PERFORMANCE EVALUATION OF VRF SYSTEMS -1ST REPORT EXPERIMENTAL EVALUATION OF STEADY STATE DRIVING-

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Abstract: VRF (Variable Refrigerant Flow) system is the air conditioning system that has some indoor units. This system is widely used for office buildings because it can be easily equipped with those buildings, and save space and cost. So far, the air-conditioning system is mainly evaluated at the rated point using COP but recently it is demanded to improve the annual efficiency of the air conditioning system. For VRF systems, it is difficult to evaluate their performance because the indoor units are driven separately in the different indoor conditions, and the refrigerant flow and control method are very complex. We built a performance test facility for VRF system, and studied the steady state performance of the VRF system that has four indoor units, experimentally. The rated cooling capacity of the VRF system is 28.0. This experiment was carried out, based on the JIS standard. Detailed part load efficiency of the system was clarified by the experiment.

Key Words: VRF system, COP, APF, multi unit

1 INTRODUCTION

In Japan, two problems are pointed out on VRF (Variable Refrigerant Flow) system these days.
First, rated capacity does not always represent system performance from a practical standpoint. Rating capacity and its measuring method are defined as JIS B 8615-1 or JIS B 8615-2[1] [2] in Japan. These standards require similar conditions to those of ISO 5151 or 13253 (Table-1). In Japan, outdoor temperature varies greatly throughout the year. Current VRF system has inverter controlled variable speed compressor for performance increasing. However, the compressor condition is not considered in the ISO. Therefore, rated capacity does not equal the performance of on-sight VRF system.
Japanese Industrial Standards, JIS 8616:2006 and JRA4048:2006 provide the APF (Annual Performance Factor) and half capacity under the rating capacity condition to deal with those circumstances. APF is a calculated value based on driving pattern, rated capacity and half capacity given by JIS. This driving pattern is based on two main assumptions. One is that load factor is in proportion to outdoor temperature. The other is that degradation coefficient is flat at the part load factor.
Second, there are no system performance evaluation methods for VRF system practically. In general, indoor air-enthalpy method (AE method), refrigerant-enthalpy method (RE method), and compressor curve method (CC method) are known. AE method can evaluate the system performance with a high level of accuracy but the installation of big apparatus is impossible. RE method, at the practical operation, costs too much. In addition, refrigerant becomes liquid-gas phase mixture in piping and it is said that measuring such refrigerant flow is not an

accurate method. CC method is the method that calculates refrigerant flow by the compressor characteristic and some VRF system built-in sensors. CC method can evaluate the system performance with a high level of accuracy. But the compressor curve characteristic is not public.

Table 1: Capacity test condition

Parameter	**ISO 5151 ISO 13253 condition T1**	**JIS B 8615-1 JIS B 8615-2 condition T1**
Cooling capacity		
Temperature of air entering indoor side (^oC) dry-bulb wet-bulb	 27 19	 27 19
Temperature of air entering outdoor side (^oC) dry-bulb wet-bulb	 35 24	 35 24
Heating capacity		
Temperature of air entering indoor side (^oC) dry-bulb wet-bulb	 20 15	 20 15
Temperature of air entering outdoor side (^oC) dry-bulb wet-bulb	 7 6	 7 6

Thus, there are not a rating capacity and a performance evaluation method that matches the VRF system performance at the practical operation.

For such a background, comparing the rated capacity with the evaluation of some methods at the practical operation is not appropriate. Author's purpose is to evaluate accurate performance of VRF system from a practical standpoint. Therefore, Author reports a basic experiment for evaluation of VRF system steady state driving.

This report describes whether there is reproducibility in the rated performance, and discusses the difference between the thermal load test and the rated performance.

2 SYSTEM

Author tested a VRF system that consists of one outdoor machine and four indoor units. Two compressors are installed in the outdoor machine, which contains a heat exchanger for sub-cooling. The indoor units are ceiling mounted cassette type. Each unit has an electric expansion valve for adjusting to thermal load (Figure 1). Table.2 shows the published specification of the system.

