Development and Verification for the Control Method Using Surplus Pressure of Primary Pumps in Chiller Plant Systems for Air Conditioning which Adopts Primary/Secondary Piping Systems

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Abstract:
The primary/secondary piping systems are often employed in large chiller plant Systems. Normally, the primary flow becomes more than secondary flow, and the flow difference returns to a chiller via decoupler, which is common to primary flow loop (chiller side) and secondary flow loop (load side). It is a huge energy loss, because the primary pumps use their head to lead much flow to the decoupler. Therefore, we have developed new control method using surplus pressure of the primary pump to reduce the primary and secondary pumps' energy. In this paper, we used this control method to the actual chiller plant buildings and verified its effectiveness. As a result, cold water conveyances, both primary loop and secondary loop, could be covered by only primary pumps during plant operating time, and the water conveyance power energy was reduced approximately 80%.

Keywords:
Primary/Secondary piping system, Water conveyance power, Control method for energy conservation

Introduction:

The primary/secondary piping systems shown in Figure-1 are often employed in large chiller plant Systems which have more than two chillers. Primary/secondary systems have two flow loops, the primary loop with chillers and the secondary loop with the loads, and each loop has primary pumps and secondary pumps for the cold water conveyance. The pipe, "Decoupler", is common to these two flow loops and is connected between chillers and loads. The load, for example, air handling units in the secondary loop, has a two-way control valve, and so the secondary flow is changing by the load. On the other hand, the primary flow through the chillers is normally constant for maintaining chiller stability. The decoupler allows the pumps to operate at different flow rate. Normally, the primary flow becomes more than secondary flow, and the flow difference returns to a chiller via decoupler. It is a huge energy loss, if the big difference continues between primary flow and secondary flow, because the primary pumps use their heads to lead much flow to the decoupler. Therefore, we developed the control method using surplus pressure of the primary pump to reduce the primary and secondary pumps' energy. This control method ensures the smallest flow to maintain the chiller operation and saves the water conveyance power by utilizing the surplus pressure of the primary pump to the secondary, though the flow has returned to a chiller via decoupler until now. The days which are required for a full capacity must be few in the buildings for business use, so the actual loads are in a low condition for the chiller capacity in most of a year. The load is a partial load for the operating chiller capability when the secondary flow becomes lower than primary flow, so the occurring frequency is very high. In this paper, we explain the “Control Method Using Surplus Pressure of Primary Pumps” and describe the substantiated effects of energy reduction in the actual plant systems.

1. Summary and Problems of Conventional Cool water conveyance Control Method

The general control as a secondary conveyance control of cool water in primary/secondary piping systems is a secondary pump inverter control to keep the secondary supply water header shown as "SH-2" in Fig.1, and the operating pump number control depending on the load flow rate. It is general that the set-point of secondary supply header pressure for the inverter control is set to the total head of its rated flow by the secondary pump. For example, in case of the rated operating point of secondary pump, 5m^3/min(flow rate) and 300kPa (total head), shown in Fig.2, 300kPa (about 30m-aqua) total heads are the target value for the inverter control. If the load flow is 1m^3/min and is controlled to 300kPa target value, the frequency decreases to only approximately 48Hz because of the pump operating frequency characteristic figure shown in Fig.3. In this case, the 300kPa total head is not required, and the pump system applies excessive pressure to the secondary cool water loop. In fact, energy saving effects are not enough because the set-point becomes high condition in most time of a year by operating the designed value of maximum load flow, though the secondary pump is controlled by an inverter.

