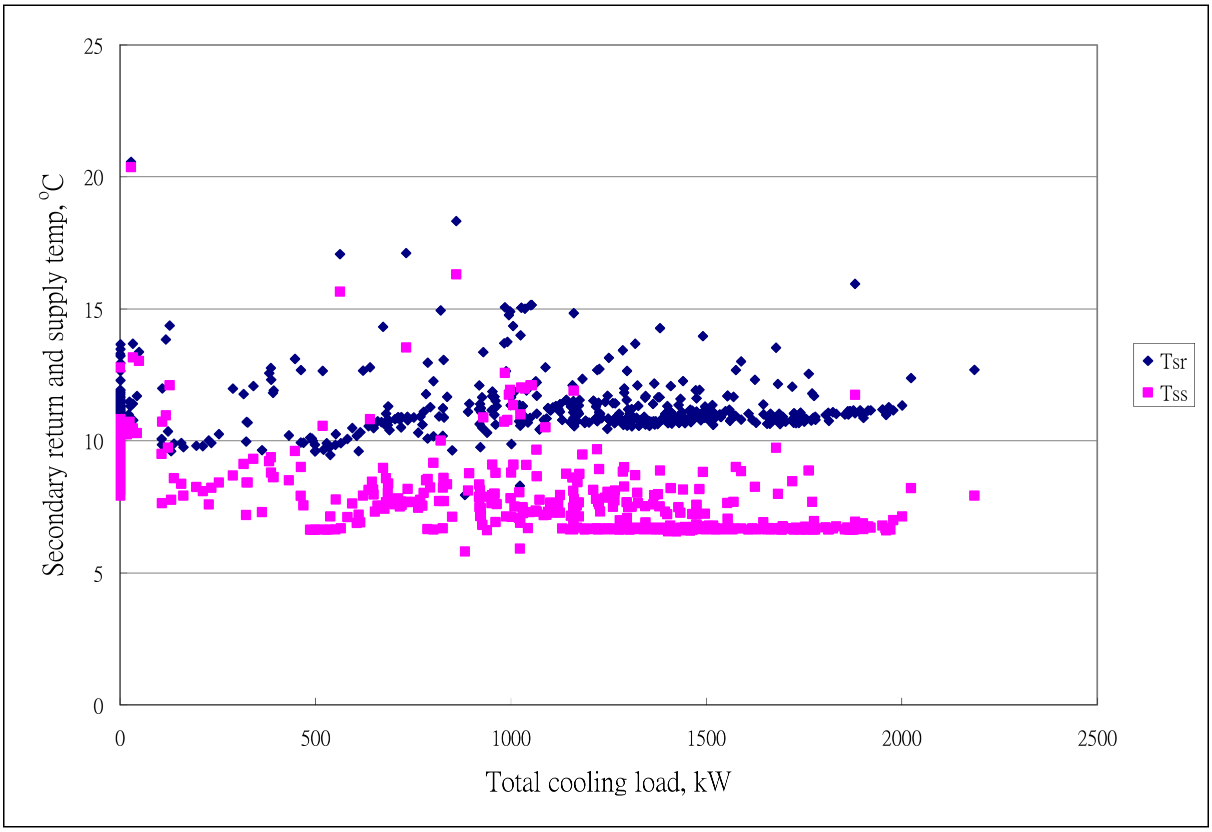


Figure 7 The main supply (T_{ss}) and return (T_{sr}) chilled water temperatures at the secondary-loop at different total cooling loads

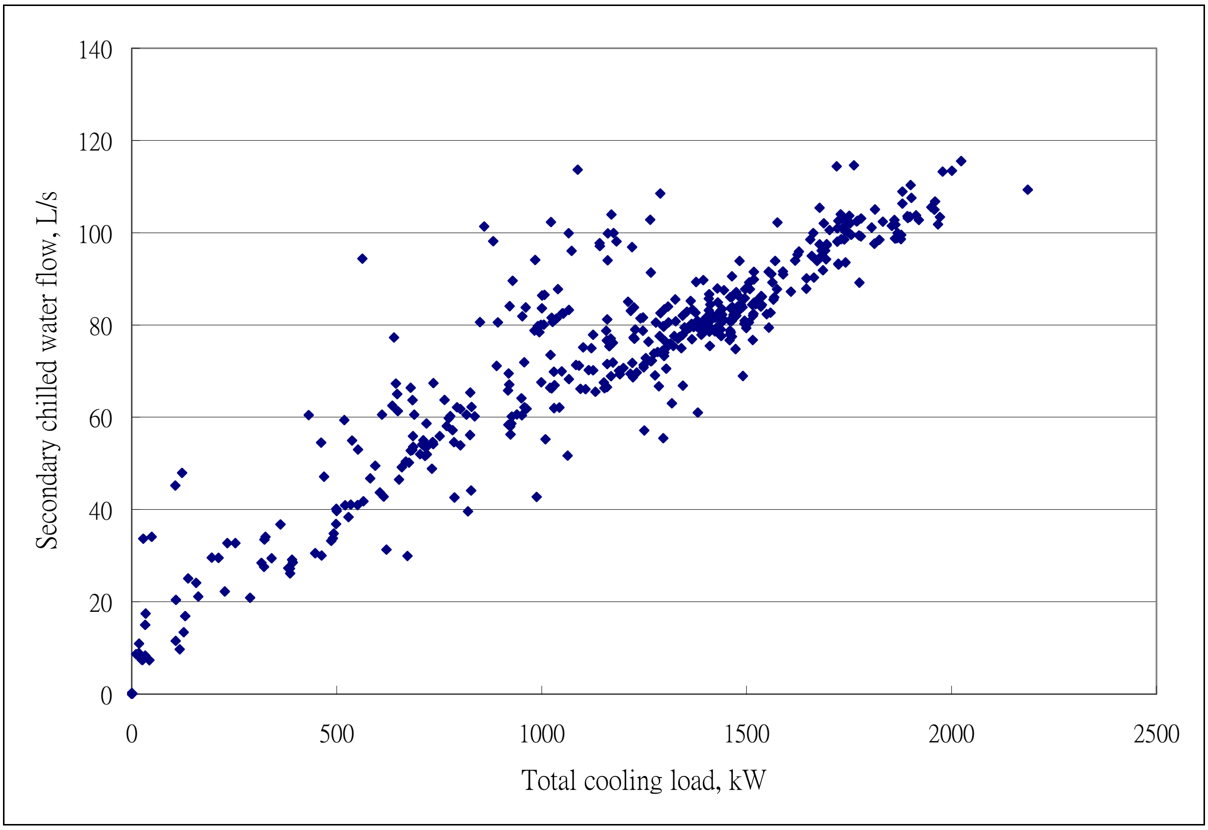


Figure 8 Relation between secondary-loop chilled water flow rate and total cooling load

