PERFORMANCE EVALUATION OF VRF SYSTEMS USING COMPRESSOR CURVE METHOD

Mugito KIKUCHI,Mr., Graduate School of Fundamental Science and Engineering, Waseda University, Shinjuku-ku, Tokyo, 169-8555, Japan;
Keisuke OHNO, Research associate, School of Fundamental Science and Engineering, Waseda University, Shinjuku-ku, Tokyo, 169-8555, Japan;
Kiyoshi SAITO, Professor, School of Fundamental Science and Engineering, Waseda University, Shinjuku-ku, Tokyo, 169-8555, Japan;

Abstract: VRF (Variable Refrigerant Flow) system is air conditioning system that has some indoor units. This system has advantage for saving space and cost but it is more difficult to evaluate its performance in actual driving. The larger system tends to be selected beyond necessity and it can't exercise its potential energy efficiency.
To solve this problem, the actual used data should be feed backed to design development phase. There are some methods to measure system's performance such as air enthalpy method, however, it is difficult to get data under actual usage. So impractical and accurate measuring method is strongly demanded.
We are focusing on the compressor curve method that predicts the performance of air conditioner from the compressor performance. This method can predict refrigerant flow rate with minimum measuring instruments. The purpose of this study is to establish compressor curve method for VRF system and evaluate its validity.
As a result, we confirmed that the performance of VRF system could be predicted using compressor curve method from comparison to other measuring methods.

Key Words: compressor curve method, performance evaluation, VRF system

1 INTRODUCTION

VRF system is now rapidly being accepted because of its capability to reduce space and cost. This is because compared to single type refrigerators, they can drive many indoor units with just one outdoor unit. We can stop or drive indoor units each with different loads. But because the outdoor unit is designed to mainly cover situations either when all indoor units are at full load or otherwise, when one particular indoor unit is at full load, whether the system is driving at its full efficiency is not clear. To obtain optimum designing for air conditioning systems, estimations of device performances is vital and furthermore, to save energy, it is important to consider efficient arrangements and driving for devices.

So in this research, we focused on the compressor curve method (C.C.). This is a method that can easily and accurately, estimate air conditioning performances with compressor characteristics and a few measurement data. We considered using this method on our VRF system.

If this is possible, device manufacturers can develop devices with higher efficiency by getting feed back from it. And equipment designers can choose devices with optimum performances and thus, driving time with low load, low efficiency is reduced and uplifting of year-round efficiency is expected. Furthermore, for example by the visualisation of driving performance, users can momentarily see and re-examine their way of driving the system and thus, save energy.

For estimating air conditioning performances, the air enthalpy method (A.E.) that is used by JIS's evaluation tests, is widely being recognised as an accurate method. But this method is far too expensive and big so that it is impossible to use it during actual driving.
So, by comparing results from the reliable A.E. and our C.C., we will make it clear that C.C. is a valid method to estimate air conditioning performances.

2 COMPRESSOR CURVE METHOD

C.C. is the advanced method of the refrigerant enthalpy method that is stated in JIS B8516-2, and it uses compressor characteristics to calculate refrigerant mass flow rate. Unlike methods such as A.E. that need large and expensive equipment like air ducts, this method can calculate performance from refrigerant pipe side and thus, small amount of censors are needed so measurements per devises are easy.
We will explain about the calculation method below using a common heat pump system as an example. Fig.1 shows the refrigerant flow and Ph diagram during refrigerating drive and Fig.2 shows that of during heating drive. Normally, compressors loaded in a refrigerator have a compressor characteristic for each one. So C.C., we will calculate refrigerant mass flow rate using these characteristics. Although there are many ways to write down calculating equations, it is common to calculate the flow rate from compressor inlet pressure, outlet pressure and rotational speed.

$$G = func\left(freq, P_1, P_6, T_6\right) \tag{1}$$

We can obtain air conditioning performance by multiplying the calculated refrigerant mass flow rate and the specific enthalpy to and fro the indoor heat exchanger.

$$Q_{cool} = G \cdot \left(h_5 - h_4\right) \tag{2}$$

$$Q_{heat} = G \cdot \left(h_2 - h_3\right) \tag{3}$$

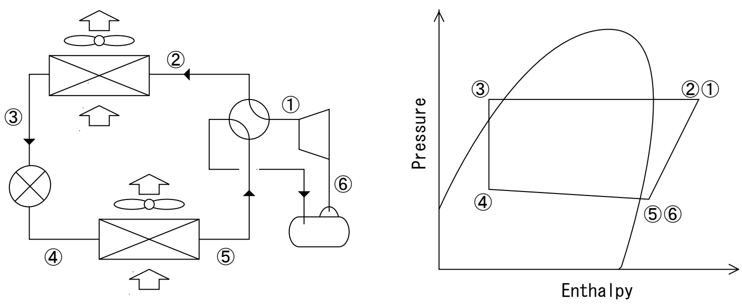

Figure 1: Example System (Cooling)