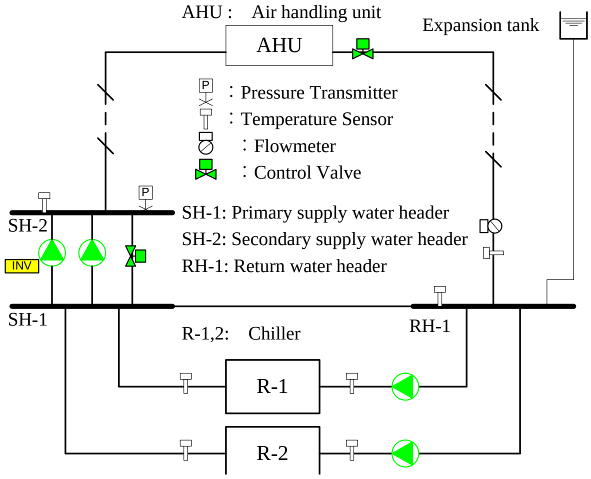

Fig.1 General Primary/Secondary Piping System

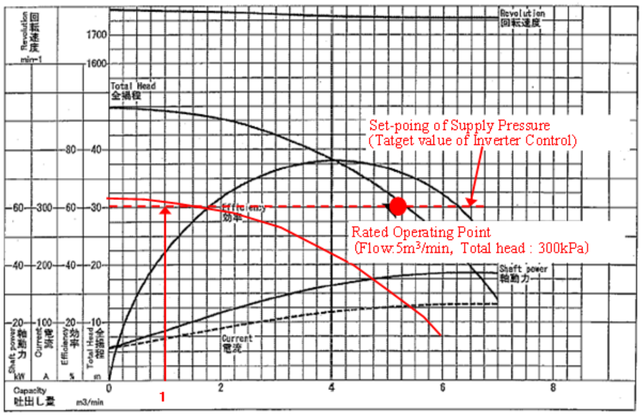

Fig. 2 Performance Curve of a Secondary Pump

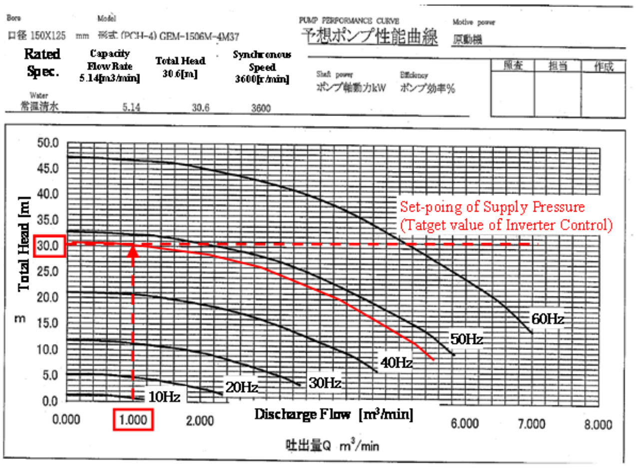

Fig.3 Frequency Characteristic of Secondary Pump

2. The Control Method Using Surplus Pressure of Primary Pumps in Chiller Plant Systems

2.1 Control Method Outline

The control method that we developed enables practical use of primary pump surplus pressure at the secondary cool water conveyance loop in order to reduce energy consumption of conveyance electric power. Instrumentation devices required for this control are only three devices shown in Fig.4, a pressure transmitter for measuring secondary supply header (Ps2) and primary supply header

(Ps1), a pressure transmitter for measuring the end pressure at the end point of the chiller plant (Pe), and a two-way control valve installed to the decoupler (V2, “decoupler valve” in this paper). The control module component group to enable the control method using surplus pressure of primary pump is Table 1, and Fig.5 and Fig.6 are the module component group control flow charts. Fig.5 shows modules controlled by end pressure (Pe) and Fig.6 shows modules controlled by primary supply header pressure (Ps1). Under this control method, Fig.7 is a diagram showing correlation between the points of pipeline in a chiller plant system and the pressure value of each point in the pipeline. A chiller plant is located in the lower floor such as the basement in this diagram, and it shows the end load, air handling units installed in the most distant point, is located on the floors above the chiller plant in this case. The decoupler valve control adjusts the primary supply header pressure and increases a suction pressure on the secondary pump as much as possible (sown ih n (1), Fig.7). Then it adjusts the end pressure to the set-point by the inverter control of secondary pump (shown in (2), Fig.7). As a result, it enables to lower the inverter output and conveyance power energy consumption is reduced.

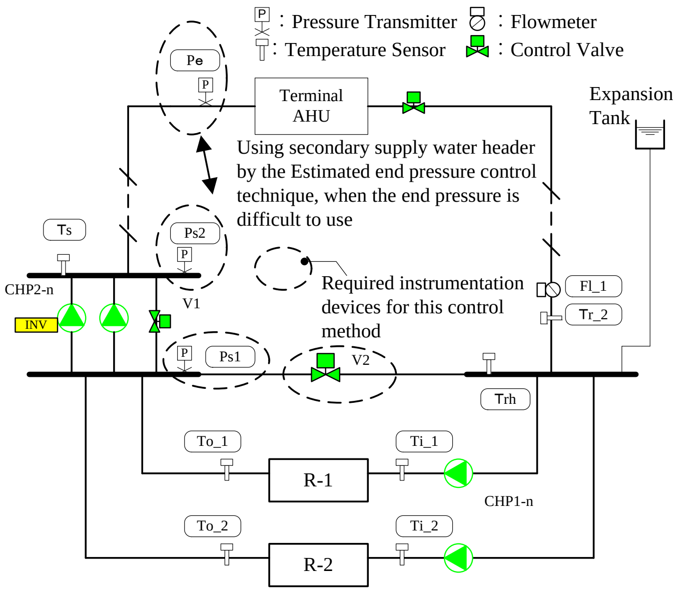

Fig. 4 Instrumentation Devices Required to the Control Method Using Surplus Pressure of Primary Pumps

Table 1: Control Point of Each Module

No.	*Control Module*	*Control Point*
1	*Inverter Control of Secondary Pump*	*End Pressure (Pe)*
2	*Number Control of Secondary Pump*	*End Pressure (Pe)*
3	*Pressure Relief Valve Control*	*End Pressure (Pe)*
4	*Decoupler Valve Control*	*Primary Supply Water Header Pressurure (Ps1)*

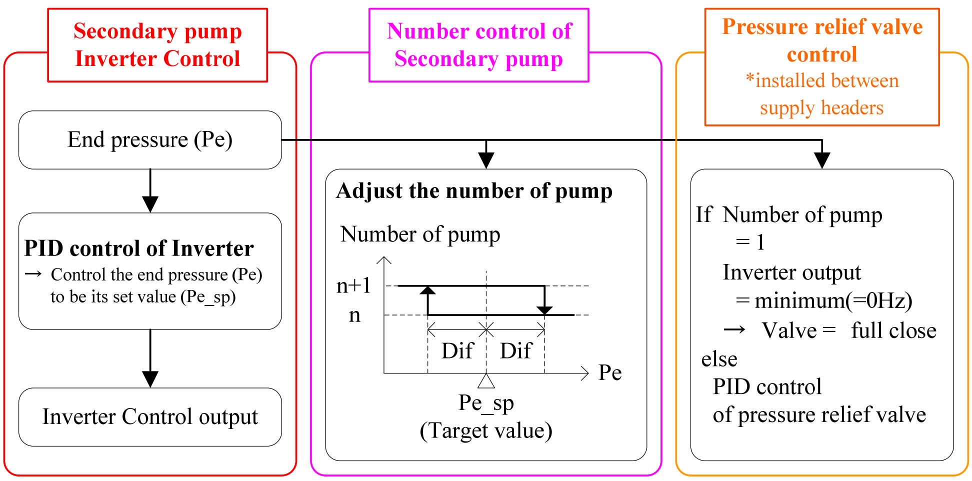

Fig. 5 Module Group Controlling by End Pressure (Pe)