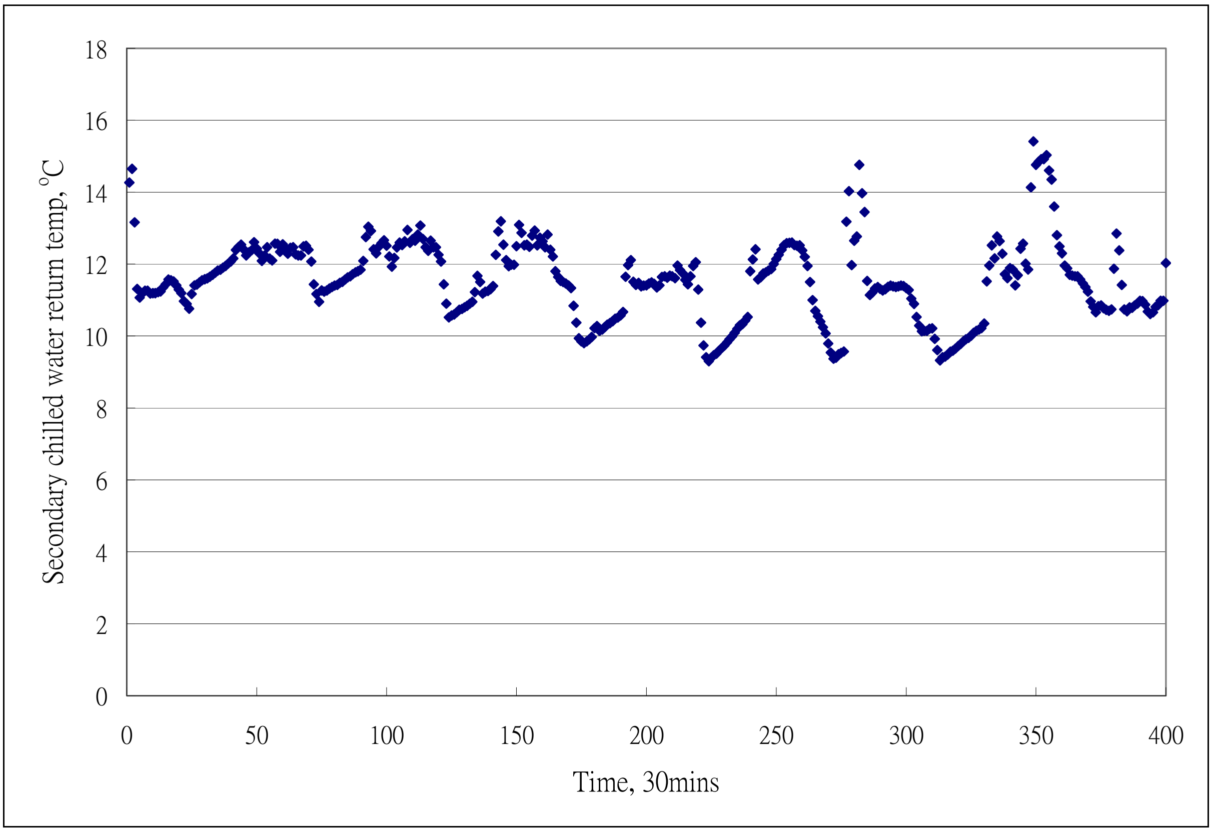
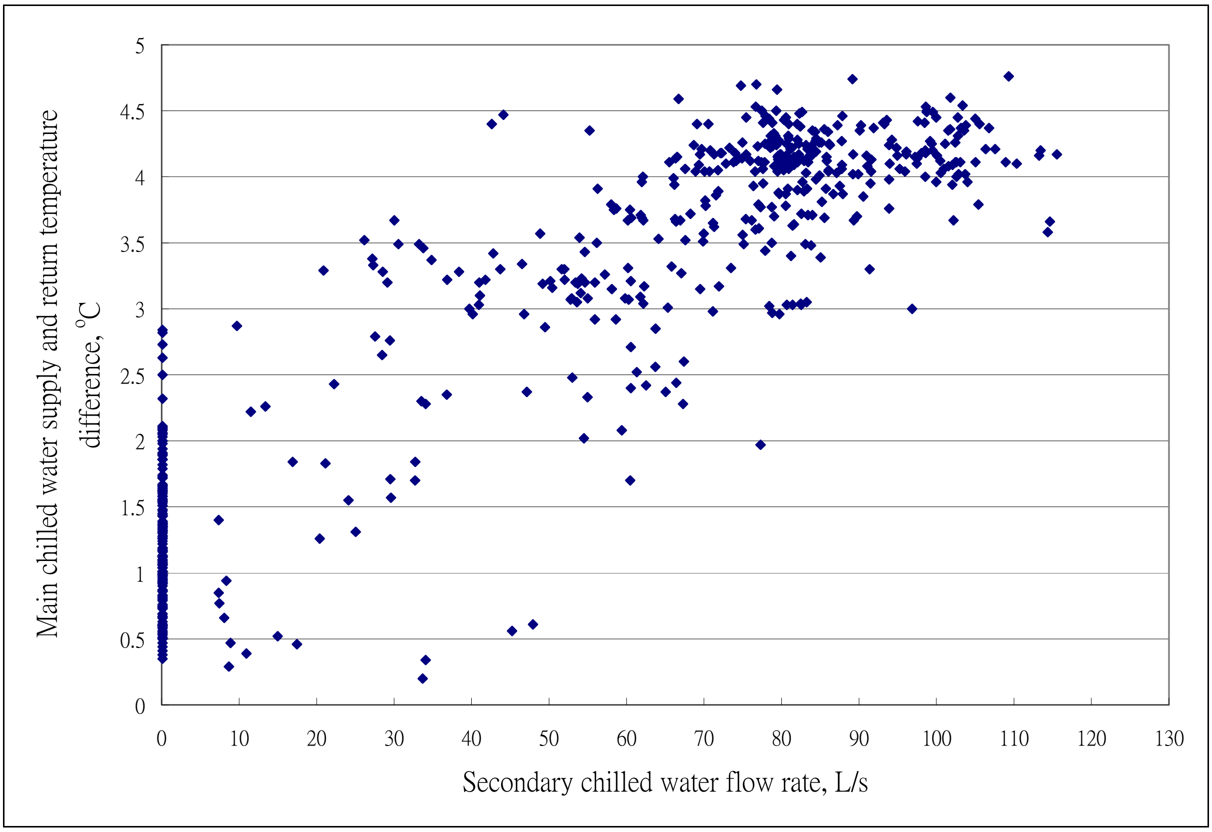


