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Study of In Situ Monitoring Method for (Cooling and Heating) Capacity of Variable Refrigerant Flow (VRF) Multi-Split Air Conditioners for Commercial Buildings

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Abstract

The in situ monitoring method for variable refrigerant flow (VRF) multi-split air conditioning and heat pump equipment (VRF system) has not yet been established, and actual capacity and efficiency (performance) cannot be sufficiently understood. A manufacturer understands actual performance by means of the compressor curve (CC) method. This method, however, cannot be applied to all models of all manufacturers because details of the method are not disclosed.
The refrigerant enthalpy method is expected to become a common monitoring method. In order to measure refrigerant flow rate, it is necessary to cut the refrigerant piping, but this is avoided, so as not to inconvenience customers. Recently, refrigerant flow rate measurement has become possible without cutting the piping by using a clamp-on ultrasonic flowmeter, however the accuracy has not been verified.
In this study, an in situ monitoring instrument is installed in the VRF system; the instrument includes an ultrasonic flow meter (UFM), a Coriolis flow meter (CFM), 12 thermocouples and an absolute pressure sensor. The mutual verification between the UFM and the CFM shows that an acceptably accurate flow rate is measured by the ultrasonic flow meter, though a constant error margin is measured. In addition, three kinds of estimated capacity are compared: (1) capacity from the CC method, (2) capacity from monitoring, and (3) capacity from flow rate correlation. The capacity from the correlation agrees with that from the CC method with an accuracy of +/-20%.

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Keywords: VRF, Commercial Building; In Situ Measurements; ultrasonic flow meter; Coriolis flow meter;

1. Introduction

No in situ monitoring method for variable refrigerant flow (VRF) multi-split air conditioning and heat pump equipment (VRF system) has yet been established and the actual capacity and efficiency (performance) cannot be sufficiently understood [1-5]. A manufacturer understands actual performance using the compressor curve (CC) method. This method, however, cannot be applied to all models from all manufacturers because the details are undisclosed [1].

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The refrigerant enthalpy method is expected to become commonplace. Although measuring the refrigerant flow rate means cutting the refrigerant piping, this is best avoided, so as not to inconvenience customers. Recently, it has become possible to measure the refrigerant flow rate without cutting the piping; using a clamp-on ultrasonic flow meter, but the accuracy has not been verified.

In this study, an in situ monitoring instrument is installed in the VRF system; including an ultrasonic flow meter (UFM), a Coriolis flow meter (CFM), 12 thermocouples and an absolute pressure sensor. Using this instrument, mutual verification between the UFM and CFM and an estimated capacity comparison are conducted.

2. In Situ Monitoring Instrument Installed in the VRF System

A performance monitoring system was installed into the VRF system that set up in the TEPCO Research Institute (in Kanagawa prefecture, Japan). This performance monitoring system can calculate capacity (for cooling and heating), power consumption and coefficient of performance (COP) using measurement values, such as temperature, pressure, flow rate, power consumption and so on. The installation configuration and a schematic diagram of the performance monitoring system are shown in Figure 1. The specification of the VRF system and the list of measurement points are shown in Tables 1 and 2, respectively.

