OPTIMAL OPERATING METHOD FOR A HEAT SOURCE SYSTEM CONSIST OF CENTRIFUGAL CHILLERS

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Abstract: The performance characteristic of a variable-speed centrifugal chiller is different from that of conventional fixed-speed centrifugal chiller. The variable-speed centrifugal chiller has highest COP point in partial load (we call optimal load), which varies according to the cooling water temperature. This optimal load can be easily calculated from the machine characteristic of centrifugal chiller. In this paper, we report the calculation of optimal load and the optimal operation method of heat source system using the optimal load.

Key Words: Centrifugal Chiller, operation method, optimal load

1 INTRODUCTION

The centrifugal chiller is a main component of heat source system, which is applied to factories, District heating and cooling (DHC) plant and large buildings. Chiller performance has been significantly advanced by using mechanical loss reduction and adopting speed control of a centrifugal compressor. Its latest model performance has been improved significantly to the rated COP 7.0 as shown in Figure 1. In order to reduce energy consumption of the system, not only chiller performance but also system control method should be improved.
This paper reports the method for controlling a number of chiller units to operate the chiller at highest COP, as an optimal control method for the heat source system. This paper also describes optimization of control of auxiliary components (chilled water variable flow rate, cooling water variable flow rate, and cooling tower control).

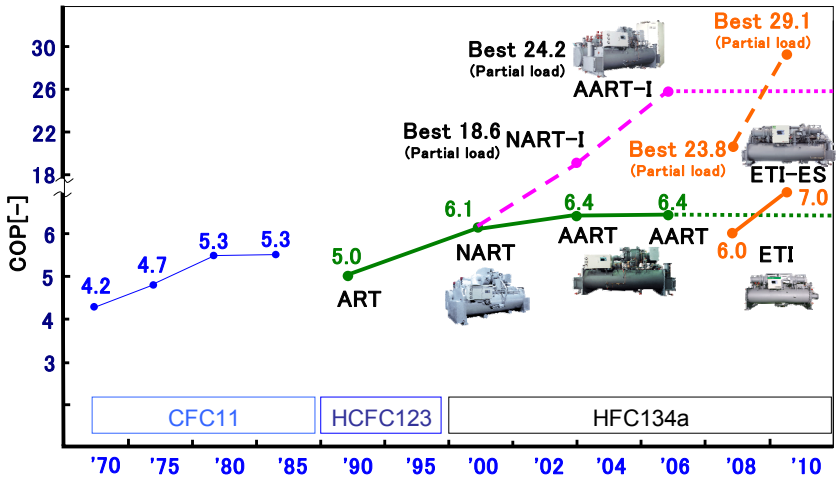


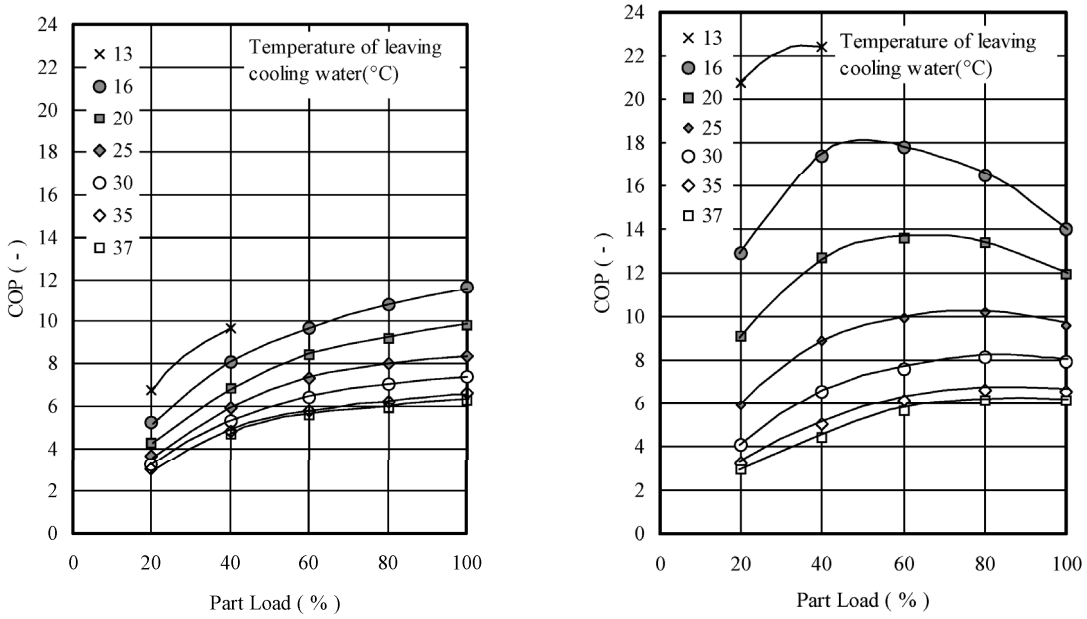
Figure 1: The history of centrifugal chiller

