Reduction of Conveyance Power Consumption of District Cooling and Heating Systems using Demand-Supply Coordinated Control

Part 2 - Energy Saving Effect of Demand-Supply Coordinated Control System

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Abstract. This study aims to discuss the effectiveness of "Demand-Supply Coordinated control" in reducing the power consumption required for conveyance by the heat transport medium in District Cooling and Heating (DHC) systems. The problem with DHC systems is that increased conveyance power is required to provide heating to consumers. As one of the measures to solve this problem, Demand-Supply Coordinated (DSC) control is introduced; however, its effectiveness and limitations have not been clarified so far. In this paper, first, the fundamental characteristics of a DHC system under DSC control are numerically examined. The results showed that the conveyance power consumption of DHC systems under DSC control can be classified into three regions, depending on the relative rate of demand change against the load-following capability of the heat source. Next, the authors compared the conveyance power of DSC control with that of Constant Supply Pressure (CSP) control adopted in conventional DHC, and showed regularity for each of the three regions mentioned above. Finally, the authors show the appearance frequency of these three regions of the practical DHC system under a real heat load in Japan. The results showed that the conveyance power than that of CSP control.

1 Introduction

District cooling and heating (DHC) systems aim to introduce an efficient large-scale concentrated heat source for supplying heat to multiple buildings in a specific area in order to reduce the energy consumption in the whole area. However, there is the problem that the conveyance power becomes large, particularly under the conditions where partial load operation of the heat source occurs^{1),2)}. Here, conveyance power is the power consumption per supply heat quantity of a pump which conveys cold and hot water from a concentrated heat source to each building (hereinafter referred to as a conveying pump). In countries with four distinct seasons such as Japan, the thermal load varies significantly depending on the season, so it is important to reduce conveyance power. To reduce conveyance power, it is necessary to control the flow rate and head of the pump to values commensurate with the momentarily changing heat demand. In recent years, the number of cases where a reduction of conveyance power by introducing an inverter driven conveying pump to attain variable flow control is attempted^{3),5)} has increased. However, in the situation where the control of heat consumers who use cold and hot water according to heat demand is independent of the supply side control that controls the operation of the heat source and conveying pump, there is a limit to the reduction of conveyance power.

One of the measures to solve this problem is demandsupply coordinated (DSC) control. This is a function to control energy at the district level in a planar and optimal manner using information communication technology. By introducing this into DHC systems, DSC control including not only the heat supply side, but also the heat demand side, becomes possible, and more effective reduction of conveyance power can be realized. Practical DHC systems that introduce DSC control⁴⁾ are just beginning to be presented, however, in a certain site in Tokyo that started operation in November 2014, a significant reduction of conveyance power⁸⁾ has been reported.

This research aims to verify, in a general manner as far as possible, how effective DHC systems to which DSC control is introduced are for reduction of the conveyance power, and what their constraints are. In this report, in the case of one heat consumer, basic characteristics of the conveyance power in DHC systems with DSC control were discussed focusing on the relative thermal load fluctuation rate to the load following speed of the heat source machine. Next, the characteristics of DHC systems with DSC control were compared with the system with ordinary constant supply pressure (CSP) control, and the conditions under which DSC control

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reduces the conveyance power more effectively than CSP control were described. The results showed that DSC control is effective to reduce the conveyance power than CSP control in general, but that the efficiency decreases with the relative thermal load fluctuation rate increases. Using the results, the authors concluded that significant energy savings can be realized by introducing DSC control into practical DHC systems in a certain site in Tokyo area, because thermal load fluctuation faster than the following speed of the heat source machine seldom appears under the practical conditions. We believe that the concept shown in this report can be applied as a guide for determining whether the conveyance energy saving measure is functioning or not.