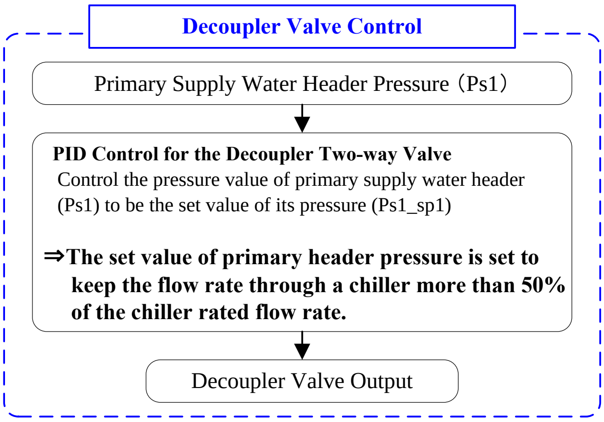

Fig. 6. Module Controlling by Primary Supply Header Pressure (Ps1)

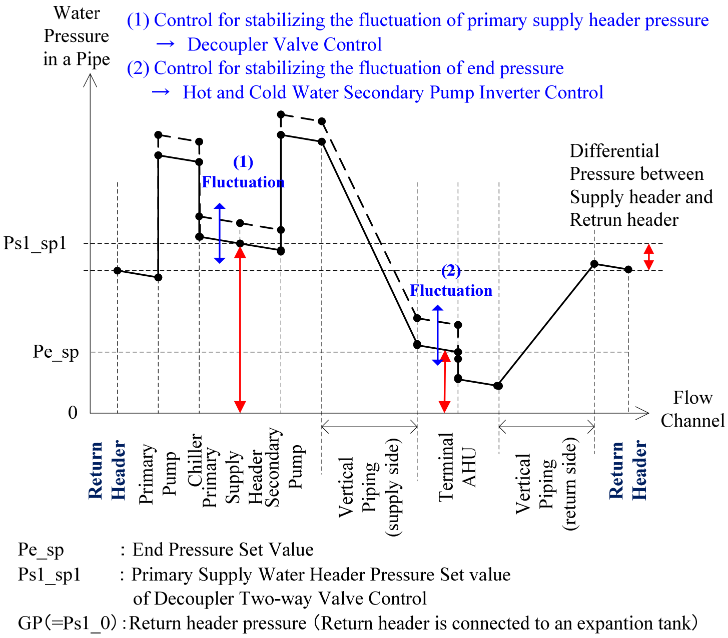

Fig. 7 Diagram about Each Pressure Value at Points of Pipeline in a Chiller Plant System

2.2 Each Control Module Details for the Control Method Using Surplus Pressure of Primary Pumps

(1) Decoupler Valve Control Module

This control module has two-way control valve and controls it for the primary supply header pressure (Ps1) to reach its target value (Ps1_sp1) by PID control. Generally, a role of primary pumps in the primary/secondary piping system is to keep a required flow rate for a chiller against the pressure loss of primary water flow loop. If the primary pump flow rate is more than the secondary one, in other words, the secondary loop flow is less than the rated flow rate of a chiller, the surplus flow returns to the chiller via a decoupler. (See Fig.8) In this situation, this control reduces the primary pump flow rate and raises its total head, if the decoupler valve is installed and appropriately adjusted. (See Fig.9) Accordingly, it enables to reduce the work of secondary pump owing to this surplus pressure. On the other hand, the chiller stops abnormally when the primary flow rate decreases too much because it is impossible to keep the minimum flow rate for the chiller operation. Therefore careful decision is required for the primary supply header pressure set-point under the decoupler valve control. It is required that the header pressure set-point of primary supply ensures a chiller minimum flow rate, even if all two-way valves at the secondary loads completely shut and the secondary side loop flow rate indicates “0”.