2 Optimal load of centrifugal chiller

2.1 Performance characteristics of centrifugal chiller

As shown in Figure 2, the performance characteristics of the variable-speed centrifugal chiller is significantly different from that of fixed-speed one. The fixed-speed centrifugal chiller has a conventionally known characteristic that the COP usually reaches its highest value at 100% load. On the other hand, the variable-speed chiller has the highest COP value at particular partial load (hereafter the optimal load), which varies according to the cooling water temperature.

It is possible to operate the heat source system efficiently with understanding the characteristics of the centrifugal chillers(Song et al. 2007). Controlling the number of chiller units to correspond the highest COP value depends on the configuration. A fixed-speed chiller is varied according to the rated load; a variable-speed chiller is varied according to optimal load. However, there is a difficulty for changing the control point of chiller unit quantity with optimal load, as shown next section.



(a) Fixed-speed centrifugal chiller (b) Variable-speed centrifugal chiller

Figure 2: COP curves of centrifugal chiller

2.2 Derivation of optimal load

Regarding variable-speed chillers, the partial load characteristics depend on compressors, heat exchangers, cooling capacity and so on. In order to operate the chiller at the optimal load, the performance characteristic should be indicated in numerical data and adapted for the control panel of heat source system. To simplify the performance characteristics, a method for deriving the optimal load, based on the combination thermodynamic characteristic of chiller and theoretical characteristic of centrifugal compressor, was proposed(Ueda et al. 2009).

The performance characteristics of centrifugal compressors are closely related to the adiabatic head, equivalent to the pressure ratio between the evaporating and condensing pressure, and the required refrigerant gas volume for cooling capacity demand. This characteristic of the centrifugal compressor is shown by two dimensionless parameters: discharge coefficient ϕ and pressure coefficient μ. The discharge coefficient indicates the refrigerant gas volume Q_{st} and the pressure coefficient indicates the adiabatic head H. The equations below indicate the loading rate to gain the maximum chiller COP at each cooling water temperature with these two dimensionless parameters.

$$\phi = \frac{Q_{st}}{\frac{\pi}{4} D^2 u} \tag{1}$$

$$\mu = \frac{gH}{u^2} \tag{2}$$

$$u = \pi DN \tag{3}$$

$$\left(\frac{H}{\mu}\right) = \kappa \left(\frac{Q_{st}}{\phi}\right)^2 \tag{4}$$

$$\kappa = \frac{16}{g \pi^2 D^4} \tag{5}$$

Where

ϕ = discharge coefficient [-]
μ = pressure coefficient [-]
u = impeller rim speed [m/s]
Q_{st} = suction volume of compressor [m^3/s]
H = adiabatic head of compressor [m]
D = external diameter of impeller [m]
N = rotating speed of impeller [rpm]
g = acceleration of gravity [m/s^2]

κ is a new fixed parameter, which is calculated from design parameter ϕ, μ and maximum chiller COP operating point Q_{st}, H. The chilled water and cooling water temperatures are almost equal to the evaporation and condensing temperatures; and they are considered to be saturation pressures. When the chiller compressor designed maximum efficient point at the condition of the 37℃ cooling water leaving temperature (indicates H) and 80% cooling load (indicates Q_{st}), κ determined from Equation(4) as κ_{80} ; where index means base

cooling load. Under operating condition, optimal load is derived from Equation(4) by substituting present adiabatic head H , and calculated κ_{80} .