Major Symbols

ujor byn	10015	
d	:Motor efficiency	[-]
Ε	:Pump efficiency	[-]
е	:Inverter efficiency	[-]
т	:Sine wave cycle number after stabilization	[-]
Pave	:Conveyance power (hourly average of P_w)	[kW•h/h]
Pmin	:Reference value of conveyance power	[kW•h/h]
P_w	: Pump power consumption	[kW]
Q_2	: Secondary-side heat load	[kW]
$d\tilde{Q}_2/dt$: Heat load gradient	[kW/s]
Q_R	:Output of heat source equipment	[kW]
dQ_R/dt	: Heat load following speed of heat source equipment	[kW/s]
q	: Secondary-side flow rate	$[m^3/h]$
\hat{q}_1	: Primary-side flow rate	$[m^3/h]$
q_{1max}	:Primary-side maximum flow rate	$[m^3/h]$
- 	:Secondary-side maximum flow	[m ³ /h]
q_{max}	rate	
q_{min}	:Secondary-side minimum flow rate	[m ³ /h]
S	:Unstable control period	[s]
Т	:Sine wave cycle	[s]
T_{in}	:Inlet temperature of heat source equipment	[C]
T_{out}	:Outlet temperature of heat source equipment	[°C]
T_r	:Inlet / outlet temperature difference of heat source	[K]
ν	equipment Heat capacity per unit of volume	[kJ/m ³ /K]
ΔT_e	:Secondary side temperature	[K]
	difference	
δ	:Nondimensionalization factor of	[-]
	:Nondimensionalization factor of	[-]
ρ	sine wave period	LJ

2 Overview of DSC control and CSP control

In DHC systems in recent years, CSP control has been carried out by introducing an inverter, which have become inexpensive, for the conveying pump. In the case of this control, if one of the heat consumers connected to the system has a large differential pressure requirement for feeding and returning cold/hot water (hereinafter referred to as "supply differential pressure"), the energy saving effect is reduced. For this reason, new DHC systems in Japan commonly introduce a heat exchanger into the heat receiving facility which is the combining point between the heat supply side and the heat demand side, so as to separate the differential pressures between the heat supply side (heat exchanger primary side) and the heat demand side (heat exchanger secondary side). In this case, a regulating valve or temperature control valve (hereinafter referred to as TCV) that adjusts the flow rate of the heat exchanger primary side is introduced so as to control the thermal load corresponding to each heat consumer' demand (Figure 1 upper row). From the viewpoint of energy saving, it is favorable to fully open the TCV of the first heat consumer for which the supply side cannot secure the required flow rate (hereinafter referred to as the farthest-end heat consumer) to reduce the supply differential pressure or conveying pump power on the heat supply side. However, in actual conditions, it is difficult to fully open the TCV of the farthest-end heat consumer, since each heat consumer has individual control. Therefore the supply differential pressure is usually kept constant. In addition, the temperature and flow rate information of each heat consumer (hereinafter referred to as secondary side information) cannot be acquired from the heat supply side. Therefore, it is necessary for the heat supply side to set the supply differential pressure to be constant to prevent a shortage of the flow and deviation of the setting value of the heat exchanger secondary side outlet temperature for all heat consumers regardless of thermal load variation and the flow rate ratio between heat consumers, resulting in wasted conveyance power⁵⁾.

On the other hand, with the heat conveyance control of DHC systems to which DSC control is introduced, it is possible to realize variable differential pressure and variable flow rate control of the conveying pump using heat demand side information. Based on the secondary side information of all heat consumer heat exchangers, the heat exchanger of the farthest-end heat consumer determined by the flow rate ratio between heat consumers is identified, and the TCV is controlled from the heat supply side so that the TCV opens fully (Figure 1 lower row). By linking this TCV opening control with the conveying pump flow rate and head control, it is possible to convey cold and hot water without waste, which enables a significant reduction of conveyance power^{61,71,81}.