Figure 9 Main chilled water return temperature at the secondary-loop after reduction in differential pressure control set point



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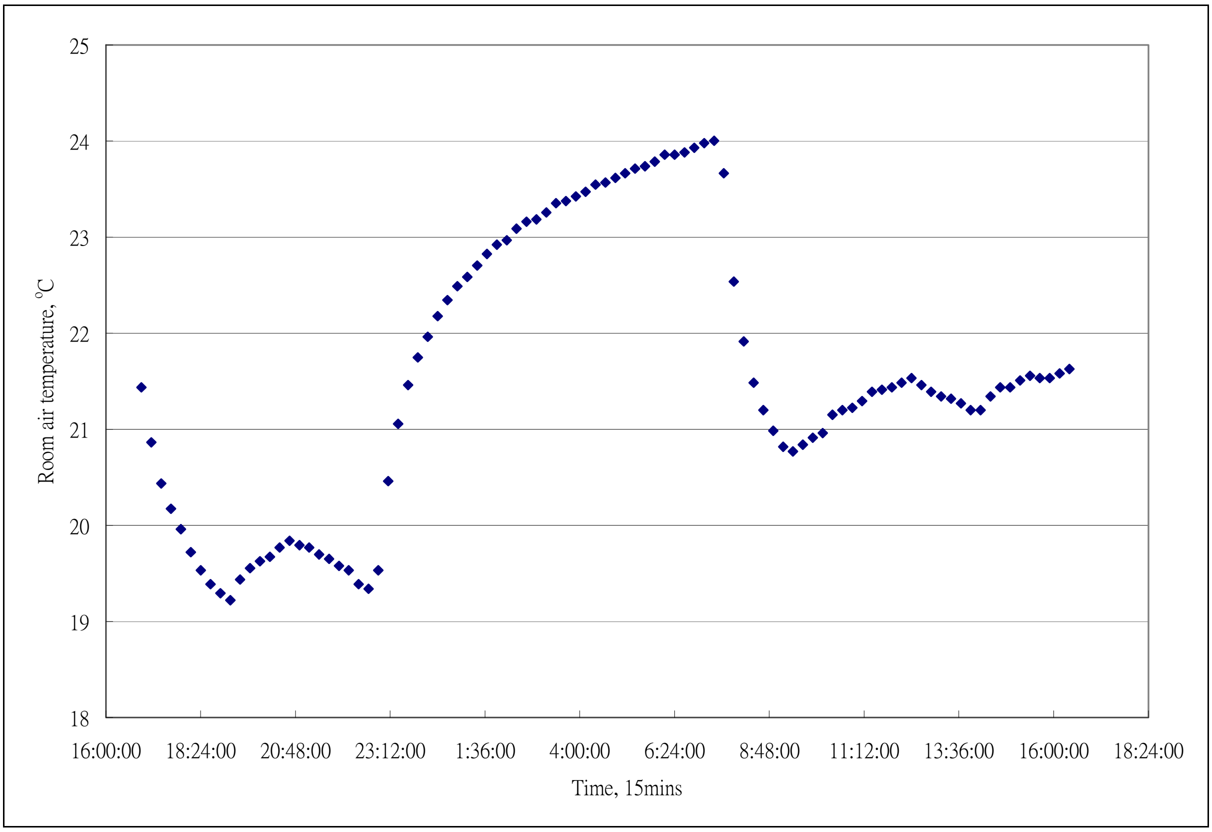

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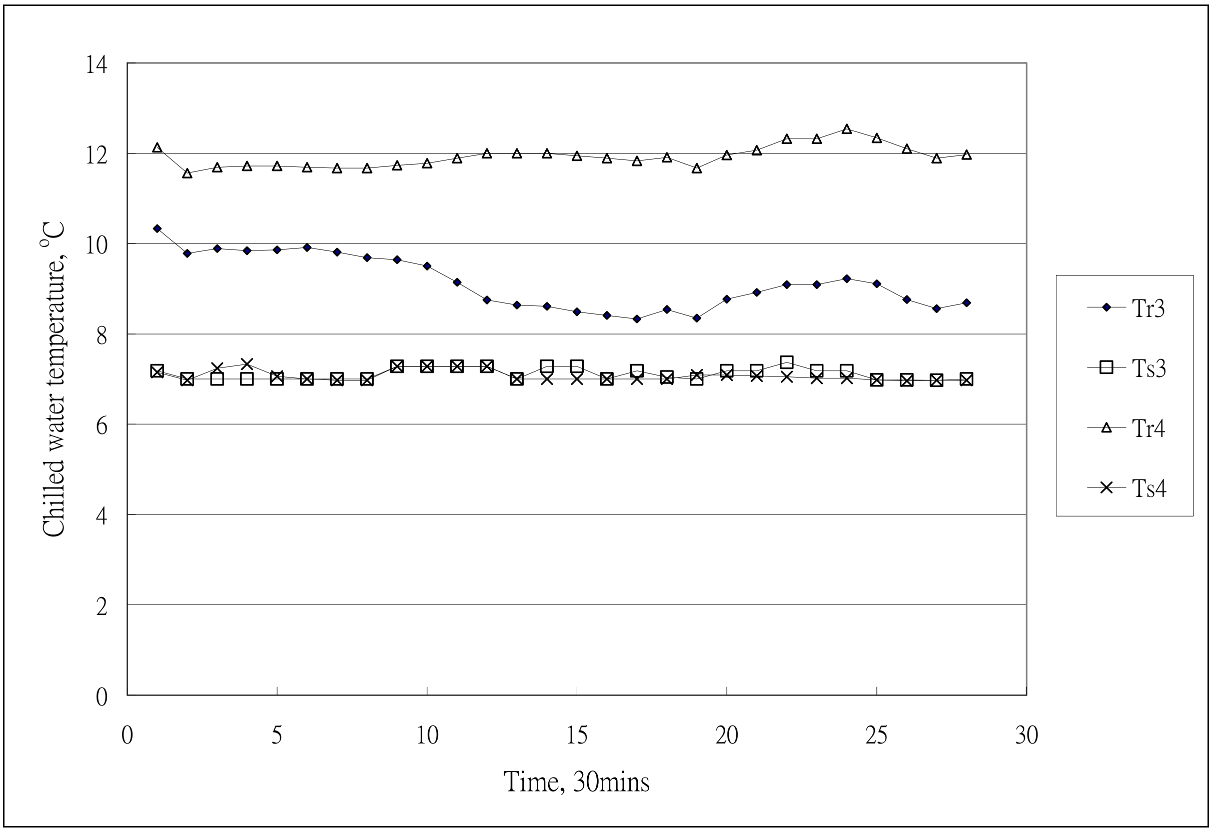