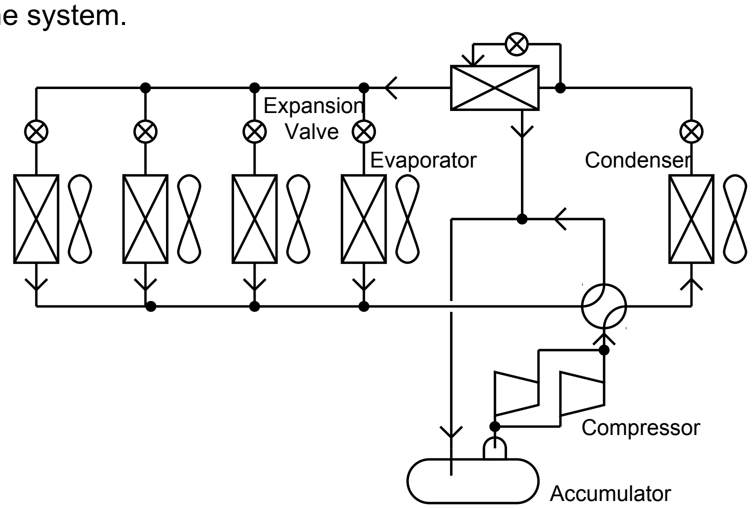

Figure 1: Refrigerant circuit arrangement (Cooling)

Table 2: Published specification of the system (60Hz) under rating capacity condition

Appellation	Specification
Indoor unit ×4	Rated blast volume：$22m^3$/min ×4
	Fan：Turbofan ×4
Outdoor unit	Compressor：Closed scroll type ×2
	Rated blast volume：$233m^3$/min
	Fan：Propeller fan
Refrigerator system	Rated cooling output：28.0kW
	Cooling electric input：7.19kW
	Rated heating output：31.5kW
	Heating electric input：8.21kW
	Refrigerant：R410A
Compressor control	Variable Speed by Inverter Constant Speed
Indoor unit×4	Rated blast volume：$22m^3$/min ×4

3 TESTING APPARATUS AND METHODS

Figure 2 shows the test room arrangement. The test room consists of two rooms (the outdoor machine room and the indoor unit room). Each room has some instruments and apparatus for tunnel air-enthalpy method. Both rooms have the testing capacity mode and the testing thermal load performance mode. The testing capacity mode tests the object driven at the steady state with outside control, and room conditioning apparatus control the temperature of the testing rooms, following ISO and JIS. The testing thermal load performance mode tests the object driven without outside control, and the room conditioning apparatus control the thermal load of the testing rooms, which is not regulated by ISO and JIS. The control method of room conditioning apparatus is the same at each mode.
At cooling operation, latent cooling load is controlled by a motor valve on steam piping, and the sensible cooling load is controlled by an electric heater. At heating operation, the heating load is controlled by brine pump flow.
It was recognized that the accuracy of these testing rooms were between -1.96% to +2.29% on testing capacity of 5HP and 8HP defined by JRAIA (The Japan Refrigeration and Air Conditioning Industry Association).

