

**Sub-cooling System on Multiple Packaged Air Conditioner by adding External Cold Heat
- A Proposal of Cooling Capacity Estimation method
for Onsite Performance Verification of Inverter-drive Packaged Air Conditioner -**

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ABSTRACT

We developed the Sub-cooling System on Multiple Packaged Air Conditioners (PACs). This system saves the energy consumption by adding the external cold heat into refrigeration cycle through an additional heat exchanger. In this study, as an application of refrigeration cycle behaviors with such the sub-cooling system, we propose an estimation method of cooling capacity, in order to realize the onsite performance verification conveniently without flow rate data of air and refrigerant, on inverter-driven PACs. And then, the possibility of proposed estimation method is evaluated.

NOMENCLATUR

G : refrigerant flow rate	[kg/h]
h : specific enthalpy	[kJ/kg]
P : pressure	[kPa]
Q : cooling capacity	[kW]
Q_{in} : cold or hot heat added to refrigerant pipe	[kW]
T : temperature	[°C]
W : compressor power consumption	[kW]
η : efficiency	[-]

INTRODUCTION

In Japan, recently, many large business use buildings have been equipped with air-cooled PACs such as building multi-unit systems that comply with the requirements of individual distributed air conditioning system. Further improvement of the PACs performance will be needed from view point of global environmental problems. Also, a simple on-site performance measurement method is desired for the maintenance and management of PACs.

We have developed Sub-cooling System on Multiple PACs^{1, 2)}. In this system, the liquid refrigerant is cooled using various cooling water heat sources via an additional refrigerant/water heat exchanger. On an inverter-driven PAC with a constant cooling capacity operation, the specific enthalpy

difference of refrigerant in the indoor unit heat exchanger is expanded in accordance with the heat exchange rate at the refrigerant/water heat exchanger, thus refrigerant flow rate is decreased and the compressor power consumption is reduced.

In this study, based on the concept of the Sub-cool System on Multiple PACs, we propose a simple method to estimate the cooling capacity on inverter-driven PACs without information about the refrigerant flow rate and the air temperature, humidity and flow rate³⁾. First, results of performance test on the energy savings and refrigeration cycle behavior under several cooling water temperatures are reported to explain the concept of the Sub-cooling System on Multiple PACs. Next, validity of the proposed estimation method is confirmed using an experimental equipment in which the cooling capacity can be accurately measured. And then, the estimation accuracy is evaluated comparing with measurement and estimation capacities.

Sub-cooling System on Multiple PACs

Overview of the System: Figure 1 shows the schematic overview of Sub-cooling System on an Multiple PAC equipped with a function to exchange heat between the refrigerant and cooling water. As shown in Figure 1, this system has a simple equipment configuration consisting of a refrigerant sub-cool unit installed in the liquid refrigerant pipe line between the outdoor unit and the indoor units. The sub-cool unit consists of a plate-type heat exchanger and check valves. In this system, the high-temperature and high-pressure refrigerant that condensed in the outdoor unit is cooled to various temperatures by cooling water at various temperatures in the sub-cool unit. The low-temperature liquid refrigerant from the sub-cool unit is sent to the indoor units. The specific enthalpy difference in the indoor unit heat exchanger is increased in the proportion to the amount of removed heat in the sub-cool unit. Therefore, the

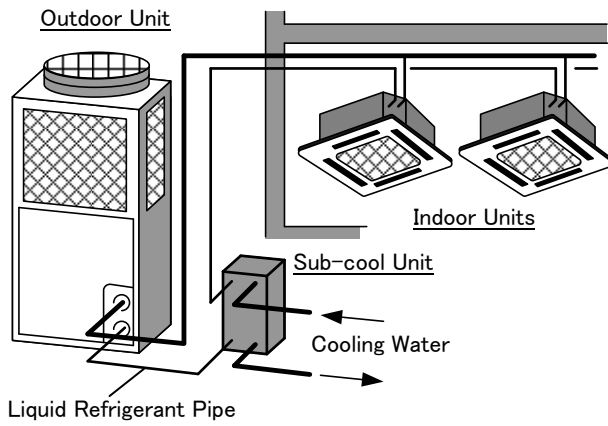


Figure 1

Overview of the Sub-cool System on Multiple PACs

refrigerant flow rate to achieve the required cooling capacity is reduced. Especially, in the case of the PACs equipped with a DC inverter compressor, which excels in partial load efficiency, the coefficient of performance (COP) would be improved significantly.