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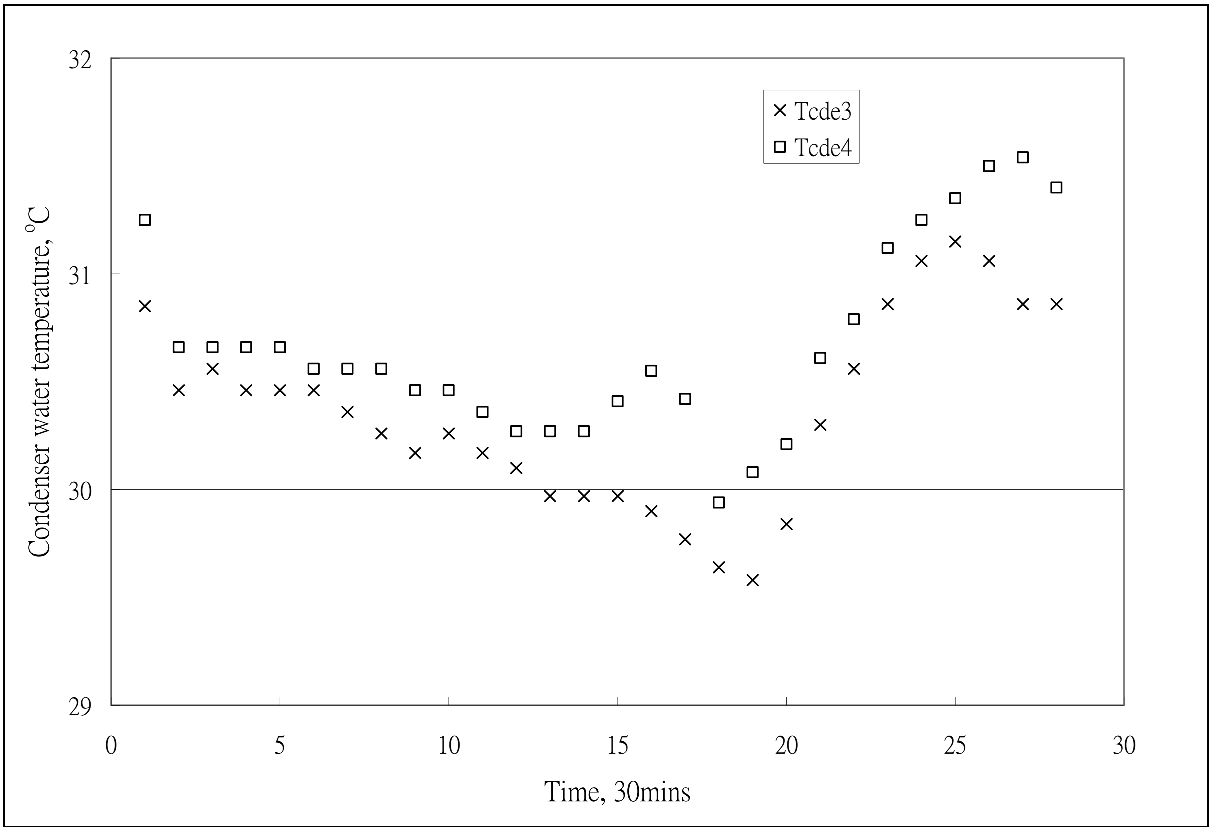
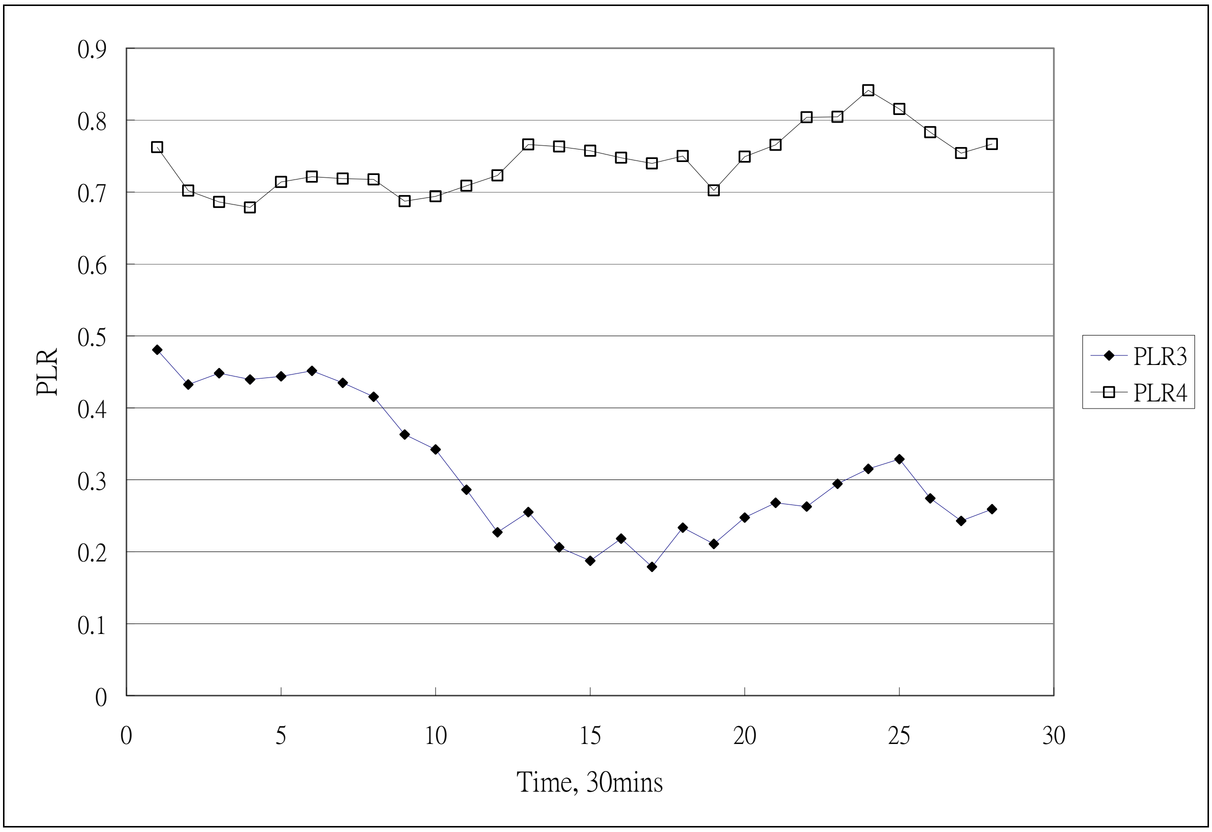