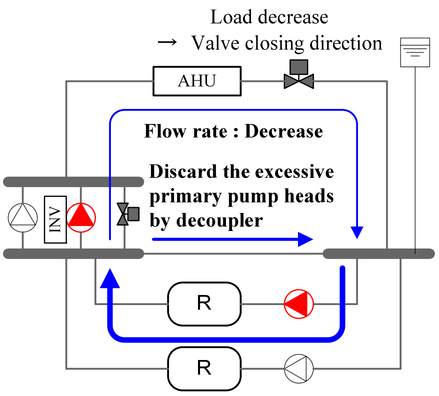

Fig. 8 Flow Situation in a Low Load Flow

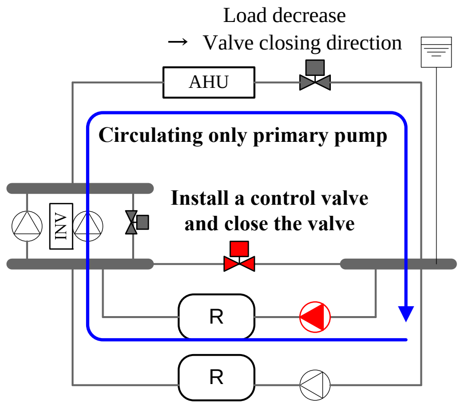

Fig. 9 Secondary Side Conveyance by the Surplus Pressure of a Primary Pump

(2) Inverter Control of Secondary Pump

Just as described in Section 1, the conventional inverter of secondary pump, the secondary supply header pressure, is controlled by supply pressure, and the set-point is set as a supply pressure value which is required for its designed maximum flow rate in most cases. As a result, an inverter frequency of secondary pump only decreases to around 35 – 45 Hz in many cases, since high set-point is kept even when the load flow is low. Therefore, the inverter is controlled by using end pressure of an inlet pipe point at the most distant air handling unit, instead of the supply pressure, since the most distant air handling unit is maximum pressure loss on the route in cool water conveyance from the chiller plant systems. If the differential pressure between inlet and outlet at the most distant air handling unit can be ensured enough flow rate, all other air handling units can be also ensured their required pressure. Consequently, this control can supply to the secondary loop flow its appropriate pressure depending on the load flow change. The conveyance power of the secondary pump can be significantly reduced by using the primary pump surplus pressure for secondary side water conveyance and reducing inverter output of secondary pump largely, if the decoupler valve control module described in (1) is used together with this inverter control of secondary pump after changing it as seen above.

(3) Number Control of the Secondary Pump

Existing control of the secondary pump controls the number of operation units by its secondary flow rate (load flow rate). For example, when the secondary flow rate becomes more than the rated flow rate of one secondary pump which is usually set-point of increase step from one to two, the number of secondary pump increases from one to two. Since suction pressure of secondary pump is changed by the decoupler valve control in the new control method which we developed, the flow rate which one secondary pump can flow changes. It means the flow rate is more than its rated flow rate of a secondary pump when the suction pressure is increased by the surplus pressure of primary pump. Therefore the number control of secondary pumps is decided by the end pressure instead of the secondary flow rate. The timing chart of control performance by this method appears in Fig.10. The control method is illustrated by using this figure as following. Normally, the end pressure (Pe) decreases if the flow rate on the secondary side increases in conjunction with the load. As the load is going to increase further, the end pressure cannot keep the target value（Pe_sp）and decreases even if an inverter output is maximum value. Secondary pumps are increased by this control method when this end pressure decreasing that we described above becomes lower than the increasing threshold value（Pe_sp－Dif_sp1）. On the contrary, secondary pumps are decreased when the end pressure becomes higher than the threshold decreasing value（Pe_sp＋Dif_sp2）.