During the peak cooling load periods in the summer, cooling water ranging in temperature from 15°C to 40°C can be applied to sub-cool the refrigerant, because the refrigerant temperature as high as about 45°C in condensation process during such periods. Therefore, cooling tower water, boiler make-up water, well water, etc. can be used as the cooling water in sub-cool unit.

Effect of Energy Savings: As the effect of energy savings on performance test conducted on an air-cooled multiple PAC with sub-cooling operation. Figure 2 shows the relationship between the load factor and compressor power consumption ratio on sub-cooling operation and no sub-cooling operation, where symbols circle, square and triangle correspond results of the no sub-cooling, cooling water temperature $T_w=30^\circ\text{C}$ and $T_w=15^\circ\text{C}$. These operations were performed in accordance with JIS B8615-1. The outdoor temperature T_a and room temperature T_R were 35°C and 27°C , respectively. The load factor is defined as relative cooling capacity to the maximum cooling capacity during no sub-cooling operation. The compressor power consumption ratio is defined as relative compressor power consumption to the compressor power consumption during no sub-cooling operation with a load factor of 1. As shown in Figure 2, the compressor power consumption is reduced as lowering T_w , and such the effect of energy saving with sub-cooling is improved as increasing the load factor. When the load factor was 1, the compressor power consumption reductions of approximately 26% and 37% were achieved by sub-cooling at T_w of 30 and 15°C , respectively.

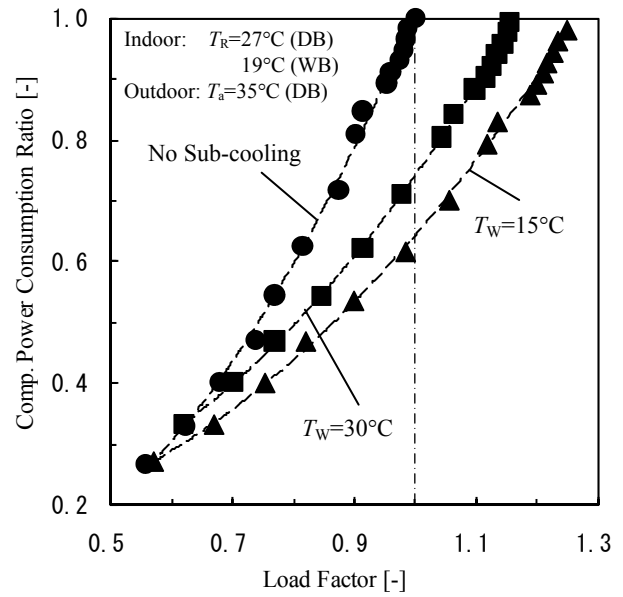


Figure 2

Effect of Energy Savings

Table 1
Refrigeration Cycle Status Changes

T_w [°C]	$\Delta h_{\text{sub}}/\Delta h_{\text{no-sub}}$ [-]	$G_{\text{sub}}/G_{\text{no-sub}}$ [-]	P_H [kPa]	P_L [kPa]
No sub-cooling	1.00	1.00	2998	948
30	1.22	0.82	2822	973
15	1.38	0.73	2734	975

Refrigeration cycle behavior: As the example of refrigeration cycle behavior, Table 1 shows specific enthalpy difference ratios in the indoor unit heat exchanger $\Delta h_{\text{sub}}/\Delta h_{\text{no-sub}}$ and refrigerant flow rate ratios $G_{\text{sub}}/G_{\text{no-sub}}$, condensing pressure P_H and evaporating pressure P_L . during no sub-cooling operation and sub-cooling operation with a load factor of 1. The $\Delta h_{\text{sub}}/\Delta h_{\text{no-sub}}$ and $G_{\text{sub}}/G_{\text{no-sub}}$ are defined, respectively, as relative Δh and the G to ones during no sub-cooling operation. As listed in Table 1, when T_w was 15°C , the Δh was expanded in 38% and the G was decreased in 27% compared with no sub-cooling operation, then the P_H considerably dropped from 2998kPa to 2734kPa and the P_L was raised a little.