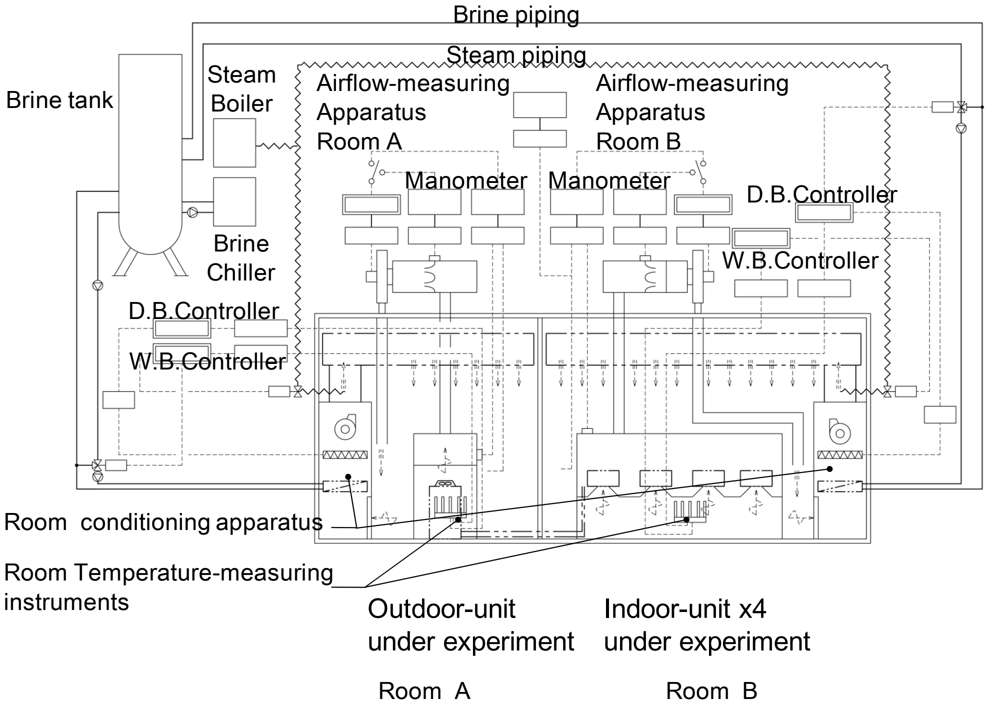

Figure 2: Experimental Room Arrangement

Table 3: Test room control

Appellation	Specification
Room	7.9 m (W) ×6.8 m (D) × 6.3 m (H)
Accuracy of control	Temperature : ±0.1 ^{o}C
	Relative humidity : ±2%
Undulation of parameter	Temperature : ±0.5 ^{o}C
	Relative humidity : ±5%

Table 4 shows the condition of the rating capacity experiment. JIS 8616 provides marking half capacity for variable compressor speed type. The tests were operated by the testing capacity mode.

Table 4: Rating capacity condition

Parameter	**C1 Rated Cooling Capacity**	**C2 Half Cooling Capacity**	**H1 Rated Heating Capacity**	**H2 Half Heating Capacity**
Temperature of air entering indoor side (^{o}C)				
dry-bulb	27	27	20	20
wet-bulb	19	19	15	15
Temperature of air entering outdoor side (^{o}C)				
dry-bulb	35	35	7	7
wet-bulb	24	24	6	6
Compressor Control	Steady	Steady	Steady	Steady

Table 5 shows the condition of the testing thermal load experiment. The tests were operated by the testing thermal load performance mode.

Table 5: Testing thermal load condition

Parameter	**C4 100% Cooling Load**	**C5 50% Cooling Load**	**H4 100% Heating Load**	**H5 50% Heating Load**
Temperature of air entering indoor side (°C)				
dry-bulb	27	27	20	20
wet-bulb	19	19	15	15
Temperature of air entering outdoor side (°C)				
dry-bulb	35	35	7	7
wet-bulb	24	24	6	6
Compressor Control	Transient	Transient	Transient	Transient

Table 6 shows the condition of the additional experiment. JIS 8616 does not provide marking quarter capacity for the variable compressor speed type.

Table 6: Additional condition

Parameter	**C3 Quarter Cooling Capacity**	**C6 25% Cooling Load**	**H3 Quarter Heating Capacity**	**H6 25% Heating Load**
Temperature of air entering indoor side (°C)				
dry-bulb	27	27	20	20
wet-bulb	19	19	15	15
Temperature of air entering outdoor side (°C)				
dry-bulb	35	35	7	7
wet-bulb	24	24	6	6
Compressor Control	Steady	Transient	Steady	Transient

4 RESULTS AND DISCUSSION

4.1 RESULTS OF RATING CAPACITY EXPERIMENT

Table 7 shows the temperature and compressor input at the rating capacity experiment. The values of heat exchanger surface and refrigerant temperature are same

Comparing with C1/H1 and C2/H2, the constant speed compressor stopped at half capacity operation (C2/H2). And the temperature of the heat exchanger surface rises corresponding to half cooling capacity.
Figure 3 shows COPj at the tests. COPj is derived from eq. (1)-(4). Then, Kr is the piping length coefficient that considers the piping heat loss. COPj was steady at each test.