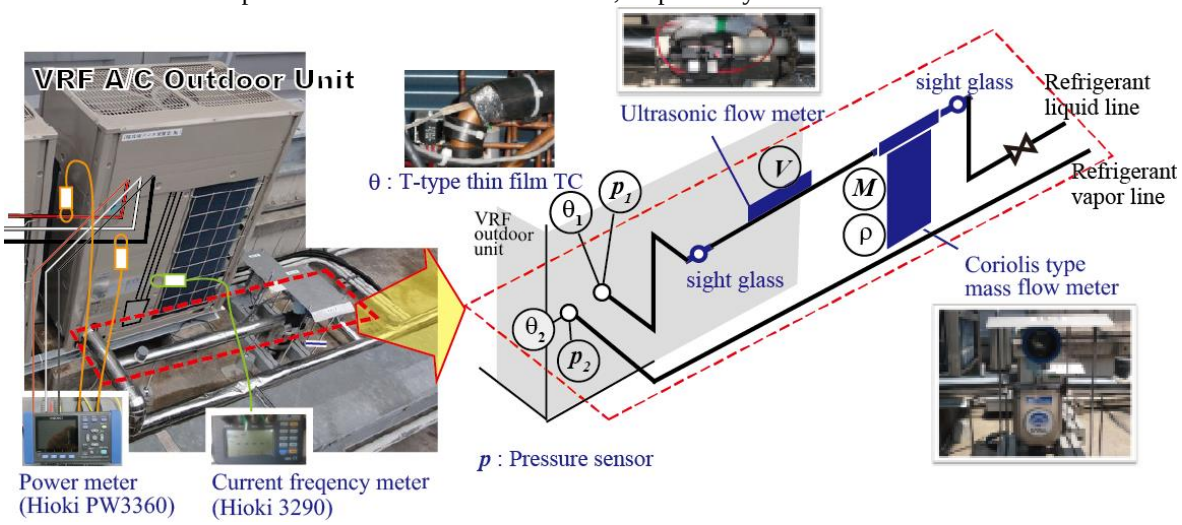


Fig. 1 Installation configuration of sensors to the VRF A/C outdoor unit to be measured

Table 1 Specifications of the VRF outdoor unit

Instrument	Outdoor Unit	Indoor Unit
Number of Units	1	3
Power supply	Three-phase 200V, 50/60Hz	Single-phase 200V, 50/60Hz
Rated cooling capacity [kW/Unit]	28.0/28.0	11.2
Rated cooling power consumption [kW/Unit]	8.58	0.165/0.194
Rated heating capacity [kW/Unit]	31.5/31.5	12.5
Rated heating power consumption [kW/Unit]	8.34	0.132/0.161
Max. low-temperature heating capacity [kW/Unit]	26.7/26.7	—
Refrigerant	R410A	

Table 2 Measurement sensor list

Sensors	Symbol	
T-type thermocouple	θ_1	Surface temperature of liquid piping of outdoor unit [°C]
	θ_2	Surface temperature of vapor piping of outdoor unit [°C]
Pressure sensor	p_1	[High pressure] Outdoor unit liquid line pressure [MPa]
	p_2	[Low pressure] Outdoor unit vapour line pressure [MPa]
Coriolis-type flow meter	M and ρ	Mass flow rate [kg/min] and density [kg/L] of liquid refrigerant
Ultrasonic-type flow meter	V	Volumetric flow rate of refrigerant liquid [L/min]
Power meter	P	Electric power consumption [kW]

Current frequency meter	f_{inv}	Current frequency after inverter [Hz]
T-type thermocouple	θ_a	Ambient temperature[ºC]
Hygrometer	R_a	Ambient humidity [%]

Refrigerant pressure sensors were installed into refrigerant charge ports at the liquid line and vapour line of the VRF system. Here, p_1 and p_2 represent the liquid-line and vapour-line pressures, respectively. In cooling, p_1 represents high pressure and p_2 represents low pressure. Conversely, when heating, both p_1 and p_2 represent high pressure. In this case, low pressure must be estimated using another method. Because this VRF system has the third refrigerant charge port near the outlet of the outdoor expansion valve (in case of heating), low pressure in heating could be measured.

The refrigerant piping surface temperature was substituted for the refrigerant temperature. Here, θ_1 and θ_2 represent the liquid-line and vapour-line temperatures, respectively.

The liquid refrigerant volumetric flow rate, V, was measured by an ultrasonic type flow meter (UFM). The UFM had a clamp-on ultrasonic sensor unit (photograph in Figure 1) and display unit. It is easy to setup, because there is no need to cut the refrigerant piping.

To verify the UFM flow rate, the liquid refrigerant mass flow rate, M, was also measured by CFM. In this study, the refrigerant liquid line piping was cut to install the CFM.

To measure the power consumption of the VRF system, a digital power meter (HIOKI PW 3360) was installed, while to measure the current frequency after the inverter, a current frequency meter (HIOKI 3290) was installed.

All the sensor outputs were connected to a data logger (Yokogawa MW100) and accumulated into a PC. An internal monitoring box (IMB) was also installed and all data was stored into a PC. The data obtained from the IMB was used to estimate heating and cooling capacities.

3. Mutual Verification of Measurement Data

In this section, measurement values obtained from two or more measuring instruments were compared as well as measurement values obtained from the IMB and other measurement equipment, whereupon the difference and tendency of measurement values were verified mutually.

3.1. Mutual verification of refrigerant volumetric flow rate measurement

The refrigerant liquid volumetric flow rate based on measurement by the CFM, V_c, was calculated using equation (1).

$$V_c = M \,/\, \rho \tag{1}$$

Here, M and ρ represent the mass flow rate and density of the refrigerant, respectively.