Figures 3(a), 3(b) and 3(c) show the energy flow of the refrigeration cycle based on the performance test results. As shown in Figures 3, heat radiation rate in the outdoor unit heat exchanger was also reduced correspond to increasing the heat exchange rate in the sub-cool unit as lowering cooling water temperature. Thus, it was found that the compressor power consumption and heat radiation rate at the outdoor unit can be reduced significantly by adding the external cold heat into refrigeration cycle via the sub-cool unit.

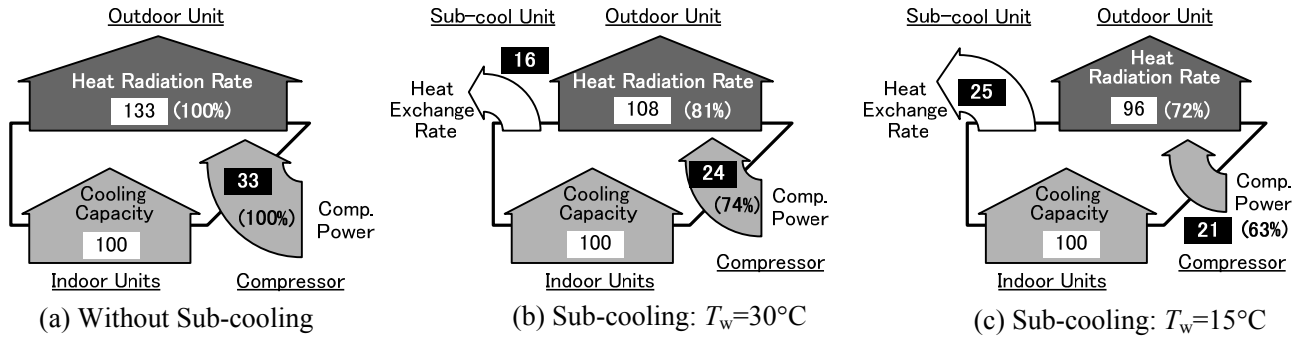


Figure 3
Energy Flow of Refrigerant Cycle on Sub-cooling System

Cooling Capacity Estimation Method for Inverter-Driven Packaged Air Conditioners

Cooling Capacity Estimation Equation: On an inverter-driven PAC under constant outdoor unit conditions (outside air conditions) and constant cooling capacity, as described above, adding the external cold heat to the liquid refrigerant pipe will change the condensing pressure, evaporating pressure, refrigerant flow rate and compressor power consumption in accordance with the amount of added cold heat.

Figures 4 show the refrigerant cycle behavior in pressure-enthalpy diagrams when the external cold heat $Q_{in(i)}$ is added to the liquid refrigerant pipe. Figure 4(a) and Figure 4(b) show the state of operation with not adding cold heat (normal operation) and adding the cold heat $Q_{in(i)}$, respectively. When the $Q_{in(i)}$ is added, specific enthalpy at the expansion valve inlet will be changed from $h_{3(0)}$ in Figure 4(a) to $h_{4(i)}$ in Figure 4(b). Also, the condensing pressure will be changed from $P_{H(0)}$ in Figure 4(a) to $P_{H(i)}$ in Figure 4(b), and the evaporating pressure will be changed from $P_{L(0)}$ in Figure 4(a) to $P_{L(i)}$ in Figure 4(b). Depending on these pressure changes, adiabatic compression coefficient will be changed from $\eta_{s(0)}$ to $\eta_{s(i)}$ and the specific enthalpy at the compressor outlet will be changed from $h_{1'(0)}$ in Figure 4(a) to $h_{1'(i)}$ in Figure 4(b). At the same time, the refrigerant flow rate will be changed from $G_{(0)}$ to $G_{(i)}$. As shown in Figure 4(a) where normal operation is performed, the specific enthalpy difference of the evaporator $\Delta h_{(0)}$ and the specific compression work $\Delta h_{cp(0)}$ are expressed by Equations (1) and (2), respectively.

$$\Delta h_{(0)} = h_{2(0)} - h_{3(0)} \quad (1), \quad \Delta h_{cp(0)} = h_{1'(0)} - h_{2(0)} \quad (2)$$