Table 7: Temperature and compressor input

Measuring Point	**C1 Rated Cooling Capacity**	**C2 Half Cooling Capacity**	**H1 Rated Heating Capacity**	**H2 Half Heating Capacity**
Temperature of air entering indoor side (°C)				
dry-bulb	27.0	27.0	20.0	20.0
wet-bulb	19.1	19.1	14.5	14.5
Temperature of air entering outdoor side (°C)				
dry-bulb	34.8	34.9	7.0	7.0
wet-bulb	24.0	24.0	6.0	6.0
Temperature of heat exchanger surface				

indoor side (°C) Unit Average	14.8	18.7	34.2	27.1
System input (kW)	7.78	2.70	8.62	3.29
Compressor ON/OFF Variable Speed by Inverter Constant Speed	 ON ON	 ON OFF	 ON ON	 ON OFF

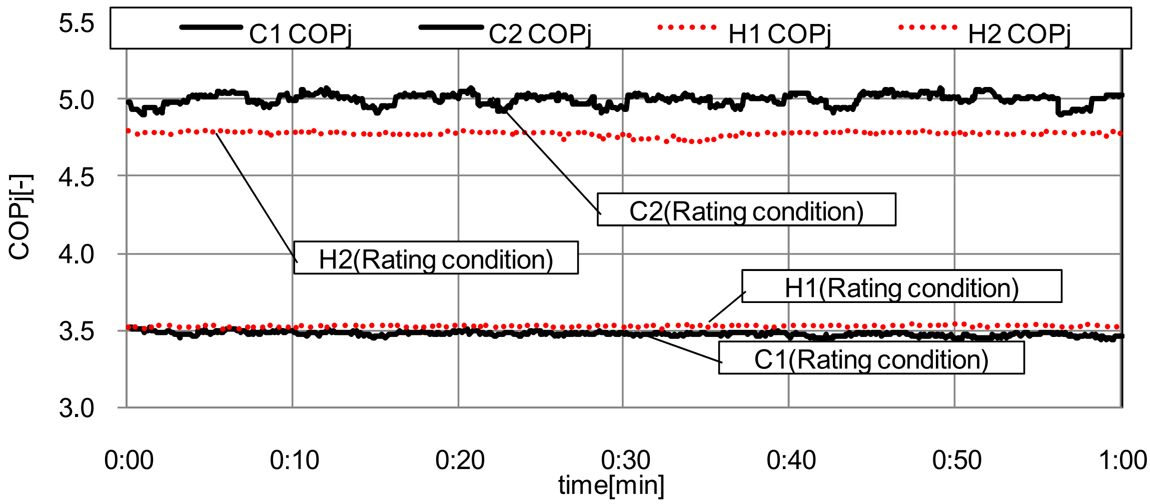

Figure 3: COPj

$$\Phi_{tcij} = \frac{q_{\mathrm{mi}}\left(h_{\mathrm{a1}} - h_{\mathrm{a2}}\right)}{v'_{\mathrm{n}}\left(1 + w_{\mathrm{n}}\right)} \quad (1)$$

$$P_{tj} = P_{ij} + P_{oj} \quad (2)$$

$$K_r = func\left(l_{pipe}, h_{pipe}\right) \quad (3)$$

$$COP_{\mathrm{j}} = \frac{\Phi_{tcij}}{P_t \cdot K_r} \quad (4)$$

Nomenclature

Φ_{tij} : total capacity at experiment, W
ha : air enthalpy, J・kg^{-1}・K^{-1}
qmi : air volume, m3/s
v' n : specific volume, m3/kg
wn : absolute humidity, kg/kg(DA)
Pi : indoor unit electric power, W
Po : outdoor unit electric power, W

lpipe : piping length, m
hpipe : piping height difference, m
K r(c) : correction factor for piping at cooling, 0.969
K r(h) : correction factor for piping at heating, 0.989
COP_j : coefficient of performance at experiment, [-]
COP_r : coefficient of performance at publication, [-]

Figure 4 shows COPj and COPr. COPr is a calculated value of published specification of the system. At C1, COPj was 7.59% lower than COPr. At C2, COPj was 2.91% higher than COPr. At H1, COPj was 4.18% lower than COPr. At H2, COPj was 4.18% higher than COPr.
Each COPj is higher than 3.5 and the difference of COPr was within +/-10%.