Fig. 1. Comparison between CSP control and DSC control

3 Simulation model

In order to quantitatively and generally examine the conveyance power reducing effect resulting from the change from CSP control to DSC control, two numerical simulation models (hereinafter referred to as model(s)) were built for simulating DHC systems to which the control shown in Figure 1 was introduced. That is, the models assumed one heat source machine and one conveying pump on the heat supply side, and multiple heat consumers. However, the number of heat exchangers assumed in the model was one for each building, and thus the effect of controlling the number of heat source machines, conveying pumps, and heat exchangers was not considered. Models of equipment and piping were simplified as much as possible, but the temperature delay of the fluid depending on the length of piping was considered because the effect could not be ignored in the evaluation of DHC systems.

As an example, a model to which DSC control is introduced is depicted in Figure 2. In combination with the element model described below, the behavior of DHC systems was simulated using MATLAB® Simulink® and SimscapeTM functions.



Fig. 2. Simulation model of DSC control

3.1. Heat source machine model

It was assumed that the heat source machine ideally controlled the outlet temperature to 47°C regardless of the inlet temperature and the flow rate. However, the amount of heat generated was limited to the rated capacity of 3.16 GJ/h or less (pump rated flow rate × standard temperature difference 5°C) and the rate of increase or decrease of the generated heat amount (hereinafter referred to as the thermal load following speed $dQ_{k'}dt$) was also limited.

The heat source heat output Q_R and the thermal load following speed dQ_R/dt of the heat source machine are given in equations (1) and (2).

The thermal load following speed dQ_R/dt was set to 3.87 kW/s, 1.93 kW/s, 0.97 kW/s (cold following speed of general steam absorption type high temperature regenerators), and 0.39 kW/s for temporal increase of the generated heat amount, and was set commonly to 5.27 kW/s for temporal decrease of the generated heat amount.

3.2. Conveyance model

One inverter driven conveying pump was used and starting and stopping of the pump was not considered for the model. The pump performance curve was set to the actual values of the pump with the maximum capacity among those introduced to a certain site in Tokyo, instead of the values in the catalog. The capacity could be changed ideally from the rated 151.2 m³/h to 15.12 m³/h (10% of the rated value) with a head of 50 m. The flow rate lower limit for protecting the heat source machine and restrictions of the flow rate changing speed were ignored. For piping conditions (piping length, number of bends size, and so on), those of the facility of a certain site in Tokyo were adopted. Heat dissipation losses were ignored.

In DSC control, the TCV installed on the primary side of the heat exchanger was always fully opened as there was one heat consumer. The inverter output of the conveying pump was PID controlled so that the outlet temperature on the secondary side of the heat exchanger was constant at 46°C (substantive secondary flow rate = primary flow rate).

In CSP control, the TCV was PID controlled so that the outlet temperature on the secondary side of the heat exchanger was constant. The inverter output of the conveying pump was PID controlled so that the supply differential pressure was constant. This supply differential pressure was set to 0.5 MPa, which was the maximum among the supply differential pressures in DSC control. When the supply differential pressure was lowered to 0.4 MPa, the supply temperature did not become constant at 47°C and became unstable.

3.3. Heat demand side model

It is assumed that there is one heat exchanger at the demand side and its heat exchange is ideal. The specification is set so that the temperature decrease of 1° C occurs when heat is transferred from the primary side to the secondary side of the heat exchanger; e.g. if the inlet temperature of the primary side of the heat exchanger is 47° C, the outlet temperature of the secondary side is 46° C.

4 Simulation assessments

Using the simulation models described in the previous section, characteristics concerning the basic properties of the relationship between thermal load variation of DSC control, load following speed of the heat source machine, and conveyance power were first evaluated.

Next, CSP control and DSC control were compared, and their regularity was evaluated.