As shown in Figure 4(b), the cold heat $Q_{in(i)}$ is added to the liquid refrigerant pipe, the specific enthalpy difference $\Delta h_{in(i)}$ corresponding to the $Q_{in(i)}$, the specific enthalpy difference $\Delta h_{(i)}$ derived by subtracting $Q_{in(i)}$ from the cooling capacity $Q_{(0)}$ and the specific compression work $\Delta h_{cp(i)}$ are expressed by Equations (3), (4) and (5), respectively.

$$\Delta h_{in(i)} = h_{3(i)} - h_{4(i)} \quad (3), \quad \Delta h_{(i)} = h_{2(i)} - h_{3(i)} \quad (4)$$

$$\Delta h_{cp(i)} = h_{1'(i)} - h_{2(i)} \quad (5)$$

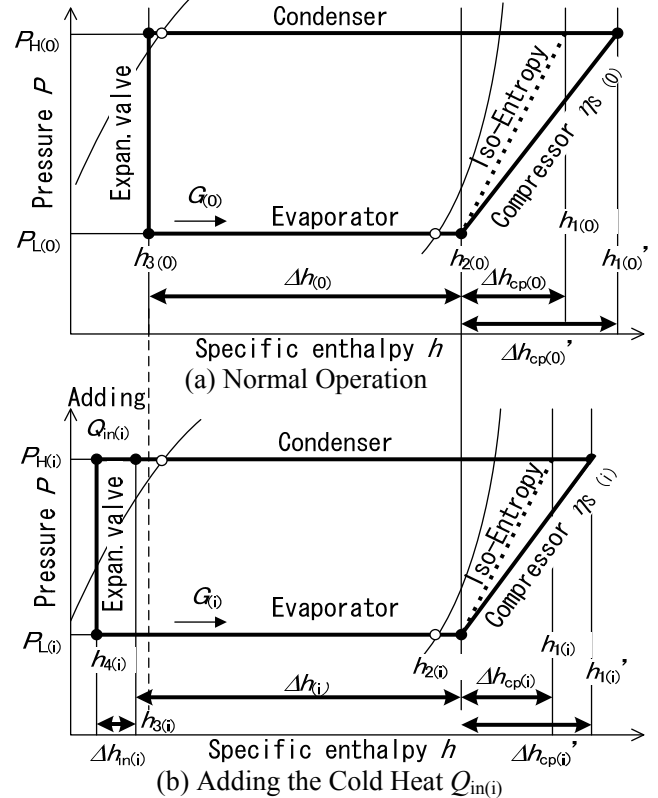


Figure 4
Refrigerant Cycle Behavior with Sub-cooling

The $\Delta h_{cp(0)}$ and the $\Delta h_{cp(i)}$ in Equations (2) and (5) are also expressed by Equations (6) and (7), respectively.

$\Delta h_{cp(0)} = \Delta h_{cp(0)} / \eta_{s(0)} \quad (6), \quad \Delta h_{cp(i)} = \Delta h_{cp(0)} / \eta_{s(i)} \quad (7)$ where $\Delta h_{cp(0)}$ and $\Delta h_{cp(i)}$ represent the adiabatic compression work in normal operation and in operation adding the cold heat $Q_{in(i)}$.

The compressor power consumptions $W_{(0)}$ in the normal operation and $W_{(i)}$ in operation adding cold heat $Q_{in(i)}$ are expressed by Equations (8) and (9), respectively.

$$W_{(0)} = G_{(0)} \Delta h_{cp(0)} = G_{(0)} \Delta h_{cp(0)} / \eta_T \quad (8)$$

$$W_{(i)} = G_{(i)} \Delta h_{cp(i)} = G_{(i)} \Delta h_{cp(i)} / \eta_T \quad (9)$$

where η_T represent the total compression efficiency of the inverter-driven compressor and is expressed by Equation (10).

$$\eta_T = \eta_s \eta_m \eta_M \eta_{INV} \quad (10)$$

where η_m , η_M and η_{INV} represent mechanical efficiency, motor efficiency and inverter efficiency, respectively.