Figure 13 Condenser water temperatures entering Chillers 3 & 4 (T_{cde3}, T_{cde4}) in Stage 2 chiller performance test



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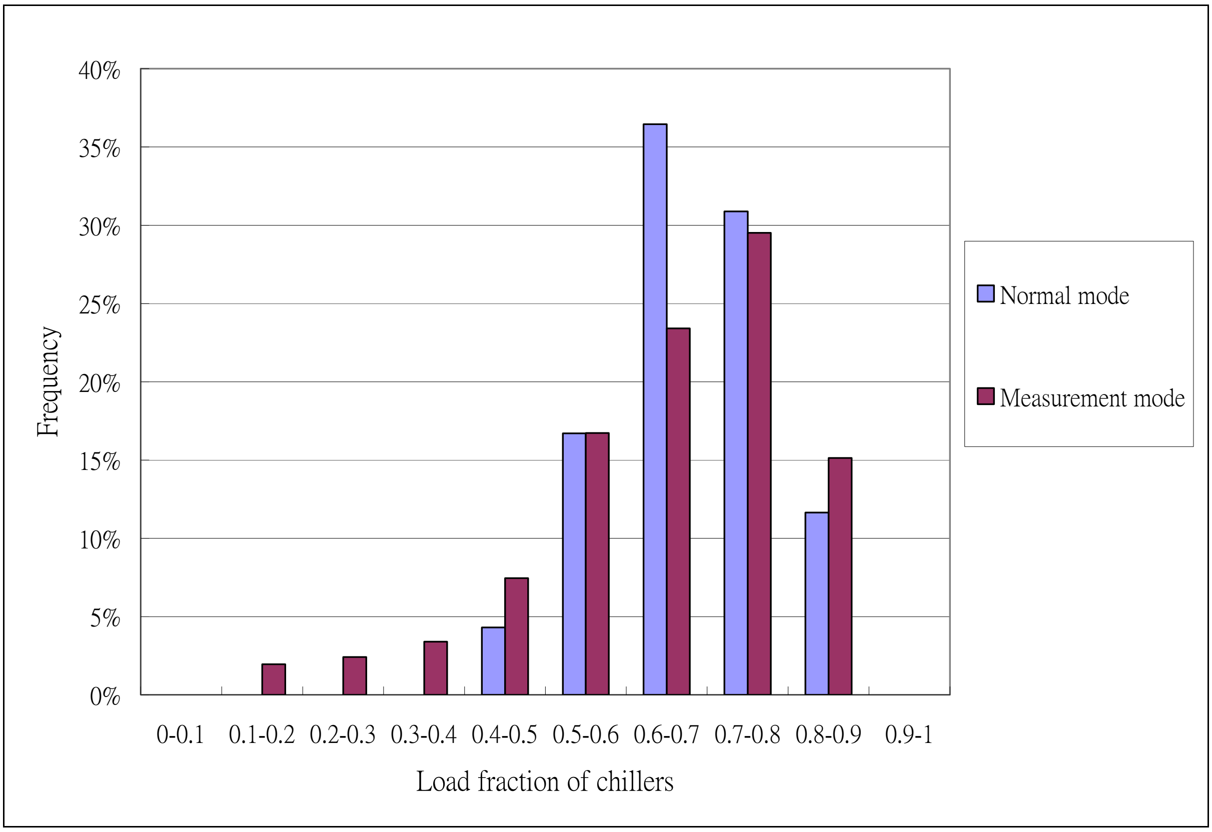