2.3 Calculation of optimal load

In order to evaluate the effectiveness of the optimal load derivation method, the formula for computation was applied to the embedded software used for chiller control. The specifications of chiller are listed in Table 1. Calculation was performed by giving various operational conditions.
The COP curves of the chiller and the calculated optimal load are shown in Figure 3. The optimal load is calculated based on κ_{60} and κ_{90} . If the chilled water outlet temperature is constant, the performance characteristics of centrifugal chiller depend on the cooling water outlet temperature. Therefore, the performance curve is directly related to the cooling water outlet temperature. To determine whether the simulated load range is within the optimal load, partial load rates at the maximum COP of 95% of each cooling water temperature are connected with a line in the performance curve.
Figure 3 shows that the calculated optimal load is within the range of high COP and the calculation formula is effective under each temperature condition of cooling water. The slightly lower load range is calculated at the high cooling water temperature and it is necessary to adjust the default operation parameters (κ).

Table 1: Specification of optimal road simulated chiller

Compressor speed	Variable
Cooling capacity	1,055kW
Chilled water temp	12℃in / 7℃out
Cooling water temp	32℃in / 37℃out

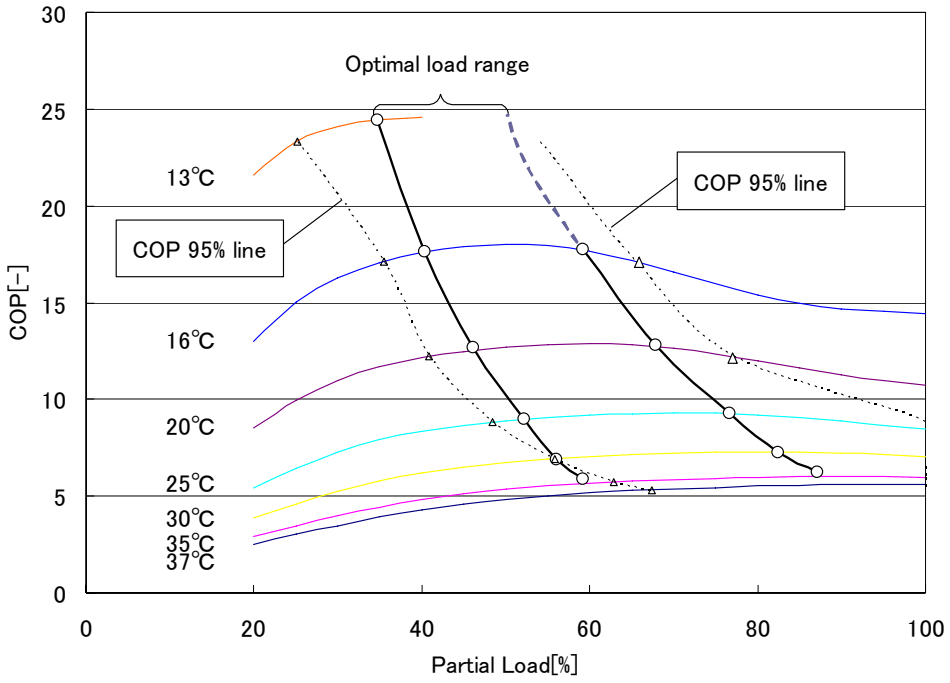


Figure 3: COP curve (Leaving cooling water temp base) and optimal load

3 Optimal control method for heat source systems

In order to reduce the energy consumption of heat source system, next two control method is typically known, such as chiller quantity control and auxiliaries (chilled water pumps, cooling water pumps, cooling towers) control. Table 2 shows the energy saving effect of auxiliaries control (decreased flow rate of the chilled water pump, decreased flow rate of the cooling water pump, decreased cooling air flow rate, and control of cooling tower capacity) on the energy consumption of each component: chiller, chilled water pump, cooling water pump, and cooling tower fan. These controls are used in combination for each heat source systems. There is a fear that the optimal control for one element may be an obstacle to the optimal control of another element.
In spite of that, this paper propose following control method (Nikaido et al. 2010). Proposing optimal control method indicates that each element can be dealt with independently.