The cooling capacity $Q_{(0)}$ in normal operation and operation adding the cold heat $Q_{in(i)}$ are expressed by Equations (11) and (12), respectively.

$$Q_{(0)} = G_{(0)} \Delta h_{(0)} \quad (11)$$

$$Q_{(0)} = Q_{(i)} + G_{(i)} \Delta h_{(i)} \quad (12)$$

Equations (8), (9), (11), and (12) can be consolidated into Equation (13).

$$Q_{(0)} = Q_{in(i)} / \{1 - (\eta_{T(i)} / \eta_{T(0)}) (\Delta h_{(i)} / \Delta h_{(0)}) (W_{(i)} / W_{(0)}) (\Delta h_{cp(0)} / \Delta h_{cp(i)})\} \quad (13)$$

The η_T is mainly influenced by the compression ratio $\varepsilon (= P_H / P_L)$ and the rotation speed. Assuming that η_T is the function only for the ε and that the function form is $a\varepsilon^b$, Equation (14) is derived.

$$\eta_{T(i)} / \eta_{T(0)} = (\varepsilon_{(i)} / \varepsilon_{(0)})^b \quad (14)$$

The $\Delta h_{(i)} / \Delta h_{(0)}$ varies depending on the heat exchange capability of the condenser when the cold heat $Q_{in(i)}$ is added. As shown in Figure 4, adding the cold heat $Q_{in(i)}$ lowers the condensing pressure, and decreases the temperature difference with the outside air or cooling water and condensing temperature, and reduces the heat exchange capability. The $\Delta h_{(i)} / \Delta h_{(0)}$ is assumed to be $\Delta T_{c(0)} / \Delta T_{c(i)}$ as change ratio of ΔT_c . The ΔT_c is a difference between the saturation temperature corresponding to the condensing pressure P_H and the outside air temperature T_a (temperature difference of condenser), the function form of which is determined as Equation (15).

$$\Delta h_{(i)} / \Delta h_{(0)} = (\Delta T_{c(0)} / \Delta T_{c(i)})^c \quad (15)$$

When Equations (14) and (15) substitute into Equation (13), Equation (16) is derived.

$$Q_{(0)} = Q_{in(i)} / \{1 - (\varepsilon_{(i)} / \varepsilon_{(0)})^b (\Delta T_{c(0)} / \Delta T_{c(i)})^c (W_{(i)} / W_{(0)}) (\Delta h_{cp(0)} / \Delta h_{cp(i)})\} \quad (16)$$

The point to be checked about this estimation method is the adequacy of the function forms for the total compression efficiency and the change ratio of the specific enthalpy difference expressed as Equations (14) and (15).

Measured Items and Estimation Method: Figure 5 shows the schematic diagram of measured items. For the cooling capacity estimation using the proposed method, the $W_{(i)}$, the $P_{H(i)}$ at the condenser outlet (or compressor outlet), the $P_{L(i)}$ and refrigerant temperature $T_{s(i)}$ at the compressor inlet and the outside air temperature $T_{a(i)}$ are measured in each cases of the $Q_{in(i)}$ is not added ($i=0$) and is added to the liquid refrigerant pipe. The $Q_{in(i)}$ is added by supplying heat source water through a detachable jacket or equivalent installed on the liquid refrigerant pipe. A compact cold or hot water supply is available for the heat source. Added the $Q_{in(i)}$ can be calculated

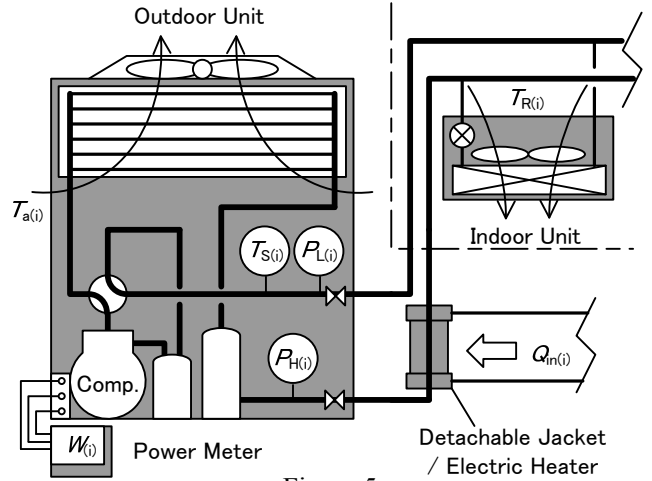


Figure 5
Measured Items

from the flow rate of the heat source water and the temperature difference at the inlet and outlet of the jacket. A hot heat can be added using an electric heater, in which case of added the $Q_{in(i)}$ is derived from the power consumption of the electric heater. The refrigerant pressure is measured using pressure gauge installed at the service valve, refrigerant charging port, etc. The refrigerant temperature is measured using a thin-film temperature sensor affixed on the surface of the refrigerant pipe well-insulated.