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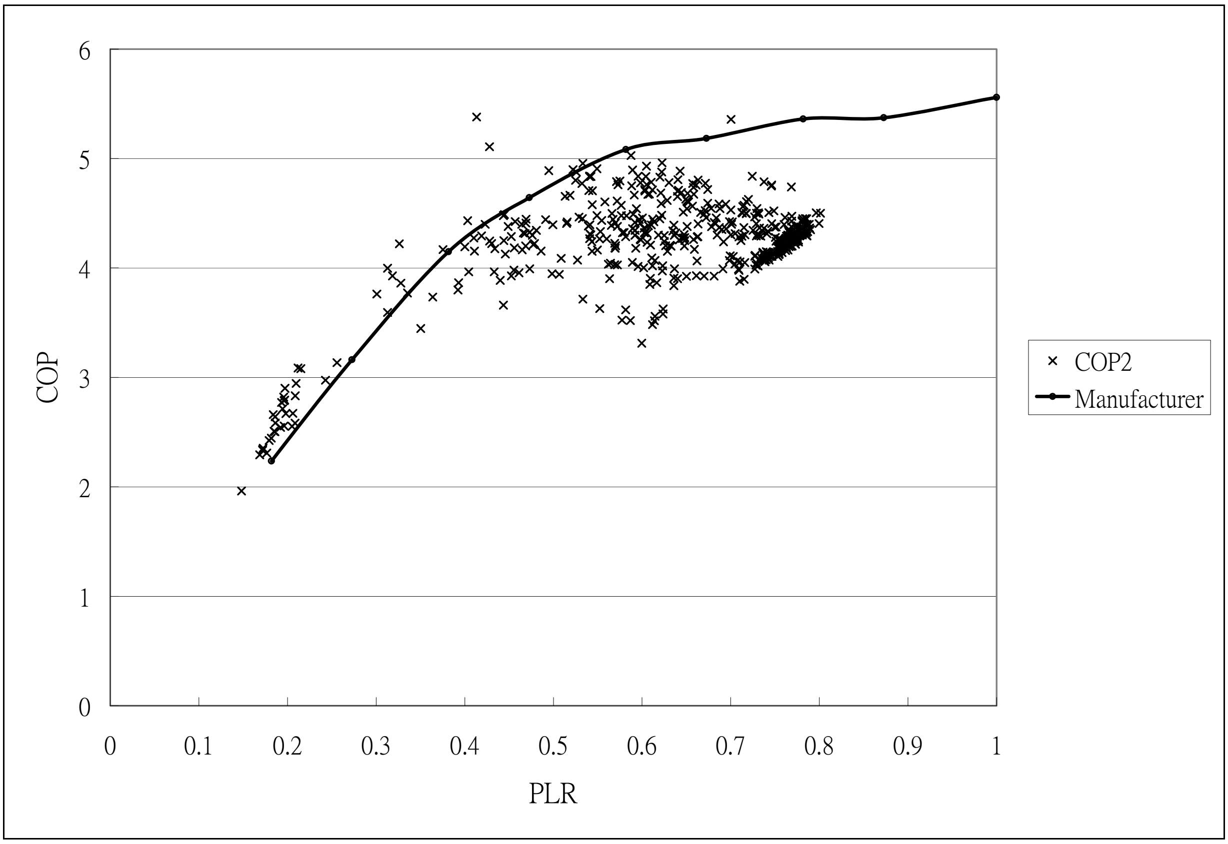
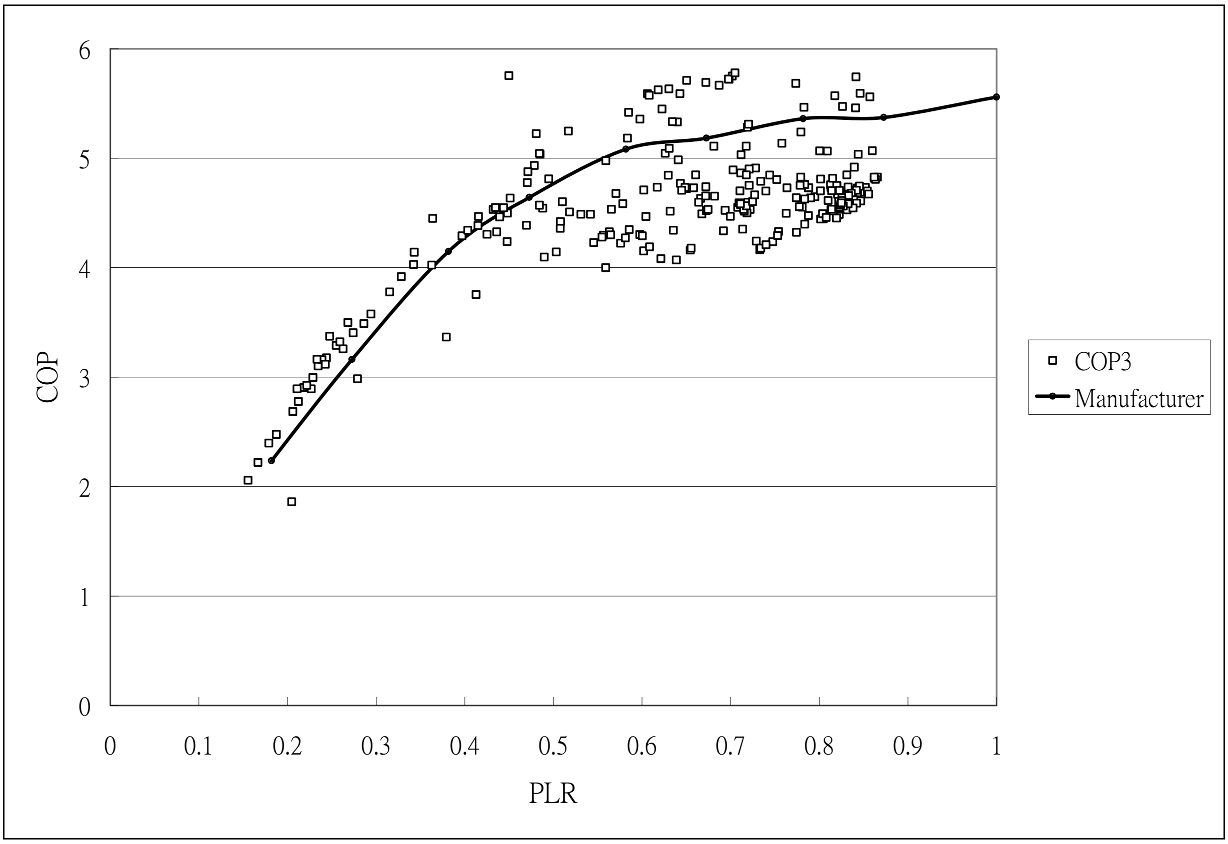


6
5
4
3
2
1
0
COP
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
PLR
COP2
Manufacturer

(a)



6
5
4
3
2
1
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COP
0
0.1
0.2
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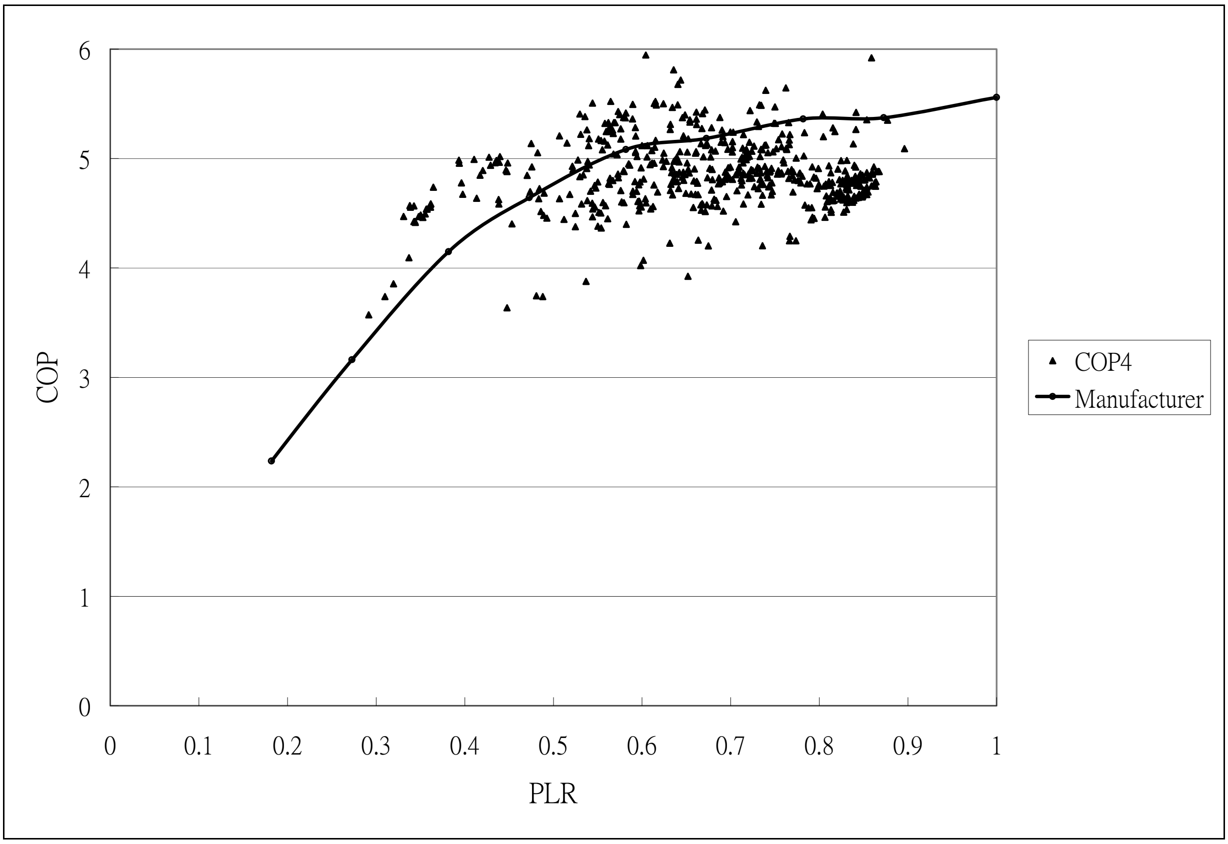
0.4
0.5
0.6
0.7
0.8
0.9
1
PLR
COP3
Manufacturer