Table 2: Energy saving methods and their effect on components and the heat source system

Auxiliaries	**Chilled water pump**	**Cooling water pump**	**Cooling tower**	
Method / **Item**	**Decreased chilled water flow**	**Decreased cooling water flow**	**Decreased cooling air flow**	**Increased number of cooling towers**
Energy consumption of chiller	No Change	Increase	Increase	Decrease
Energy consumption of chilled water pump	Decrease	—	—	—
Energy consumption of cooling water pump	—	Decrease	No change	No change
Energy consumption of cooling tower fan	—	No change	Decrease	Increase
Energy consumption of heat source system	Decrease	Increase or Decrease	Increase or Decrease	Increase or Decrease

3.1 Chiller quantity control

Based on the operation of an appropriate number of chiller units corresponding to the cooling load, increase the number of stages if the load rate of the chiller is higher than the calculated optimal load.

3.2 Auxiliaries control

-Chilled water variable flow rate controlled with respect to the chilled water demand of the load.
-Cooling water variable flow rate controlled with respect to the cooling load, show as Figure 4.
-The air flow rate controlled with wet bulb temperature. If the cooling water pipework is integrated within the cooling tower (at the headers), cooling tower capacity is controlled with respect to the cooling load and wet bulb temperature.

3.3 Optimal control method

Considering the actual operation of heat source systems, the control method should be developed by a simple algorithm. Figure 5 shows an algorithm in which the calculation starts with the wet bulb temperature and cooling load ratio. It is possible to determine the chilled water flow rate, number of cooling towers, and cooling water flow rate after determining the number of chillers in accordance with the cooling load and the outside air condition.

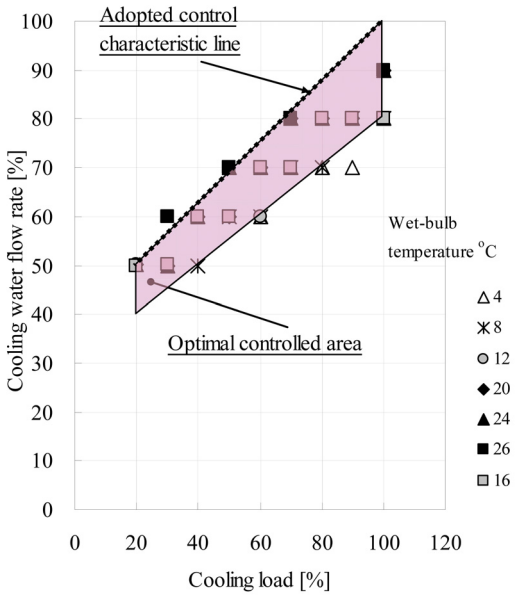


Figure 4: Optimal control of cooling water flow rate

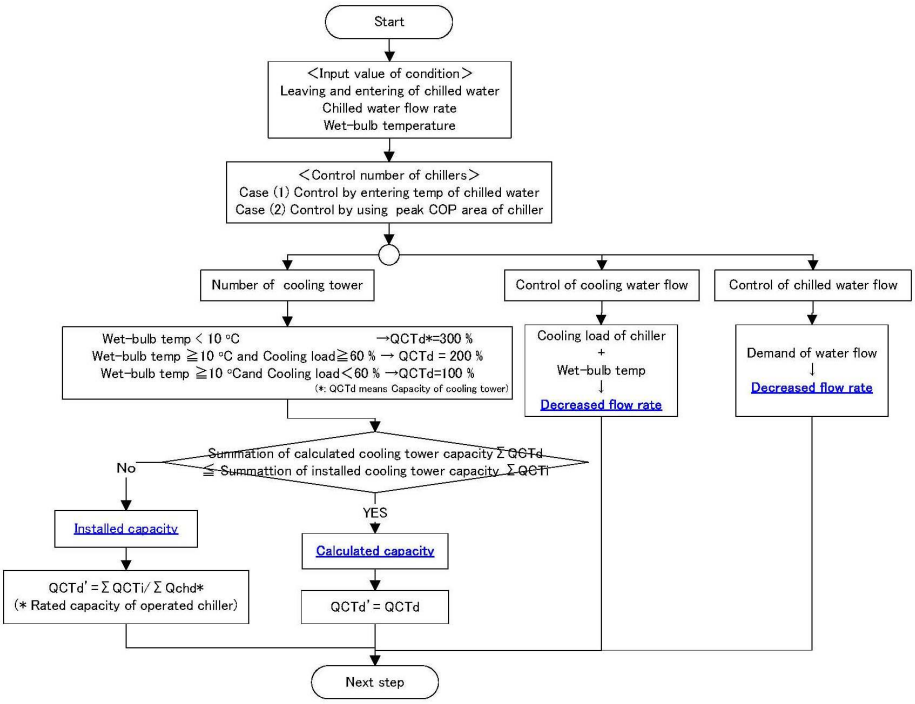


Figure 5: Proposed Control algorithm of heat source system

4 Effectiveness of energy conservation

The effectiveness of Optimal Control for heating source system was confirmed by the simulation.