Conversely, V means the volumetric flow rate of refrigerant obtained by the UFM. A time-series volumetric flow rate comparison between the CFM and UFM is shown in Figure 2. When an error signal occurred on the UFM, the output was set to become negative. The reason for the disorder in the UFM volumetric flow rate is the frequent error signal. The UFM cannot measure the two-phase flow rate, namely, because a small bubble would disturb the measurement. Observing from a sight glass that was located at the UFM inlet reveals that a small bubble was found in the refrigerant liquid immediately after the start, but disappeared after 5 minutes' operation. Error signals, however, occurred frequently since that time. It has emerged that error signals occurred when the refrigerant flow rate changes according to the change in operation status of the VRF system and that the UFM can only measure the flow rate when it is kept nearly constant. Moreover, it has also emerged that the volumetric flow rate by the UFM was smaller than that by the CFM without the error signals.

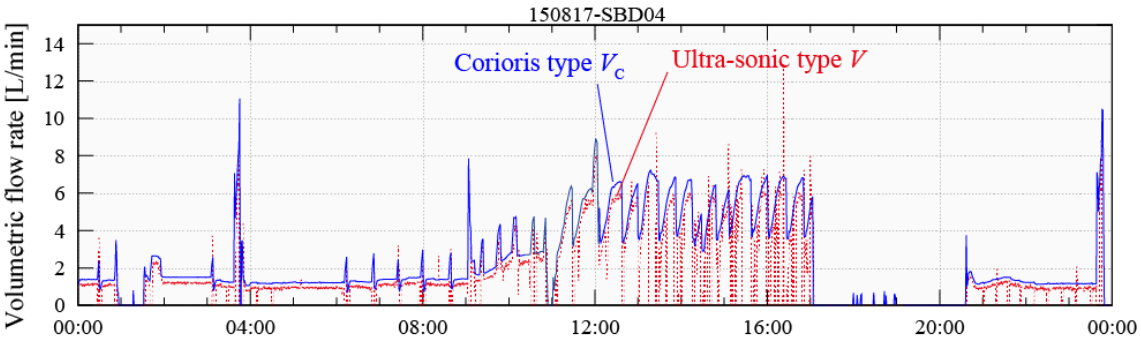


Fig. 2 Time-series volumetric flow rate comparison between Coriolis-type and ultrasonic-type

A comparison between the volumetric flow rate by the UFM and that by the CFM is shown in Figure 3. The displayed data shows one-minute values for a seven-day cooling period; excluding data for the time when the error occurred. The x-axis is the volumetric flow rate calculated from the measurement value by the CFM and the y-axis is that by the UFM.

It clearly emerges that the volumetric flow rate by the UFM is about 10% lower than that by the CFM. Although mutual verification between the UFM and CFM shows that an acceptably accurate flow rate is measured by the UFM, a constant error margin is measured and further verification, such as reproducibility on another site, is necessary.

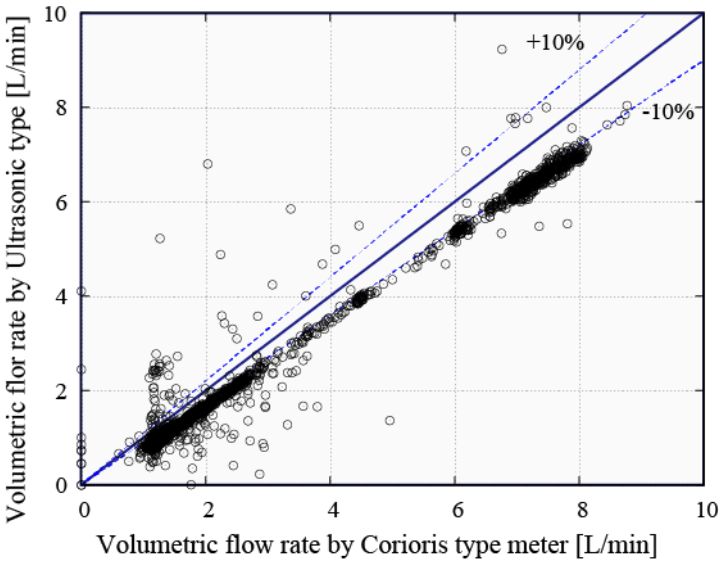


Fig. 3 Volumetric flow rate comparison between Coriolis-type and ultrasonic type

3.2. Mutual verification between rotating speed of the compressor and electricity current frequency after inverter.

The compressor rotating speed obtained by the IMB and one-third of the current frequency after the inverter are compared in Figure 4. Clearly, a strong positive correlation between them and the rotating speed of the compressor can be estimated by measuring the current frequency after the inverter.