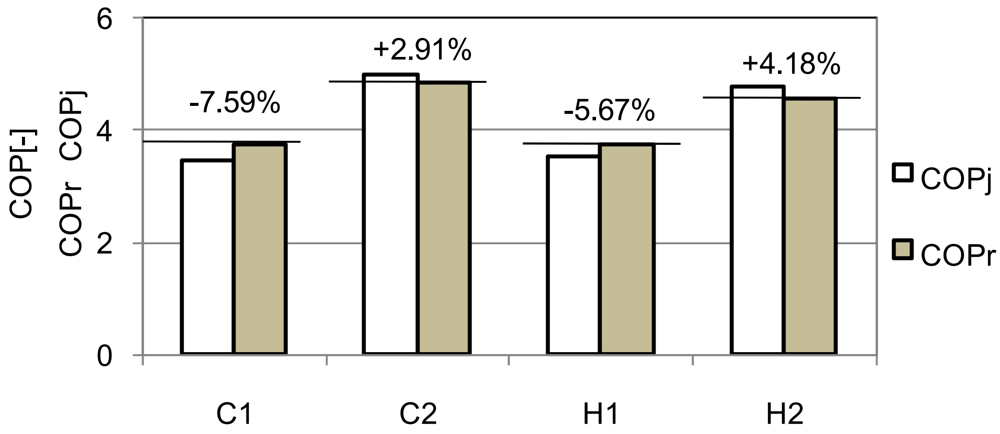

Figure 4: COPj and COPr

From this result, there is reproducibility in rated performance under the rating capacity condition.

4.2 RESULTS OF THERMAL LOAD EXPERIMENT

Figure 5 shows the amount of total indoor unit heat exchange. The record interval was 5 seconds. The case of C3/C6 was the additional condition (Table 6). In C3, the total heat exchange on these indoor units was steady under the rating condition. In C4, heat exchange was steady. But in C5 and C6, heat exchange was unsteady.

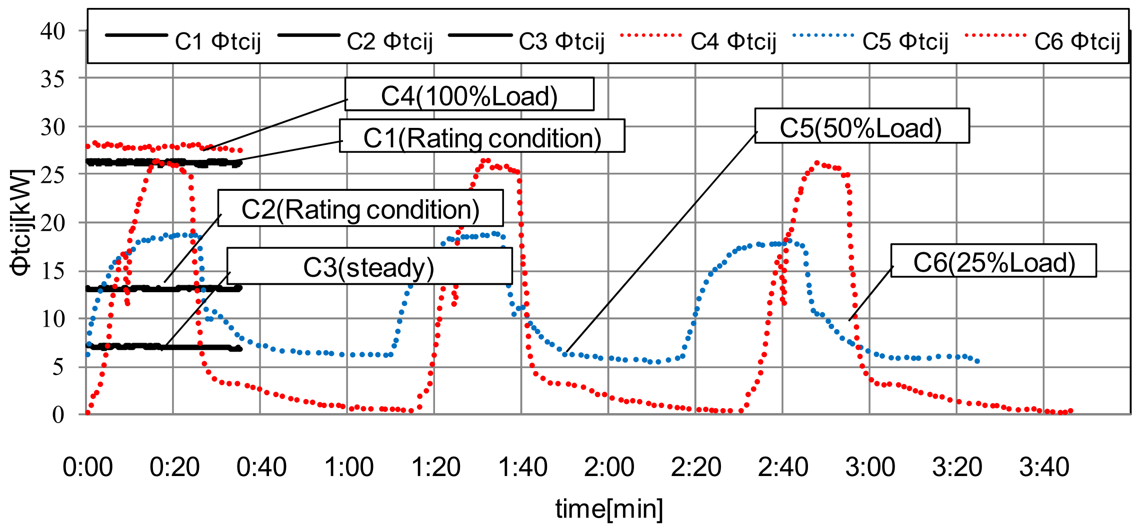

Figure 5: Total heat exchange on indoor units (cooling)

Figure 6 shows the heat exchange on these indoor units. The record interval was 5 seconds. The case of H3/H6 was the additional condition (Table 6). In H4, the total heat exchange on indoor unit was steady under the rating condition. In H4, heat exchange was steady. But in H5 and H6, heat exchange was unsteady.