(b)



(c)

Figure 16 Comparison of measured COP of chillers with the manufacturer data: (a) Chiller 2; (b) Chiller 3; (c) Chiller 4

Table 1 Characteristics of major equipments in the chiller plant

	New chillers		Existing chillers	
Type	Water-cooled screw		Air-cooled reciprocating	
Number of chillers	3		2	
Chiller numbers	CH-2 to CH-4		CH-1 & CH-5	
Rated cooling capacity (each), TR	400		265	
Rated chilled water inlet temp., C	12		12	
Rated chilled water outlet temp., C	7		7	
Rated chilled water flow rate (each), kg/s	67		44	
Rated water pressure drop across evaporator, kPa	110		60	
Rated condenser water / air inlet temp., C	32		35	
Rated condenser water outlet temp., C	37		-	
Rated condenser water flow rate (each), kg/s	83.8		-	
Rated water pressure drop across condenser, kPa	58		-	
Rated power demand, kW	254		316	
Rated coefficient of performance (COP)	5.5		2.9	
	New PCWPs	New CWPs	Existing PCWPs	Existing SCWPs
Number of pumps	3	4	2	3
Pump numbers	PCWP-2 to PCWP-4	CWP-1 to CWP-4	PCWP-1 & PCWP-5	SCWP-1 to SCWP-3
Chilled water flow rate (each), kg/s	67	83.8	44	125.6
Pumping pressure, kPa	150	230	150	375

Table 2 Measurement accuracies of sensors in the chiller plant

Instruments	System variables measured	Accuracy
Thermistors	Secondary-loop main chilled water supply and return temperatures Common chilled water return temperature to chillers Chilled water entering and leaving temperatures at individual water-cooled chillers (built-in sensor in chillers) Condenser water entering and leaving temperatures at individual water-cooled chillers (built-in sensor in chillers)	±0.3˚C
Insertion type flow meters	Chilled water flow rate in and bypassing the secondary-loop Chilled water and condenser water flow rates through individual water-cooled chillers (built-in sensors in chillers)	±4% of full scale
Watt meters	Power demands of chillers and pumps	±1% of full scale

Table 3 Measurements of system variables available from BMS records

System variables	Symbol
Chilled water supply temperature at chiller *i*	T_{si}
Chilled water return temperature at chiller *i*	T_{ri}
Condenser water temperature entering chiller *i*	T_{cdei}
Condenser water temperature leaving chiller *i*	T_{cdli}
Main secondary-loop chilled water supply temperature	T_{ss}
Main secondary-loop chilled water return temperature	T_{sr}
Common chilled water return temperature to chiller(s)	T_{rch}
Chilled water flow rate through chiller *i*	m_i
Bypass chilled water flow rate	m_{by}
Secondary-loop chilled water flow rate	m_{sec}
Power of chiller *i*	W_i
Refrigerant condensing temperature in chiller *i* (water-cooled)	T_{cdi}
Refrigerant evaporating temperature in chiller *i* (water-cooled)	T_{evpi}
Refrigerant discharge temperature in chiller *i* (water-cooled)	T_{disi}

Table 4 Comparison of temperature measurements retrieved from the BMS with snap-shot measurements made on-site

	Measurement with liquid-in-glass thermometer (°C)	BMS Reading (°C)	Absolute deviation (°C)
Secondary-loop chilled water return temp	11.7	11.4	0.3
Common chilled water return temp to chillers	11.2	11	0.2
Secondary chilled water supply temp	7	7.2	0.2
Chiller 2 supply temp	7.1	6.8	0.3
Chiller 2 return temp	11.2	10.9	0.3
Chiller 3 supply temp	7.2	7	0.2
Chiller 3 return temp	11	11.1	0.1
Chiller 4 supply temp	7	7	0
Chiller 4 return temp	12.2	12.2	0

Table 5 Statistical analysis on the deviations of individual chilled water temperature sensor readings from the reference readings

Temperature sensor	Chiller 2		Chiller 3		Chiller 4		Common chilled water return temperature (T_{rch}) (°C)
	Supply (T_{s2}) (°C)	Return (T_{r2}) (°C)	Supply (T_{s3}) (°C)	Return (T_{r3}) (°C)	Supply (T_{s3}) (°C)	Return (T_{r3}) (°C)	
Main bias error	0.18	0.19	0.10	0.11	0.09	0.20	-0.05
Root mean square error	0.34	0.24	0.27	0.21	0.22	0.28	0.22

Table 6 Statistical analysis on the individual chilled water flow rate of individual chiller

Flow meter	Chiller 2		Chiller 3		Chiller 4	
	Chilled water flow rate (l/s)	Condenser water flow rate (l/s)	Chilled water flow rate (l/s)	Condenser water flow rate (l/s)	Chilled water flow rate (l/s)	Condenser water flow rate (l/s)
Mean	49.13	88.27	53.27	89.26	51.04	87.47
Standard deviation	0.42	2.42	0.52	3.63	0.61	2.35

Table 7 Chilled and condenser water flow rates estimated from pressure drops across chillers