4.1. Heat load condition

In this simulation, a sinusoidally fluctuating thermal load was assumed in order to perform verification as generally as possible. The secondary side flow rate q of the thermal load was set to a sinusoidal wave as shown in equation (3), and the phase was not taken into account because of there being only one building. The secondary side flow rate changed from the maximum value q_{max} of 151.2 m³/h to the minimum value q_{min} of 15.12 m³/h, which is the same as the changeable flow rate range of the conveying pump. Cycle time T of the sine wave was changed in the range of 500 to 80,000 s. For the thermal load temperature, the inlet side temperature of the heat exchanger secondary side was fixed at 41°C, and the outlet temperature of the secondary side was changed at a setting of 46°C. The thermal load Q_2 and the thermal load changing rate dQ_2/dt are expressed by equations (4) and (5).

$$q(t) = \frac{\left(q_{max} - q_{min}\right)}{2} \sin\left(\frac{2\pi}{T} \cdot t\right) + \frac{\left(q_{max} + q_{min}\right)}{2} \qquad \dots \dots (3)$$

$$Q_2(t) = \frac{q(t)}{3600} \cdot \Delta T_e \cdot \gamma \qquad \dots \dots (4)$$

$$\frac{dQ_2}{dt} = \frac{\pi \cdot dT_e \cdot \gamma \left(q_{max} - q_{min}\right)}{3600T} \cos\left(\frac{2\pi}{T} \cdot t\right) \qquad \dots \dots (5)$$

The thermal load changing rate dQ_2/dt , which is an important factor in this verification, reaches the maximum when $cos(2 \pi t/T)$ in Equation (5) equals 1. On the secondary side of the heat exchanger, the outlet temperature and the inlet temperature become almost stable at 46° C and 41° C, respectively, so it is assumed that ΔT_e is fixed at 5° C (refer to Figure 3).



Fig. 3. Heat load profile approximated by a sine wave

4.2. Calculation method of conveyance power

Equation (6) gives the calculation formula of conveyance power. In order to exclude the influence of the transient characteristic, the value over the time period of 5,000 s immediately after the start of control is fixed at the lower limit value of the sine wave q_{min} as the initial value (control instabilization time s), and this time period is not included in the calculation of conveyance power (refer to Figure 3).

$$P_{ave} = \frac{\int_{s}^{s-T \cdot m} P_{w}(t) dt}{T \cdot m} \qquad \dots \dots (6)$$

4.3. Fundamental characteristics assessment of DSC control

Figure 4 shows the basic characteristics of conveyance power of the DSC control applied to a DHC system. The x-axis of this graph is the cycle time T of thermal load (secondary flow) and the y-axis is the avarage conveyance power P_{ave} . When the period of the thermal load T is sufficiently large, the conveyance power P_{ave} converges to the estimated minimum value (7.83 kW•h/h) regardless of the thermal load following speed of the heat source machine dQ_R/dt . Even when the cycle time T is changed to 40,000, 80,000, 160,000, and 320,000 s with the thermal load following speed dQ_R/dt = 3.87 kW/s, the value of the conveyance power P_{ave} became an identical value 7.83 kW•h/h. This means that, therefore, the conveyance power is uniquely determined without depending on the characteristics of the heat source machine if the thermal load fluctuation is very small. As shown in Figure 4, the conveyance power rapidly increases when the cycle time T of thermal load becomes shorter than a certain value, and the threshold value depends on the thermal load following speed of the heat source machine dQ_R/dt .



In order to discuss the change of conveyance power more generally, the authors rearranged the data by using the relative changing rate of thermal load, or dimensionless period coefficient, ρ defined in Equation (7). The result was shown in Figure 5. In this figure, the data were plotted against the ρ_{max} which was evaluated based on the maximum value of the thermal load changing rate. The conveyance power was also normalized by using the standard conveyance power P_{min} , which is the conveyance power in the case where the one-hour integrated average value of the primary flow rate is assumed to be a constant value. The normalized conveyance power δ is defined as Equation (8).