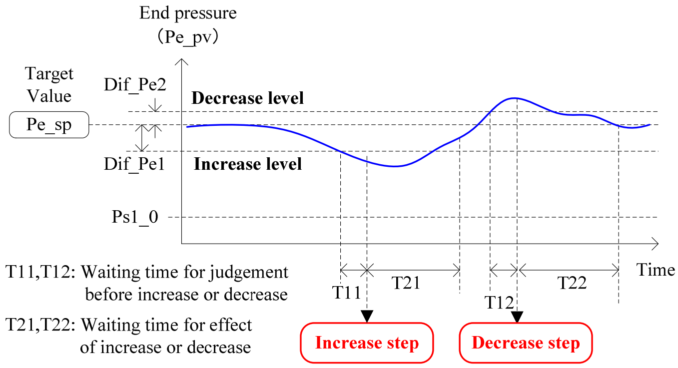

Fig.10 Time Chart of Secondary Pump Number Control

3. Installation to Verify

3.1 Installation Site

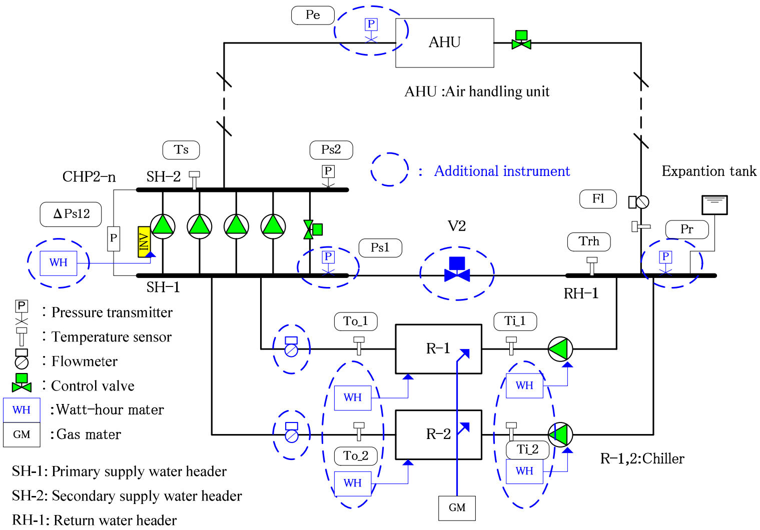

Fig. 11 Instrumentation Diagram of Chiller Plant System

We installed a system with "the control method using surplus pressure" in an actual site to have an effect verification. The site is “Ritsumeikan Uji Junior and Senior High School” located in Uji-shi, Kyoto, Japan and it has four floors above ground with total floor space of 19,172 square meters. A primary/secondary piping system provides as its air-conditioning system and it has two units of “gas fired double effect absorption chiller heaters” with 1,582 kW cooling capacity. (“Gas fired double effect absorption chiller heaters”, i.e. “gas absorption chiller heater”) Four units of hot-cold water secondary pump are in the system and one of them is an inverter. The inverter unit makes variable flow operation. Loads are air handling unit (AHU), outdoor air handling unit (OHU), and fan coil unit (FCU). Fig.11 is an instrumentation diagram of chiller plant system and main equipment specification appears in Table 2. Pressure transmitters are installed at the inlet piping of FCU located at end point, primary supply water header of primary pumps, return water header, and two-way control valve are installed in decoupler. For the verification of effects using this control method, measuring instruments are also installed to measure "its flow rate through chillers" and "consumption energy of each equipment".

Table 2: Main Equipment Specification of the Chiller Plant

Name of equipment	*Specification*
Gas absorption chiller (R-1, R-2)	*Freezing capacity ：1,582kW， Cool water flow rate ：3,240L/min,* *Cool water temperature: 14-7 ℃,* *Cooling water flow rate ：7,400L/min， Cooling water temperature: 32-37.5 ℃* *Rated gas consumption ：123m ^{3} (N)/h*
Primary cool water pump (CHP-1,2)	*Rated flow rate ：3,240L/min， rated head ：150kPa， rated shaft power ：15kW*
Cooling water pump (CDP-1,2)	*Rated flow rate ：7,500L/min， rated head ：200kPa， rated shaft power ：15kW*
Close-type cooling tower (CT-1,2)	*Cooing Capacity ：2,878kW （Outdoor air ：27 ℃WB），* *Circulationg water amount ：7,500L/min，* *Temperature of cooling water temperature ：37.5-32 ℃,* *Rated power consumption of the fans ：5.5kW×3*
Secondary water pump (CHP-3,4,5,6）	*Rated flow rate ：1,620L/min， Rated head ：300kPa， Rated shaft power ：15kW*

3.2 Situation Analyses before Introducing the Control Method

We analyzed data of air-conditioning period, during June to September, 2008, for the situation understanding before introduction. Fig.12 shows an occurring frequency distribution of load flow rate and Fig.13 shows an occurring frequency distribution of secondary pump inverter output, and Fig.14 shows an occurring frequency distribution of Delta-T of primary loop (difference between inlet temperature and outlet temperature of chiller) and secondary loop (difference between secondary supply temperature and return temperature). The inverter output value in this figure assumes that 60 to 0Hz is 100 to 0%. As will be noted from the Fig.12, the percentage of load flow rate that becomes less than 1,620L/min is 97%. 1,620L/min means a rated flow of a secondary pump. Therefore it turns out load flow rate is mostly covered by a secondary pump. The rated flow rate of one chiller unit, the rated flow rate of primary pump, is 3,240L/min. Since the load flow rate is constantly less than half the percentage of 3,240L/min, it is considered that at least 50% water conveyance of primary pump is returned to decoupler. As you see Figure-14, the frequency that the temperature difference on primary loop Delta-t becomes less than 3K is 85%, whereas the frequency on secondary loop Delta-t stays only 40%. Since the average temperature difference on primary

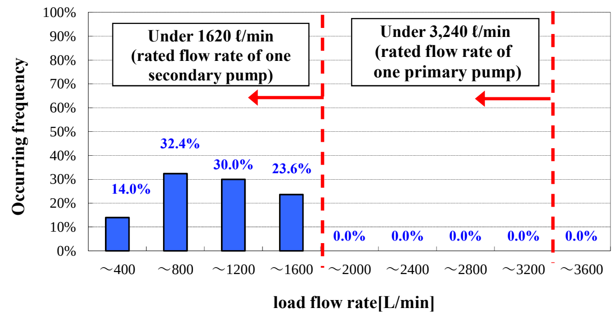

Fig.12. Occurring Frequency Distribution of load flow rate (before introduction)

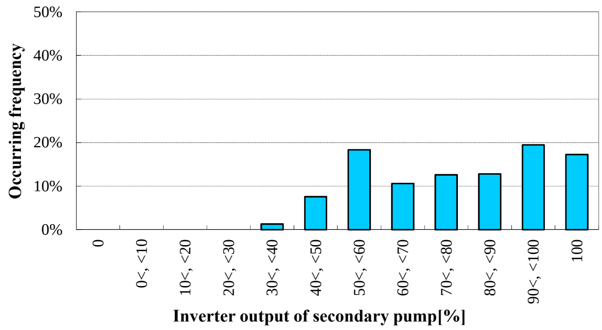

Fig.13. Occurring Frequency Distribution of Inverter Output (Before introduction)