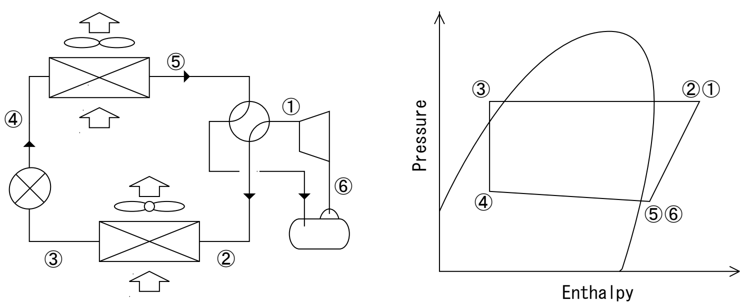

Figure 2: Example System (Heating)

Now because specific enthalpy under single-phase conditions can be calculated from the temperature and pressure of each point, we can obtain performance from inside the system with minimum measuring devices. Meanwhile, for point h4 where it becomes saturated, we adopted conditions from point h3 where it is sub cooled and thus measurable.

3 TARGET SYSTEM

The system we are considering is a VRF which consists of one outdoor unit, 4 indoor units and two compressors. The compressor outlet refrigerant that passed through the two

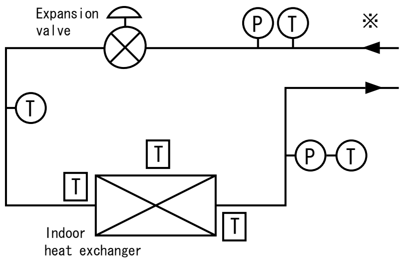

Figure 3: Indoor Unit x4 (Cooling)

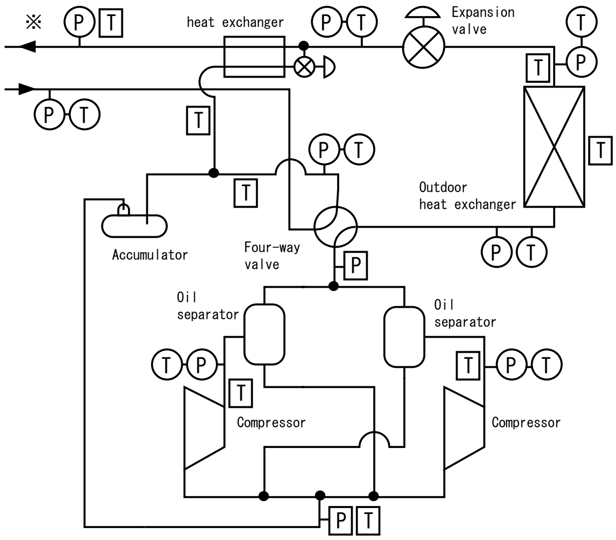

Figure 4: Outdoor Unit (Cooling)

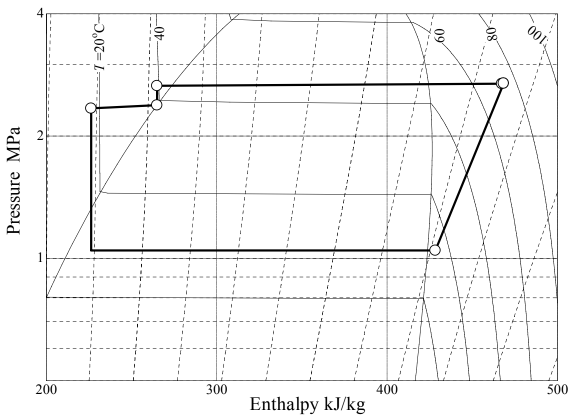

Figure 5: Ph-diagram (Cooling)

compressors with oil tempering paths joins together at the condenser. Then after being depressurised by the first expansion valve, a part of refrigerant passes another valve and so it sub cools the refrigerant of the main cycle at the sub cooling heat exchanger.
Fig. 3 and 4 show the refrigerant flow and measurement points of the system. We will measure pressure and temperature to and fro elements such as compressors, heat exchangers and expansion valves. And also, the data of compressors and fans from the system main board is obtained. The sensor enclosed in square is a set in sensors that controls the device, and the one enclosed in circle is the measuring instrument we newly adopted. Fig.5 shows the Ph diagram for rated cooling condition of this system.
We will divide the test room into indoor unit side and outdoor unit side, and set them at each environment. With electrical heaters, brine and boilers, we can set temperatures with margins ranging at ±0.1℃ and humidity at ±2%. And for outlet air for each unit, we adopt ±2% to collect and measure flow rates. The test room and flow of air is shown in Fig.6.