Using Equation (16), $\varepsilon_{(i)}$ and $\varepsilon_{(0)}$ can be calculated from the measured $P_{L(0)}$, $P_{L(i)}$, $P_{H(0)}$ and $P_{H(i)}$. $\Delta T_{c(0)}$ and $\Delta T_{c(i)}$ can be calculated from the saturation temperature derived from the physical property database for refrigerants⁴⁾ using measured $P_{H(0)}$, $P_{H(i)}$, $T_{a(0)}$ and $T_{a(i)}$. The $\Delta h_{cp(0)}$ and the $\Delta h_{cp(i)}$ can be derived from the physical property database for refrigerants⁴⁾ using the measured $T_{s(0)}$, $T_{s(i)}$, $P_{L(0)}$, $P_{L(i)}$, $P_{H(0)}$ and $P_{H(i)}$. Consequently, there are three unknown values in Equation (16): the $Q_{(0)}$ to be derived and the exponents b and c . If the above measurements are made representing in the cases of operation with no adding $Q_{in(i)}$ and operations with adding three or more $Q_{in(i)}$ ($i=1$ to 3 or more), three or more simultaneous Equations (16) can be derived. The $Q_{(0)}$ can be estimated by determining the most probable values for exponents b and c that satisfy these conditions. The prerequisite for this estimation method, the constant outside air condition and cooling load, can be ensured by measuring the room temperature $T_{R(i)}$ and checking that $T_{R(i)}$ and $T_{a(i)}$ do not vary during the measurement time.

Evaluation of Estimation Method Using Test Equipment

Test Equipment and Conditions: Figure 6 shows a schematic overview of test equipment to validate the proposed estimation method. The test equipment

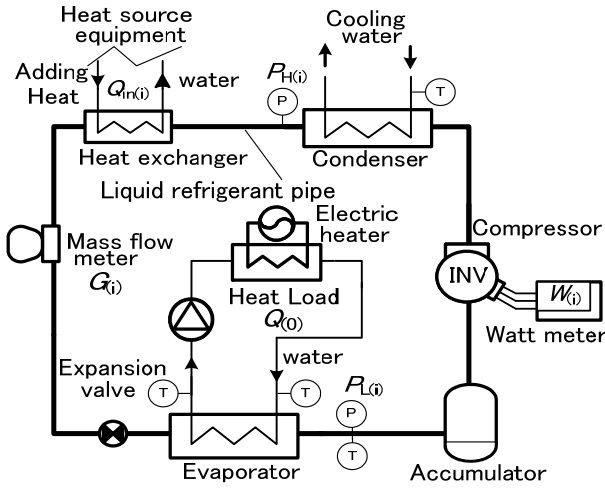


Figure 6

Schematic Diagram of Test Equipment for Validation

Table 2

Test Conditions to Evaluate the Proposed Method

Case	Cooling Capacity $Q_{(0)}$	Added Heat $Q_{in(i)}$		
	[kW]	i	[kW]	Cold / Hot
1	2.84	14	0.18~0.62	Cold
2	2.21	9	0.13~0.36	Cold
3	1.57	8	0.10~0.24	Cold
4	2.24	7	0.71~0.95	Hot
5	1.48	7	0.70~0.96	Hot
6	0.79	7	0.40~0.51	Hot

consists of an inverter-drive compressor (rotary type), a water-cooled condenser, a heat exchanger to add external heat, an electronic expansion valve, an evaporator, an accumulator and an electric heater as heat load. The electronic expansion valve controls and keeps constant the degree of refrigerant superheat at the evaporator outlet. The compressor rotation speed controls and keeps constant the cooled water temperature at the evaporator outlet. The electric heater controls and keeps constant the cooled water temperature at the evaporator inlet.