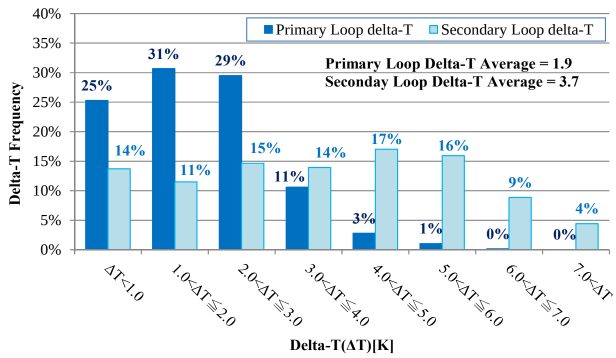

Fig.14. Delta-T Distribution between Primary Loop and Secondary Loop before introduction (June to September, 2008)

loop is 1.9K and the average temperature difference on secondary loop is 3.7K, it means primary loop average is a half of secondary loop average; the maximum thermal load is only about 50% of a chiller capacity. Therefore, it is inferred chiller runs in an inefficient low load operation. Fig.13 shows the inverter output is controlled in 40%-100%. The occurring frequency becoming less than 800L/min is 55%. (800L/min means half percentage because the rated flow rate of secondary pump is 1620L/min.) However the rate that the inverter output becomes less than 50% is only about 10%. Additionally, the case that the inverter output becomes less than 40% had not appeared. As described in chapter 1, “Summary and Problems”, it shows the inverter output does not be under a certain value, because it is required the setting value of water conveyance pressure is kept to satisfy a designed maximum flow rate even if the load flow rate is low.

3.3 Evaluation after Introducing the Control Method

Occurrence frequency appears in each figure as following, Fig.15 shows occurrence load flow rate before or after introduction, Fig.16 shows inverter outputs, and Fig.17 shows power consumption of conveyance pumps. Fig.15 and Fig.16 show inverter outputs of secondary pump are lowered sharply, despite the load flow increases significantly as compared to 2008 (Before introduction) and 2010 (After introduction). As a result, the power consumption of secondary pump was reduced by 93%. The 0% (0Hz) inverter output, as shown in Fig.16, means secondary pump is not operated substantially and all conveyances, both primary conveyance and secondary conveyance, can be covered by only primary pump. It accounts for 37% of the occurrence frequency. In addition, both of secondary pump with inverter and without inverter run idle, when the inverter output of secondary pump is 0, all the other secondary pumps without inverter stop, and only primary pump operates. In point of the pump without inverter running idle, there is no problem because electric voltages are not energized to the standard motor. On the other hand, even if the motor with an inverter is 0% (0Hz), there are concerns about the potential of heat generation since voltages are applied to the motor and it acts as a regenerative brake. However, we think the potentials that the power is applied to the pump as a regenerative brake are extremely low because the discharge flow rate of primary pump is distributed to multiple secondary pumps. Through this test period, the occurring frequency of 0% (0Hz) inverter output of secondary pump was occupied

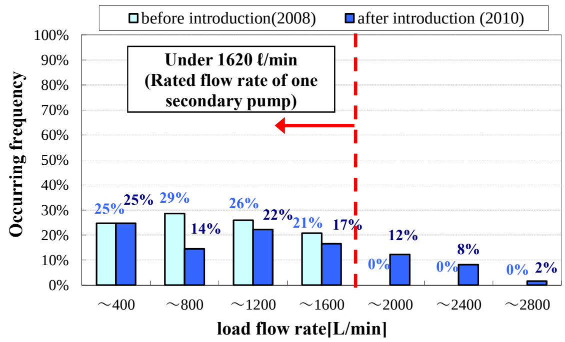

Fig.15. Occurrence load flow rate before or after introduction

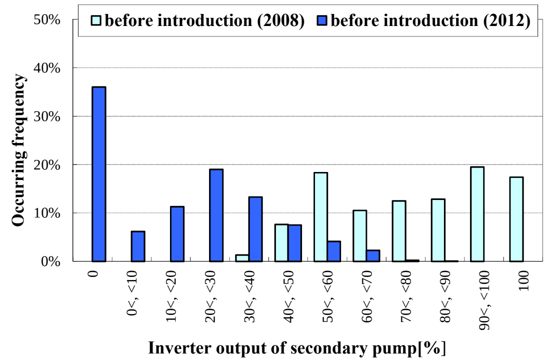

Fig.16. Occurrence Inverter Output of Secondary Pump Before or After Introduction

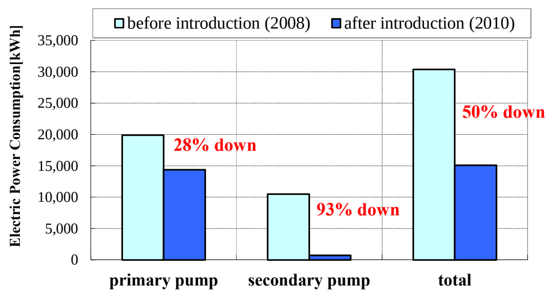

Fig.17. Power Consumption of Conveyance Pumps Before or After Introduction (from June to September)

45% as noted above, but any problems did not occur. We have more evaluation tests in this point hereafter. As shown in Fig.17, the power consumption of primary pump was reduced 28% as with the secondary pump. It can be considered by this result that the power consumption of primary pumps was reduced as much as chiller flow rate was reduced by controlling decoupler valves. 50% reduction of the power consumption was made by both the primary pumps and secondary pumps. Fig.18 shows each an occurring frequency distribution of Delta-T of primary loop and secondary loop and primary loop, as a comparison before and after this control method introduction, which is from June to September in 2008 and same period in 2010. As you see left figure of Fig.18, Delta-T of primary loop largely increased from 1.9K (2008) to 5.2K (2010). Although it is small increase, right figure of Fig.18 shows that Delta-T of secondary loop also increased. As you see Fig.19, the primary loop flow after introduction decreased compared to before introduction (in 2008) by the operation of decoupler valve control. Whereas the primary flow rate in 2008 is approximately constant between 3200 and 3400 L/min, the most frequent flow rate in 2010 was 2400 to 2800 L/min. As a result, the increase in Delta-T of primary loop is thought to be due to the reduction of primary flow rate. The reduction of energy consumption by improvements of chiller operational efficiency is much expected when Delta-T of primary loop increases.