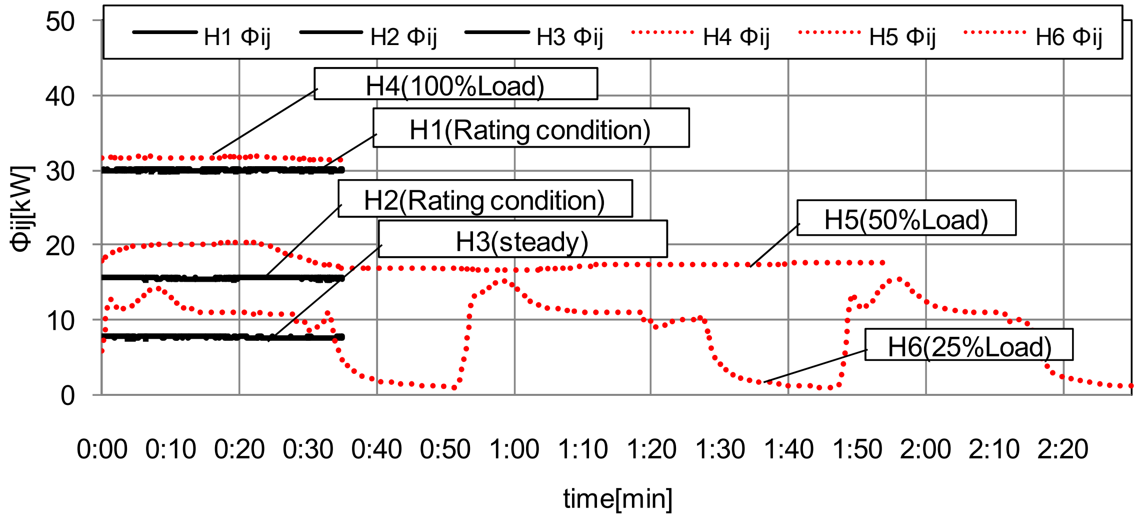

Figure 6: Total heat exchange on indoor units (heating)

From this result, heat exchange was unsteady under the low testing thermal load condition. According to Figure 5 and Figure 6, heat exchange was unsteady under the low testing thermal load condition.

Figure 7 shows the system input histograms of each test. Under the rating capacity condition, the system input was steady. However, the system did not correspond when driven under the thermal load to that of under the rating capacity. And the constant speed compressor stopped at the half and quarter capacity tests.