As illustrated in Figure 5, dimensionless conveyance power was divided into the following three regions based on dimensionless period coefficient ρ_{max} :

- Region 1 $\rho_{max} < 0.3$

- Region 2 $0.3 < \rho_{max} < 1.6$

- Region 3 $1.6 < \rho_{max}$

Region 5 $1.0 < p_{max}$

In the region 1, dimensionless conveyance power coefficient δ is about 1.53 (within \pm 5%) and independent of the thermal load following speed dQ_{R}/dt . As an example, Figure 6 depicts a time history of supply temperature and conveying pump power (conveyance power), in which dimensionless period coefficient ρ_{max} is 0.08 (thermal load period *T* is 80,000 s) and the thermal load following speed dQ_{R}/dt is 0.39 kW/s. The supply feeding temperature and the supply returning temperature are stable. Since the heat exchanger primary flow rate (supply flow rate) forms a sine wave, the pump head (supply differential pressure) proportional to the square of the supply flow rate and the conveyance power proportional to the cube of the supply flow rate form ideal graphs.

In the region 2, the supply temperature and control are stable, and the conveyance power slightly increases with increasing in the dimensionless period coefficient ρ_{max} (decreasing in the period *T* of the thermal load). As an example, Figure 7 shows the supply temperature and the conveying pump power when ρ_{max} is 1.00 (period *T* of thermal load is 6,426 s) and dQ_R/dt is 0.39 kW/s. Period *T* in the region 2 is shorter than that in the region 1, and thus the transient state immediately after control instable time *s* in the conveyance power becomes relatively longer than in the region 1. This is one reason why the

conveyance power slightly increases in the region 2. Another reason is that, since the supply temperature slightly dropped due to a control delay near the maximum thermal load (the upper limit of the sine wave) the supply differential pressure or conveyance power was slightly increased to follow up as presented in Figure 8.

The region 3 is an unstable control region in which the target value of the supply feeding temperature cannot be maintained. As an example, Figure 9 gives the supply temperature and the conveying pump power in which ρ_{max} is 1.61 (the period of thermal load *T* is 4,000 s) and dQ_R/dt is 0.39 kW/s. Because a delay occurred in the pump flow rate (inverter) control and the inverter output command value peaked at the upper limit setting, the pump flow rate (electric power) was not a sine wave and distortion occurred. Consequently, both the supply feeding temperature and the supply returning temperature deviated from their target values near the maximum thermal load, resulting in unstable control. For this reason, conveyance power higher than usual is required.







Fig. 8. Relationship between cycle and conveyance power (in an enlarged view)





Using the similar concept shown in Figure 5, the basic characteristics of conveyance power of CPS control were compared with those of DSC control in Figure 10. As shown in this figure, in the region 1 ($\rho_{max} < 0.3$), dimensionless conveyance power coefficient δ was nearly constant at 5.15 in CPS control, while 1.53 in DSC control; conveyance power of the DHC system with CPS control is 3.37 times larger than that of DSC control. Figure 11 shows a time history of the conveying pump power (conveyance power) in CSP control under the same conditions as in Figure 6. Comparing DSC control and CSP control, the supply flow rate changes in the same manner because the change of thermal load is identical with each other. However the supply differential pressure is fixed at 0.5 MPa in CSP control, while it fluctuates regularly within the range of 0.01 to 0.31 MPa resulting in the average of 0.15 MPa in DSC control. Therefore, the conveyance power in CPS control is much larger than in DSC control even under the conditions of region 1.

In the region 2 (0.3 < ρ_{max} < 1.6), the gradient of dimensionless conveyance power coefficient δ with respect to dimensionless period coefficient ρ_{max} is gentler in CSP control than in DSC control. Therefore, the larger dimensionless period coefficient ρ_{max} was, the smaller the difference in dimensionless conveyance power coefficient δ between DSC control and CSP control became. For example, Figure 12 depicts a time history of conveying pump power (conveyance power) in CSP control under the same conditions as in Figure 7. The conveyance power in the transient state is smaller in CSP control (Figure 12) than in DSC control (Figure 7). Therefore, the shorter period T is, the smaller the difference in the conveyance power in the transient state becomes, so the difference in dimensionless conveyance power coefficient δ between DSC control and CSP control becomes smaller.