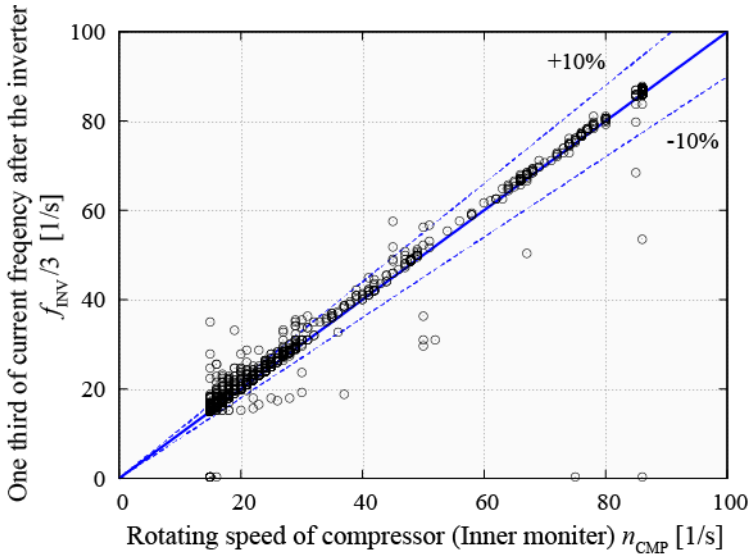


Fig. 4 Relation between the rotating speed of the compressor and the electricity current frequency after the inverter

4. Verification of the Simplified Measurement Method

4.1. Formation of correlation equation for the refrigerant flow rate

To obtain the estimation of the refrigerant flow rate, a correlation equation between the flow rate and the rotating speed of the compressor was formed. Equation (2) is well known as the relation between the suction flow rate into the compressor and the rotating speed of the compressor.

$$M_{suc} = \eta_{vol}\,\rho_{suc}\,v_{cmp}\,n_{cmp} \tag{2}$$

Here, M_{suc} is the mass flow rate into the suction of the compressor, η_{vol} is the volumetric efficiency of the compressor, ρ_{suc} is the refrigerant vapour density at suction of the compressor, v_{cmp} is the compression room volume and n_{cmp} is the rotating speed of the compressor. Next, equation (2) is transformed using volumetric flow rate (V_{suc}), whereupon equation (3) is obtained.

$$V_{suc} = \eta_{vol}\,v_{cmp}\,n_{cmp} \tag{3}$$

In equation (3), v_{cmp} is a peculiar constant of a compressor, η_{vol} multiplied by v_{cmp} is considered ξ and equation (4) is obtained.

$$V_{suc} = \xi\ n_{cmp} \tag{4}$$

The correlation equation will be formed by calculating ξ by the measurement values. Here, the unit of ξ is a volume. In equation (4), the volumetric flow rate (V_{suc}) was calculated using the mass flow rate by the CFM. The volumetric flow rate (V_{suc}) was calculated using the following equation:

$$V_{suc} = M\ /\ \rho_{suc} \tag{5}$$

The rotating speed of the compressor was obtained by the IMB and the relation between the rotating speed and the volumetric flow rate calculated using equation (5) is shown in Figure 5. The volumetric flow rate and the rotating speed are considered proportional and the result of the linear function fitting ($f(x) = a\,x$) is the following equation:

$$V_{sucE1} = f(n_{cmp}) = 2.9614\,n_{cmp} \tag{6}$$

The flow rate and cooling capacity calculated using equation (6) are identified by subscript "E1". The fixed coefficient in equation (6) and $=g(\pi)$, that is described later in equation (7), are not only the inherent value of

this VRF system, but also the inherent value of this installation. It is thought that it is not available when different indoor unit is used, when length of piping is different, when elevation between indoor unit and outdoor unit is different, and so on, even if it is the same outdoor unit would be used.