The compressor power consumption $W_{(i)}$ which including the inverter loss was measured using a wattmeter (accuracy: ± 0.2 , F.S.) at the inverter primary side. The refrigerant flow rate $G_{(i)}$ was measured using a mass flow meter (accuracy: ± 0.5 , F.S.). The condensing pressure $P_{H(i)}$ and the evaporating pressure $P_{L(i)}$ are measured using precision pressure gauges (accuracy: ± 0.25 , F.S.). The refrigerant and cooled water temperatures at the evaporator inlet and outlet, cooling water temperature at the condenser inlet and outlet, and heat source water temperature at the heat exchanger for adding heat inlet and outlet were measured using sheath-type platinum resistance temperature detectors (Pt 100 Ω ,

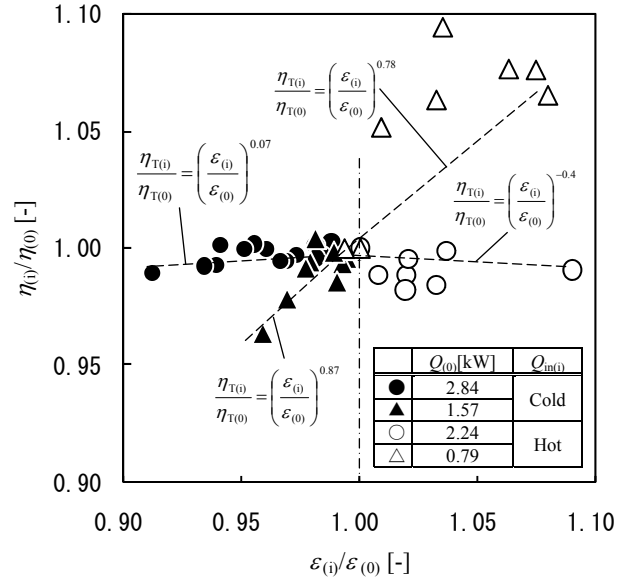


Figure 7

Relationship between $\varepsilon_i/\varepsilon_0$ and η_i/η_0

Class A) inserted into the pipe. The accuracies of the cooling capacity $Q_{(0)}$ and the added heat $Q_{in(i)}$ are ± 36 W and ± 15 W, respectively.

Table 2 shows the test conditions. R-410A was utilized as the refrigerant. The test was performed for cases where adding various cold and hot heat $Q_{in(i)}$ under the several cooling capacity $Q_{(0)}$ listed in Table 2 keeping constant the degree of refrigerant superheat at the evaporator outlet (approximately 10°C). And the evaporating temperature was 0°C during operation adding the $Q_{in(i)}$. In each test condition, the $W_{(i)}$, the $G_{(i)}$, temperatures and pressures at various positions were measured after the refrigeration system reached a steady state.

Validation of Estimation Equations: Function form of the total compression efficiency [Equation (14)]:

Figure 7 shows the relationship between the $\varepsilon_i/\varepsilon_0$ and the $\eta_{T(i)}/\eta_{T(0)}$ in the case of adding $Q_{in(i)}$ under the test conditions of the case 1, 3, 4 and 6 as listed in Table 2. Close and open symbols represent the results of adding the cold heat $Q_{in(i)}$ and the hot heat $Q_{in(i)}$ respectively, where symbols close circle, close triangle, open circle and open triangle correspond results of the $Q_{(0)} = 2.84, 1.57, 2.24$ and 0.79 kW. As shown in Figure 7, it is found that $\varepsilon_i/\varepsilon_0$ and $\eta_{T(i)}/\eta_{T(0)}$ have a good correlation for each $Q_{(0)}$ values in each cases of adding the cold heat and adding the hot heat, respectively. Since the proposed estimation method is to estimate a cooling capacity individually under the specified cooling conditions, there is no need to identify a function that can express the change characteristics of the compressor efficiency in a wide range of cooling capacity. Therefore, function form of