4.1 Calculation conditions

The cooling load is shown in Figure 6 below. Figure 6 is the actual data of a factory operating for 365 days with 24 hours operation. The heat source system consists of three centrifugal

chillers, three chilled water pump, three cooling water pumps and three cooling towers, as shown Figure 7. Table 3 shows the machine specification. Table 4 shows the operational simulation cases. Case2 uses the latest fixed-speed chillers; Case 3,4 & 5 use the latest variable-speed chillers; Case 4 & 5 use the optimal control of auxiliaries and Case 5 utilizes the control of a number of units at the optimal load.

The heat source system COP (COP_{sys}) is defined by Equation(6) and calculated by dividing the chiller heat output Q_{tb} (from which the heat input of chilled water pumps is deducted) by the energy input. Energy input is the total energy consumption of the chiller P_{tb}, chilled water pumps P_{chp}, cooling water pumps P_{clp}, and fan motor of the cooling tower P_{ct}.

$$COP_{sys} = \frac{Q_{tb} - \eta_{mp} \cdot P_{chp}}{P_{tb} + P_{chp} + P_{clp} + P_{ct}} \tag{6}$$

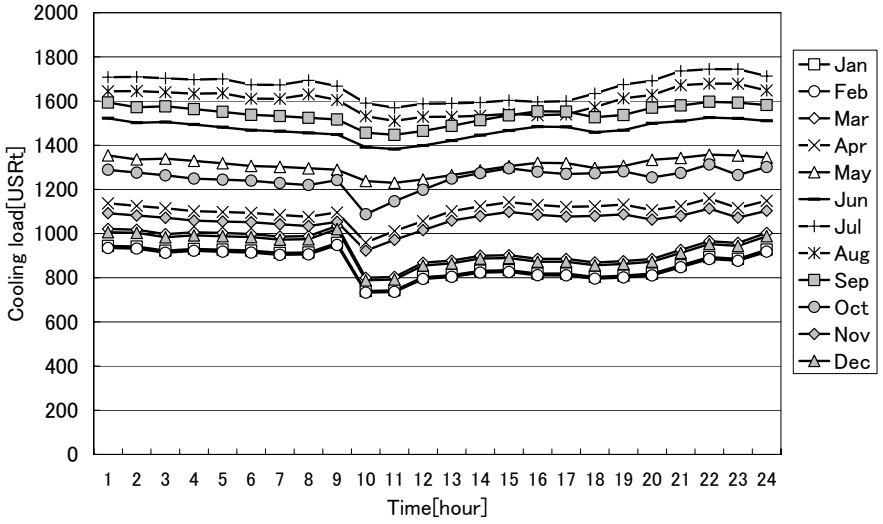


Figure 6: Cooling load

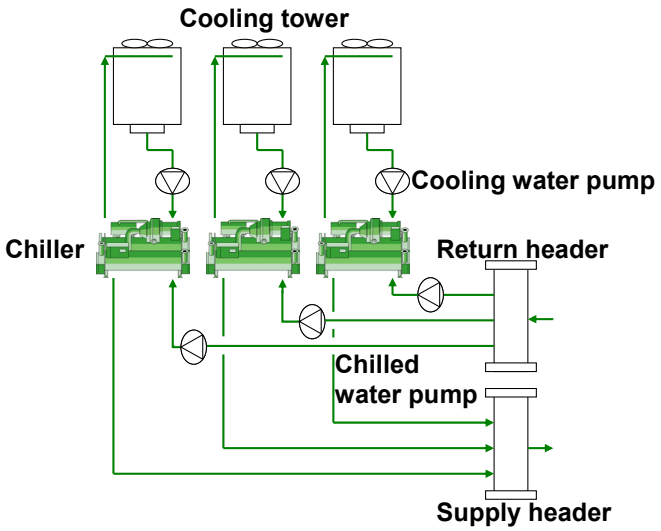


Figure 7: Heat source plant configuration

Table 3: Machine specification

Chiller	**Unit**	3
	Cooling capacity	3,516kW
	Chilled water temp	12°Cin / 7°Cout
	Chilled water flow	604.8m^3/h
	Cooling water temp	37°Cin / 32°Cout
	Cooling water flow	710.8m^3/h
Cooling tower	**Unit**	3
	Fan motor input	7.5kW × 4fan
Chilled water pump	**Unit**	3
	Pump motor input	37kW× 2set
	Pump efficiency	0.75
Cooling water pump	**Unit**	3
	Pump motor input	45kW × 2set
	Pump efficiency	0.75