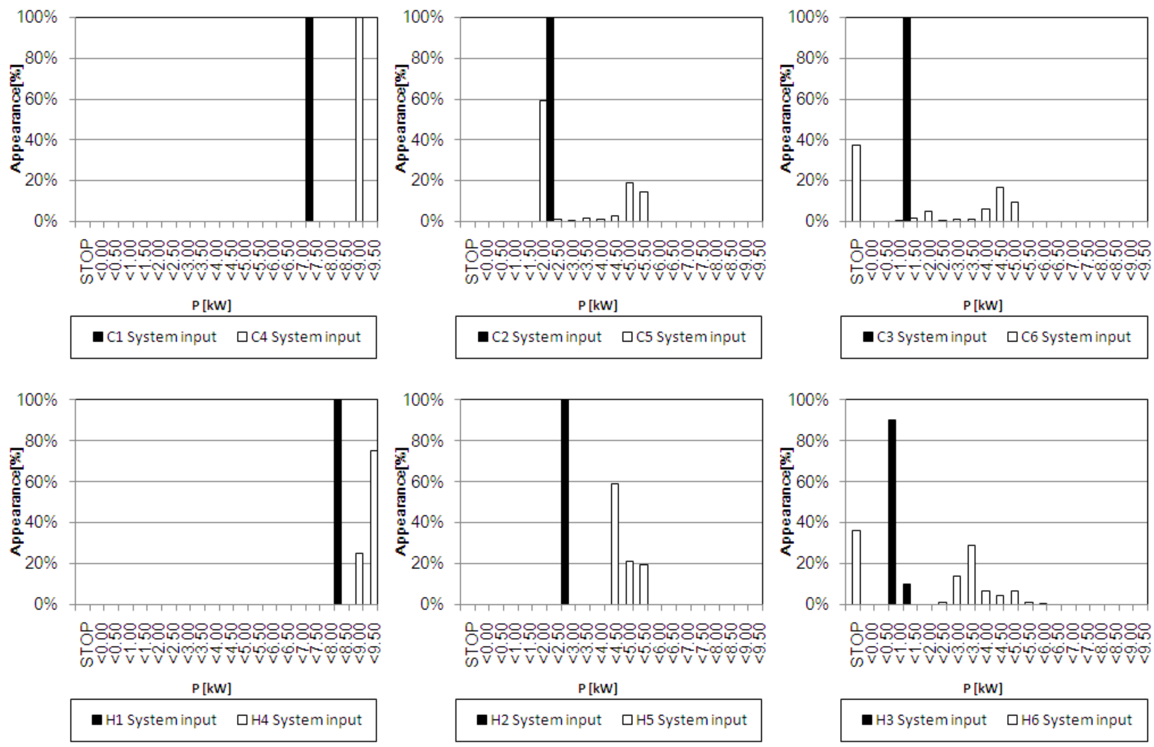

Figure 7: System input histograms

Figure 8 shows the system performance characteristic based on this experiment. Effect of the additional condition is not included, because the compressor was driven on-off. Under the testing thermal load, the system input was more than that of under the rating capacity. COP under the testing thermal load was lower than that of under the rating capacity.

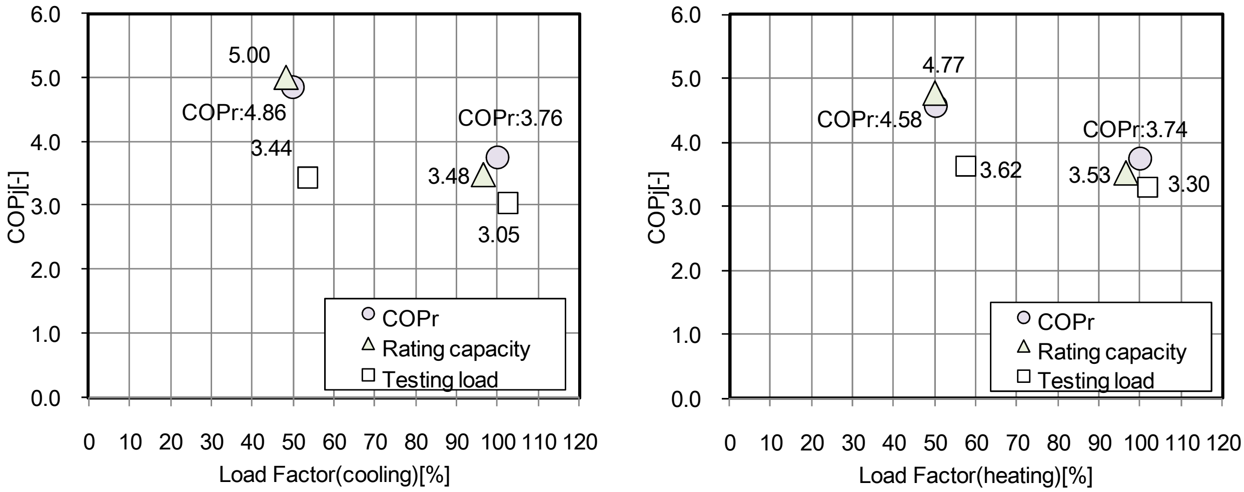

Figure 8: Results of the system performance

5 CONCLUSIONS

We studied the evaluation method of VRF system. As a result of the experiment based on JIS standards, the difference between the load test and the rated performance is +/-10% (form -7.59% to 2.91%). Thus, the rated performance has reproducibility when the conditions of suction air at the outdoor machine and the indoor units are satisfied, and when the compressor is driven constant.

Comparing JIS test and the thermal load test, the electricity of the system input increased at the thermal load test. Therefore, it can be said that there is a running control difference between the rated performance and thermal load test. Moreover, the compressors were driven transient at partial load test. That was caused by the variation of thermal processing value. This variation needs to be considered in control logic of VRF system.

6 REFERENCES

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