In the region 3 (1.6 < ρ_{max}), the difference in dimensionless conveyance power coefficient δ between DSC control and CSP control has no regularity. This is because the control in this region is unstable, and thus the difference in dimensionless conveyance power coefficient δ between DSC control and CSP control may become small or large depending on the controllability.





Fig. 11. Region 1 ($\rho_{max} = 0.08$) detailed graphs in CSP control



Fig. 12. Region 2 ($\rho_{max} = 1.00$) detailed graphs in CSP control

5 Energy saving effect of DHC system with DSC control

The previous chapter showed that the conveyance power of DHC systems with DSC control can be classified into three regions by dimensionless period coefficient ρ_{max} , and was compared with that in CSP control in each region. In order to discuss the effectiveness of DHC system with DSC control from the practical point of view, the authors evaluated the appearance frequency of each region in the real world from the actual thermal load of a DHC system with DSC control at a certain site in Tokyo on December 18, 2017, a representative winter day.

Figure 13 shows time histories of the actual thermal load and corresponding dimensionless period coefficient ρ_{max} . The thermal load data plotted in this figure are the moving average values for 20 minutes so as to eliminate the fluctuations due to control process. For calculating the dimensionless period coefficient ρ_{max} , the thermal load following speed dQ_R/dt of the heat source machine was set to 0.97 kW/s, which was the value of the machine operating on the day.

As shown in this figure, the dimensionless period coefficients of this day were distributed in the range from 0 to 1.4, but there was no datum exceeding the value of 1.6. Namely, the appearance frequency of ρ_{max} in the region 3 where the control is unstable and the energy saving effect is unknown was 0%. In contrast, the appearance frequency of ρ_{max} in region 1 was 93.4% where energy savings of 70% can be expected by changing from CSP control to DSC control.

To make sure of the deduction, similar evaluations were carried out for the data measured on a representative spring day (March 16, 2016) at the same site. Even in this case, the appearance frequency of ρ_{max} in the region 3 was 0%, and in region 1 was 96.1%.

It is also inferred that the tendency in autumn is similar to that in spring, and that in summer over an entire day falls in region 1 due to the small thermal load. For these reasons, it was expected that, in the country with its four distinct seasons like Japan, the introduction of DSC control to DHC systems realizes significant energy savings.



6 Conclusions

In this paper, the fundamental characteristics of a DHC system under DSC control are numerically examined.

In the numerical calculations, heat demand fluctuation was assumed by sinusoidal functions, and the conveyance power consumption of DHC systems with heat source equipment having different load-following capabilities was evaluated for various rates of heat demand change. The results showed that the conveyance power consumption of DHC systems under DSC control can be classified into three regions, depending on the relative rate of demand change against the loadfollowing capability of the heat source. Namely, in "Region 1 (the relative rate < 0.3)", the conveyance power is constant. In "Region 2 (0.3 < the relative rate <1.6)", there is a slight increase in the conveyance power with increase in the relative rate of demand change. In "Region 3 (1.6 < the relative rate)", there is a sharp increase in the conveyance power, because the supply temperature is uncontrollable.

Next, the authors compared the conveyance power of DSC control with that of CSP control adopted in conventional DHC. The results showed that the conveyance power required for DSC control is markedly lower than that of CSP control, and that the difference is constant in the region 1. In the region 2, however, the difference slightly decreases by increasing the relative rate of demand change, since the conveyance power of DSC control increases much faster than for CSP control case. The region 3 has no regularity in the difference.

Finally, the appearance frequency of these three regions of the practical DHC system under a real heat load in Japan was discussed. The appearance probability of the region 2 is 6.6% in winter and 3.9% in spring, and most of it was in the region 1. Region 3 did not appear in the practical DHC system. This means that DSC control

could effectively realize energy saving in practical DHC systems.

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