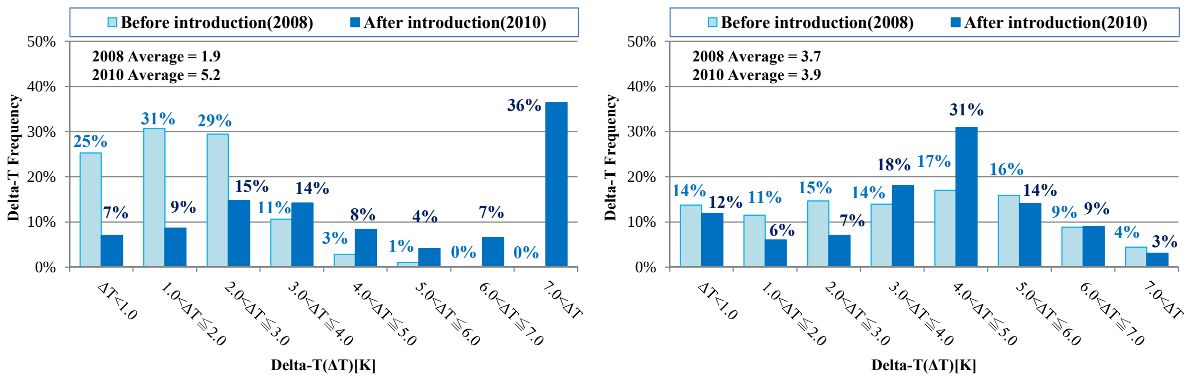

Fig.18 Delta-T Distribution as a Comparison between Before and After Introduction
(Left : primary loop, Right : secondary loop)

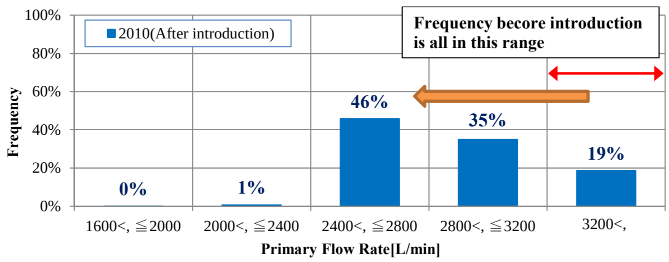

Fig.19 Flow Rate Distribution of Primary Loop after Introduction (June to September)

3.4 Application Condition of This Control Method

This instance is an excessive case such as an actual cooling peak load is less than 50% capacity of a chiller even in the summer maximum load period. Even if the case is extreme, this control method is fundamentally effective for the chiller plants which consist mostly of the partial load operation. The specific applicable condition of this control method is following three points shown in Fig.20;

1) All Primary pumps are constant-speed pumps.

2) At least one secondary pump has inverter, and variable speed control is available.

3) The condition that an actual load is less than a chiller capacity occurs frequently. Especially the case is effective that an actual load is far below its capacity at the time like less than 50% of capacity. Similar case occurs frequently in the office buildings, though the load difference is small in the factories throughout a year.

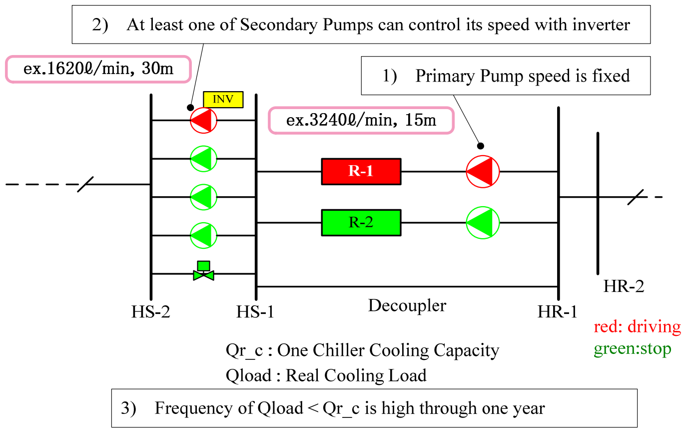

Fig.20 Effective Condition in using the Control Method Using Surplus Pressure of Primary Pumps in Chiller Plant System

4. Extension of Using the Control Method Using Surplus Pressure of Primary Pumps

4.1 Analysis of Differential Pressure on Secondary Side

The differential pressure on secondary side indicates the difference between secondary supply header and return header. It is, therefore, differential pressure of whole water conveyance on secondary side. Fig.21 is a distribution map which shows a correlation between the load flow rate and the differential pressure on the secondary side from June to September. It shows the maximum load flow rate during this period is 2,700L/min which is approximately 83% of the rated flow rate per chiller (3,250L/min). The figure also shows the differential pressure is higher than 70kPa through the whole flow rate, which is a differential pressure value required for total load flow rate (6,480L/min=3,240L/min×2) that is confirmed by this verification experiment. In fact, when the load flow is at least less than 2,700L/min, the secondary pump is no longer required and the superfluous differential pressure is supplied to the secondary side by only operating the primary pump as well. For this reason, further reduction of conveyance power is expected by introducing an inverter to a primary pump and controlling it appropriately.