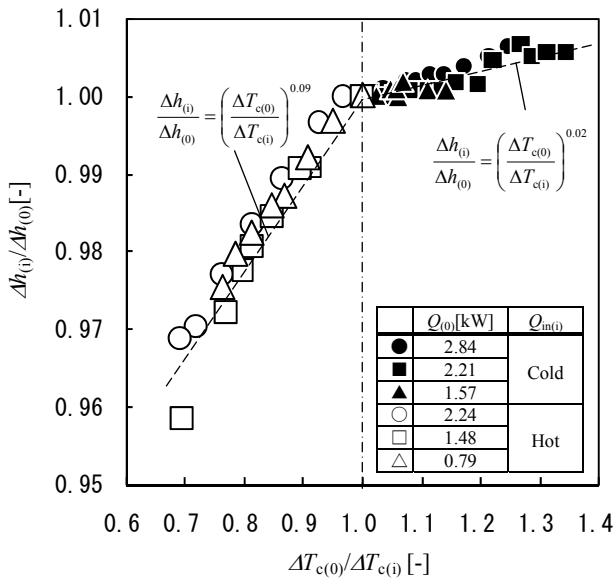


Figure 8

Relationship between $\Delta T_{c(0)}/\Delta T_{c(i)}$ and $\Delta h_{(i)}/\Delta h_{(0)}$

the total compression efficiency in this estimation method can be established as a function only for the compression ratio as shown by Equation (14).

Function form of the change ratio of specific enthalpy difference in the condenser [Equation (15)]: Figure 8 shows the relationship between the $\Delta T_{c(0)}/\Delta T_{c(i)}$ and $\Delta h_{(i)}/\Delta h_{(0)}$ in the case of adding $Q_{in(i)}$ under all test conditions listed in Table 2, where close and open symbols represent the results of adding the cold heat $Q_{in(i)}$ and the hot heat $Q_{in(i)}$, and symbols close circle, close triangle, close square, open circle, open triangle, open square correspond results of the $Q_{(0)}=2.84, 2.21, 1.57, 2.24, 1.48$ and 0.79 kW, respectively. As shown in Figure 8, it is found that the correlation with $\Delta h_{(i)}/\Delta h_{(0)}$ and $\Delta T_{c(0)}/\Delta T_{c(i)}$ was the same regardless of the cooling capacity $Q_{(0)}$ whether cold heat or hot heat was added.

As described above, it is confirmed that the adequacy of function forms for the total compression efficiency and the change ratio of the specific enthalpy difference expressed as Equations (14) and (15).

Evaluation of Estimation Accuracy: As the result of cooling capacity estimation of the test equipment using proposed estimation method, Table 3 shows the measured values (true values) and the estimated values of $Q_{(0)}$, estimation accuracies, and exponents b and c . Values of b and c are obtained through calculation shown in Figures 7 and 8. As shown in Table 3, the estimation accuracies of $Q_{(0)}$ are +7.5 or less when the cold heat $Q_{in(i)}$ is added and are +4 or less when the hot heat $Q_{in(i)}$ is added. Thus, it is found that a cooling capacity can be estimated with a sufficient accuracy whether cold heat or hot heat is

Table 3
Cooling Capacity Estimation Results and Accuracy

Case	Cooling Capacity $Q_{(0)}$		Accuracy [%]	Exponents	
	True [kW]	Estimation [kW]		b [-]	c [-]
1	2.84	2.89	101.7	0.07	0.02
2	2.21	2.38	107.5	0.19	0.02
3	1.57	1.63	103.8	0.87	0.02
4	2.24	2.23	99.5	-0.40	0.09
5	1.48	1.54	104.0	-0.03	0.09
6	0.79	0.81	102.0	0.78	0.09

added to the liquid refrigerant pipe if these exponents can be adequately identified for the function of the total compression efficiency and a change ratio of the specific enthalpy difference in the condenser that defined in the proposed estimation method. As shown by the exponent c in Table 3, the fact that c is a constant value in both cases of cold heat and hot heat addition is useful for laborsaving in determining the exponent c from measurement data in the actual capacity estimation on the PACs facility.

CONCLUSIONS

Based on the concept of the Sub-cooling System on Multiple PACs, we propose a simple cooling capacity estimation method for inverter-driven PACs. We confirmed the following by evaluating the validity of proposed method and estimation accuracy using the test equipment.

- 1) The estimation equations defined in proposed method is adequacy for the refrigeration cycle behavior.
- 2) The validity of the proposed estimation method was confirmed as results of estimation accuracy of $\pm 8\%$.

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