Table 4: Simulation pattern

	Case1	**Case2**	**Case3**	**Case4**	**Case5**
Chiller	Old fixed-speed chiller COP : 4.91	Latest fixed-speed chiller COP : 6.32	Latest variable-speed chiller COP : 6.16		
Auxiliaries control	Fixed-speed			Variable-speed	
Quantity control	Cooling load base (Change by 70% load)				Optimal load base

4.2 Result of calculation

The result is shown in Figure 8. Replacement of centrifugal chiller, from Cace1 to Case2 and from Case2 to Case3, shows a significant energy reduction. Case 4 shows that the power consumption of overall configuration is significantly reduced by introducing optimal control of auxiliaries. It also shows that the chiller COP decreases from 10.3 to 9.5, because the cooling water outlet temperature increases as the cooling water flow rate decreases. Case 5 shows that the COP_{sys} increases by 0.1 from 7.2 to 7.3, demonstrating the number of units are controlled at the optimal load. The chiller COP is also increased from 9.5 to 9.6 by chiller quantity controlling at the optimal load.
The result of the calculation shows that annually a 52% reduction of power consumption can be expected by updating the chiller to the latest variable speed model and optimizing the system control.

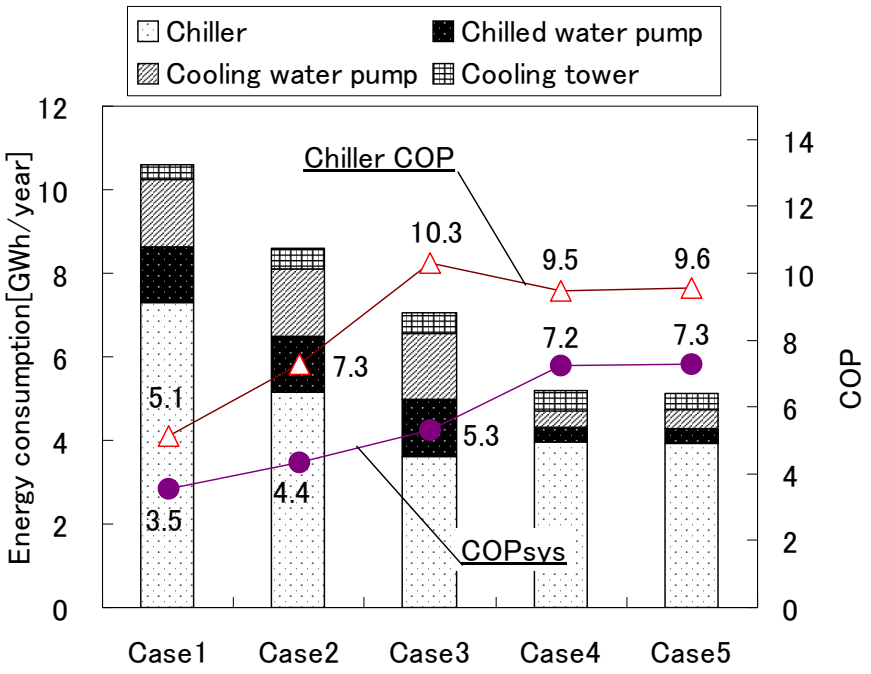


Figure 8: Simulation result

5 Conclusion

- It was confirmed that the calculated of the optimal load based on the thermodynamic property of the chiller was appropriate.
- The system efficiency has been improved by optimizing auxiliaries control and further efficient operation can be achieved by optimizing the control of number of units.

The control method introduced in this paper has been achieved with the MHI control panel called “Ene-conductor”. It can be expected that this control method will be applied to the heat source equipment and that further energy conservation will be promoted.

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