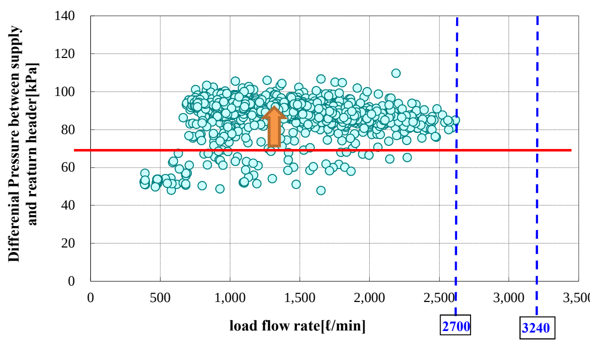

Fig.21 Distribution Map about Correlation between the Load Flow Rate and the Differential Pressure on the Secondary Side from June to September in 2010

4.2 Further Extension of the Control Method Usage

We suggest a countermeasure as handling for the secondary side differential pressure becoming excess by only the primary pump operation. The mechanism is to install an inverter to a primary pump and this inverter is controlled only when the secondary pump is no need to operate. The inverter output of primary pump is fixed around its rated speed when the secondary pump operation is required. When the secondary pump operation is not required, the primary pump inverter is controlled to maintain its appropriate pressure at the secondary side and to secure the smallest chiller flow rate at the same time. When the secondary pump is not required, the controller stops the secondary pump inverter control, and switches from the secondary pump inverter control to the primary pump inverter control. Fig.22 and Fig.23 show the flow chart about the primary pump inverter control.

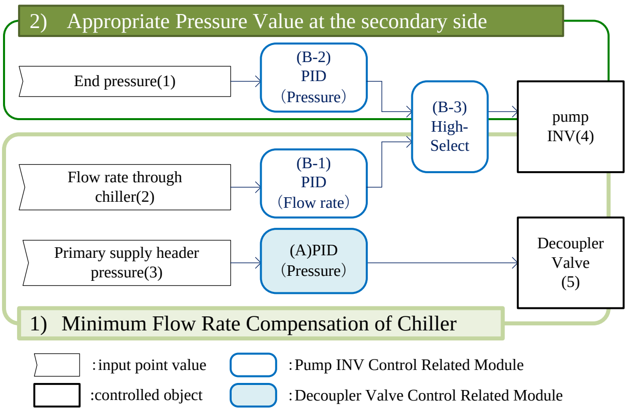

Fig.22 Control Module Constitution at the Time of the Second Pump-free Judgment

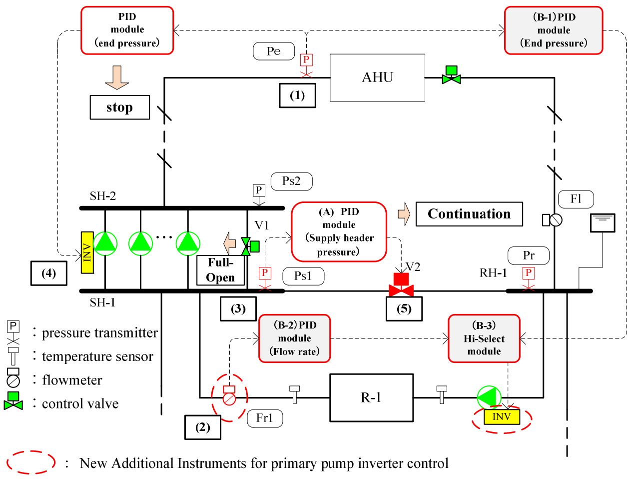

Fig.23 Control Flow at the Time of the Second Pump-free Judgment

1) Stop the secondary pump inverter control. Continue the decoupler valve control with no change.
2) Full-open the bypass valve between secondary supply header and primary supply header.
3) Start the primary pump inverter control. The final inverter output value is decided by high-selecting the result of PID output by the end pressure and by flow rate through chillers in this control.

The secondary pump can stop completely by using the control method that we described, if the actual peak load is lower than expected and can be covered by the primary pump head.

Conclusion:

Since the 1990s, the primary/secondary pump system whose secondary flow fate rate is ever-changing by controlling with two-way valves on the load side is introduced into many buildings. If the primary pump of chiller plant system for this project happened to be designed by over-capacity, downsizing the primary pump to a proper ability was one of the countermeasures. However the primary pump size was selected properly to meet a rated flow rate of a chiller. In other words, it is possible to describe as a result that whole chiller plant system including a chiller was designed excessively instead of a primary pump. There are many cases that a real peak load is much smaller than a design expected peak load and the capacity of chiller plant system has been accordingly excessive against the actual load. In such cases, it is highly possible that the conveyance power can be dramatically reduced by introducing our “Control Method Using Surplus Pressure of Primary Pumps”. In addition, this control method is available by only installing a two-way valve in decoupler on an existing chiller plant system and newly installing pressure sensors at primary supply header and the inlet of distant load, air conditioning. As a result, it shows the installation can be achieved at low cost and it is a cost-effective countermeasure against energy saving. An inverter did not be installed to the primary pump in this instance. However, we consider from this report that significant energy reduction is possible by using this control method after the inverter installation to a primary pump.

Reference:

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