Recorded data	Unit	Chiller 2	Chiller 3	Chiller 4
Cooler inlet pressure	kPa	117	117.4	119.3
Cooler outlet pressure	kPa	48.9	46.1	47.8
Measured chilled water flow	l/s	49	53	51
Evaporator in/out pressure difference	kPa	68.1	71.3	71.5
Estimated evaporator flow rate	l/s	52.7	53.9	54
Deviation between measured and estimated flow rate	l/s (%)	3.7 (7)	0.9 (1.7)	3 (5.6)
Condenser inlet pressure	kPa	133	132.3	135
Condenser outlet pressure	kPa	76	75.2	77
Measured condenser water flow	l/s	88	89	87
Condenser in/out pressure difference	kPa	57.0	57.1	58.0
Estimated condenser water flow rate	l/s	83.1	83.1	83.8
Deviation between measured and estimated flow rate	l/s (%)	4.9 (5.9)	5.9 (7)	3.2 (3.8)

Table 8 Chilled water flow rates estimated from primary chilled water pump pressure

Recorded data	Unit	PCWP 2	PCWP 3	PCWP 4
Suction pressure	kPa	7.2	7.3	7.3
Discharge pressure	kPa	126.3	123.4	124.4
Pressure difference	kPa	119.1	116.1	117.1
	mH_2O	12.1	11.8	11.9
Chilled water flow rate	l/s	50	52.3	51.4
(from pump curve)	m^3/h	180	190	185
Chilled water flow rate	l/s	49	53	51
(measured)	m^3/h	176.4	190.8	183.6

Table 9 Condenser water flow rates estimated from condenser water pump pressure

Recorded data	Unit	CWP 1	CWP 2	CWP 3
Suction pressure	kPa	1	1	2
Discharge pressure	kPa	188.4	189.3	190.9
Pressure difference	kPa	187.4	188.3	188.9
	mH_2O	19.1	19.2	19.3
Condenser water flow rate	l/s	86.1	84.7	83.3
(from pump curve)	m^3/h	310	305	300
Condenser water flow rate	l/s	88	88	88
(measured)	m^3/h	316.8	316.8	316.8

Table 10 Open / closed status of the isolation valves for Stages 1 to 3 tests

		Valve for Chiller #	1		2		3		4		5	All
			Valve Code									
Stage	Last Chiller	Preferred Chiller	a	b	c	d	e	f	g	h	i	j
1	Chiller 4	Chiller 2	C	C	C	O	C	O	C	O	C	C
2	Chiller 3	Chiller 4	C	C	C	O	**O**	**C**	C	O	C	**O**
3	Chiller 2	Chiller 3	C	C	**O**	**C**	C	O	C	O	C	**O**

Note: O denotes open; C denotes closed.

Yik, F.W.H., Lee, S.H., Lai, J.H.K. and Chan, K.T. (2010), Experience of using a chilled water circuit design to expedite in situ chiller performance measurement, Building Services Engineering Research and Technology, Vol. 31, No. 3, pp. 279-300

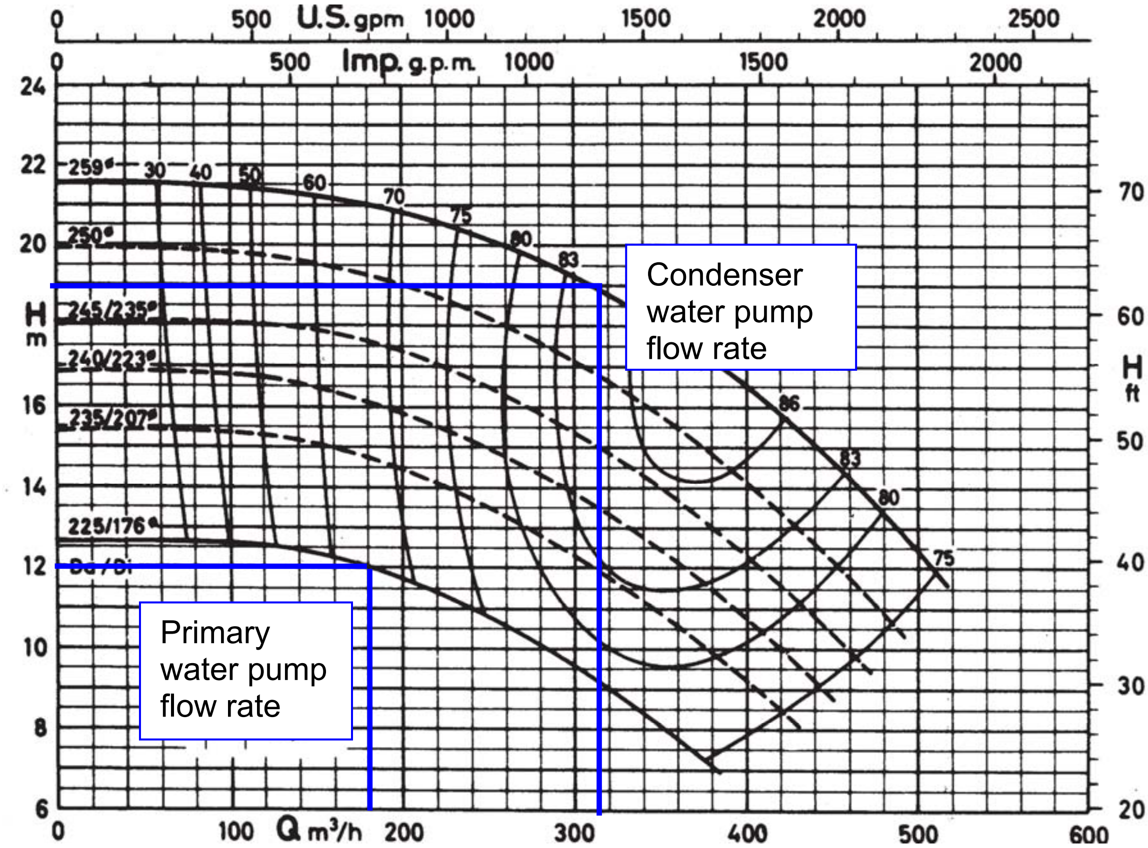


Figure 4 Chilled water flow rate estimation from the pressure measurement and pump curve

Table 10 Open / closed status of the isolation valves for Stages 1 to 3 tests

		Valve for Chiller #	1		2		3		4		5	All
			Valve Code									
Stage	Last Chiller	Preferred Chiller	a	b	c	d	e	f	g	h	i	j
1	Chiller 4	Chiller 2	C	C	C	O	C	O	C	O	C	C
2	Chiller 3	Chiller 4	C	C	C	O	O	C	C	O	C	O
3	Chiller 2	Chiller 3	C	C	O	C	C	O	C	O	C	O

Note: O denotes open; C denotes closed.