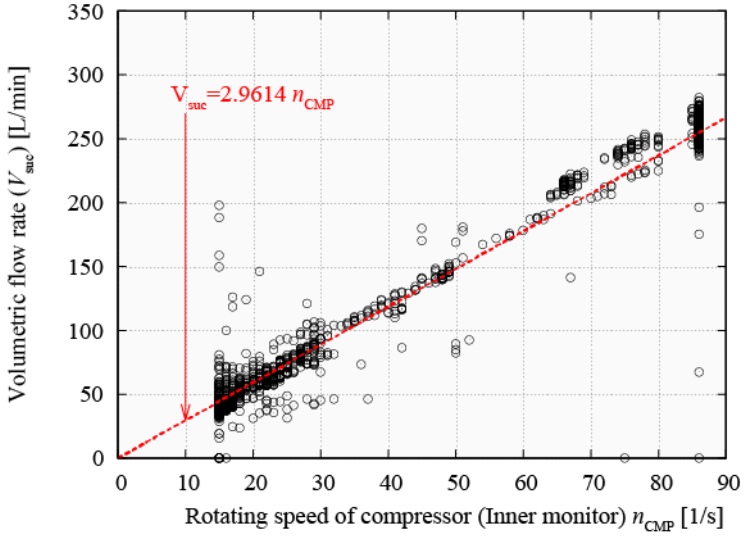


Fig. 5 Relation between the volumetric flow rate of the compressor suction line and the rotating speed

In addition, the case where ξ is a function of π was also studied. The relation between ξ ($= \eta_{vol} \times v_{cmp}$) and π ($= p_H / p_L$) is shown in Figure 6. Only data where n_{cmp} > 30 was displayed on the graph and it can be considered as a quadratic function with a maximum value peaking at around $\pi = 2.5$. Applying to the quadratic function using the minimum square method, the following correlation equation was obtained:

$$\xi = g(\pi) = -0.268871\pi^2 + 1.30359\,\pi + 1.49957 \tag{7}$$

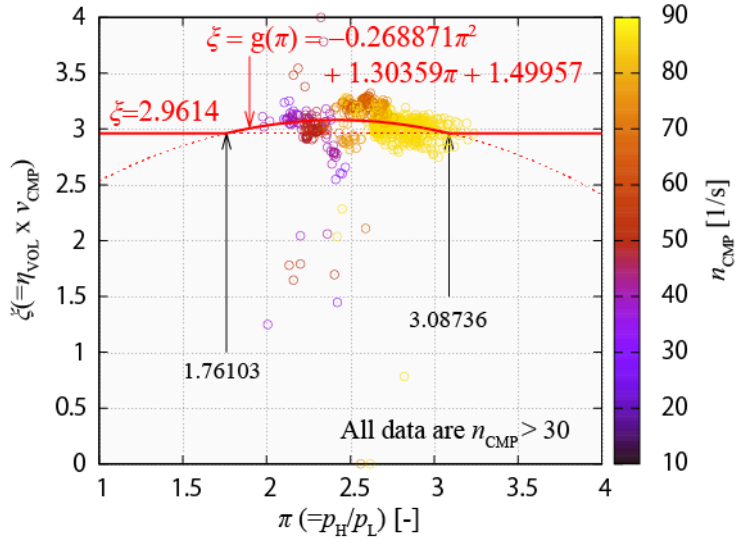


Fig. 6 Relation between the volumetric efficiency of the compressor and the compression ratio

The flow rate and cooling capacity calculated using equation (7) are identified by subscript “E2”.

It is thought that the reason why there are many dispersed measurement point in Figure 3 - Figure 6 is that non-steady state measure points are represented in these figures.

4.2. Estimation of cooling capacity

Using equations (6) and (7), the cooling capacity was estimated.

The cooling capacity of the VRF system can be estimated using the specific enthalpy difference between the inlet and outlet of the outdoor unit and the refrigerant mass flow rate. The liquid specific enthalpy (h_1), which is

the outlet specific enthalpy from the outdoor unit, can be estimated using the liquid line pressure (p_1) and the liquid line temperature (θ_1). The vapour specific enthalpy (h_2), namely the inlet specific enthalpy into the outdoor unit, can be estimated using the vapour line pressure (p_2) and the vapour line temperature (θ_2). To estimate specific enthalpy by pressure and temperature, the NIST Refprop V9 program was used.

The suction vapour volumetric flow rate can be estimated using equation (6) and the liquid line refrigerant flow rate was calculated using the following equation:

$$M_{E1} = \rho \times V_{suc,E1} \tag{8}$$

Here, M_{E1} is the liquid line mass flow rate, ρ is the liquid line refrigerant density and $V_{\mathrm{suc,E1}}$ is the volumetric flow rate calculated by equation (6).

The cooling capacity can be estimated using the following equation:

$$Q_{c,E1} = M_{E1}\,(h_2 - h_1) \tag{9}$$

Conversely, another liquid line mass flow rate, M_{E2}, can be estimated using equation (7) and the following equation:

$$M_{E2} = \rho\, V_{suc,E2} = g(\pi) \times \rho \times n_{cmp} \tag{10}$$

The cooling capacity can be estimated using the following equation:

$$Q_{c,E2} = M_{E2}\,(h_2 - h_1) \tag{11}$$

4.3. Mutual verification of cooling capacity by the refrigerant flow rate estimation method

Four kinds of estimated capacity are compared:

(1) Cooling capacity by the CC method (was obtained from the manufacturer),

(2) Cooling capacity based on the measured mass flow rate by the CFM, $Q_{c,\mathrm{RE}}$

$$Q_{c,RE} = M(h_2 - h_1) \tag{12}$$

(3) Cooling capacity estimated by M_{E1} and

(4) Cooling capacity estimated by M_{E2}.

The cooling capacity by the CC method and estimated cooling capacities are compared in Figure 7. The x-axis shows the cooling capacity by the CC method, while the y-axis shows the estimated cooling capacity estimated using the other three methods.

The cooling capacity estimated with the CC method and that by the measurement correlate to within $\pm$ 10%. Though cooling capacities based on the estimated mass flow rate using equations (8) and (10) deviate by over 10%, it is thought the method can be used to understand the in situ capacity of the VRF system.

$\left|Q_{c,\mathrm{CC}} - Q_{c,\mathrm{E2}}\right|$ is almost equal to $\left|Q_{c,\mathrm{CC}} - Q_{c,\mathrm{E1}}\right|$, or rather larger than $\left|Q_{c,\mathrm{CC}} - Q_{c,\mathrm{E1}}\right|$. It is shown that the effect of volumetric efficiency correction by the compression ratio is limited. It is necessary to verify the effect of the variable volumetric efficiency by other measurement results and long-term, continuous measurement results.

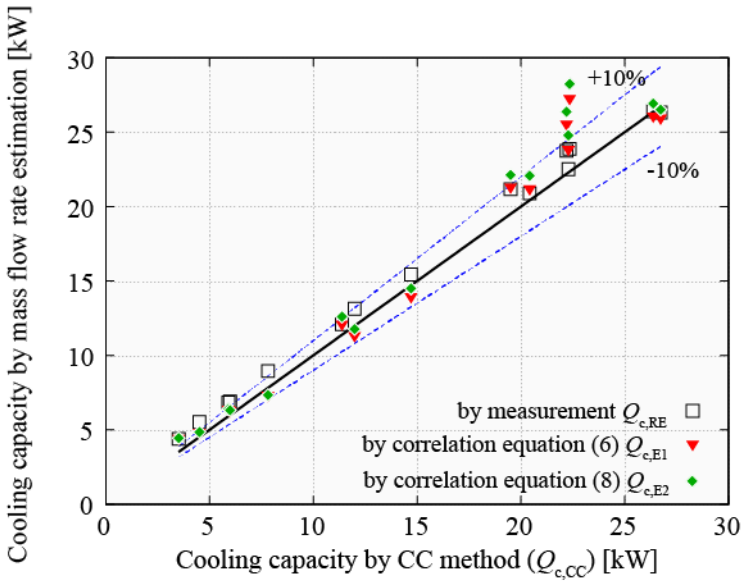


Fig. 7 Comparison by method to estimate cooling capacity

5. Conclusion

The following results were achieved by this study:

- The UFM can measure the volumetric flow rate if the refrigerant is not a two-phase flow.
- Though the measurement value by the UFM is about 10% smaller than that by the CFM, the value seems to give an indication of the refrigerant flow rate.
- It emerges that the UFM is an effective measuring instrument to determine the in situ status of the VRF system.
- Some challenges remain to be solved;
    - This result is merely a measurement example in one site.
    - Good examples must be accumulated.
    - Though the performance monitoring system established in this study was easier to install than a conventional system, a simpler system is required for more penetration.
    - To simplify the introduction of IoT technology, handy sensor usage, energy harvesting technology and so on are requested.

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