4 CALCULATION PROCESS OF COMPRESSOR CURVE METHOD

Taking our equipment for example, we will explain specifically about the compressor curve

Table 1: Specifications

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

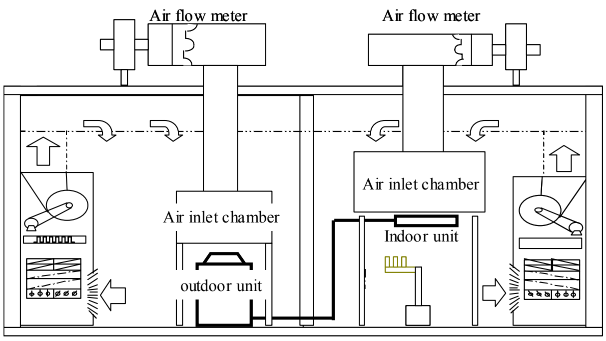

Figure 6: Airflow

Table 2: Test room control

appellation	specification
Room	7.9 m (W) ×6.8 m (D) × 6.3 m (H)
Accuracy of control	Temperature:±0.1℃
	Rerative humidity :±2%
Undulation of parameter	Temperature:±0.5℃
	Rerative humidity :±5%

method.
The characteristics of the compressor set in the device, can be given as a function of 3 parameters. Pressures to and fro the compressor, suction temperature and compressor frequency. Because there are 2 compressors set inside our system, we added the flow rates obtained from each one back and forth.

$$G = func\left(freq, P_{com\_I}, P_{com\_O}, T_{com\_I}\right) \tag{4}$$

The performance can be calculated from subtracting heat loss of fans from the product between the pressure difference of each heat exchanger and the obtained refrigerant flow rate. As for our apparatus, which has 4 indoor units, we adopted values of refrigerants before separation and after interflowing. For refrigerant property we used REFPROP.

$$Q_{cool} = G \cdot \left(h_{eva\_O} - h_{eva\_I}\right) - Q_{fan} \tag{5}$$

$$Q_{heat} = G \cdot \left(h_{con\_I} - h_{con\_O}\right) + Q_{fan} \tag{6}$$

$$h_{eva\_O} = func\left(P_{eva\_O}, T_{eva\_O}\right) \tag{7}$$

$$h_{eva\_I} = func\left(P_{con\_O}, T_{con\_O}\right) \tag{8}$$

$$h_{con\_I} = func\left(P_{con\_I}, T_{con\_I}\right) \tag{9}$$

$$h_{con\_O} = func\left(P_{con\_O}, T_{con\_O}\right) \tag{10}$$

From these equations, specific enthalpy can be calculated from temperature and pressure. But because the saturated state of the evaporator inlet refrigerant is difficult to calculate, so we adopted values at the condenser exit considering that heat moving in and out of expansion valves are little.
Overall, from the compressor inlet pressure P_{com_I}, inlet temperature T_{com_I}, outlet pressure P_{com_O}, pressures and temperatures to and fro heat exchangers P_{eva_I}, P_{eva_O}, P_{con_I}, P_{con_O}, T_{eva_I}, T_{eva_O}, T_{con_I}, T_{con_O}, we can obtain the performance of the system.

5 EXPERIMENTAL RESULTS AND DISCUSSIONS

Driving the testing machine under several conditions, we will measure its performance by both A.E. and C.C..

5.1 Test Method And Results

The test was run under several conditions for both cooling and warming. In addition to rated and middle conditions from JIS, we measured performances under different outdoor temperatures and loads. The conditions are as in the following table.
For Fig.7, we plotted the calculated performances by A.E. along the horizontal axis, and plotted those of the C.C. along the vertical axis. The same kind of behaviour can be observed from both methods under various conditions such as cooling, heating and different loads. Although the maximum relative error for AE is roughly 15%, errors where less than 4% across the regression line through both cooling and heating conditions thus, high accuracy was obtained and, we can expect further improvement in accuracy by considering and correcting more specifically phenomenon.

5.2 Consideration Of Error Source

The supposed causes for error in calculating performances by C.C are as follows: error margin of thermo couples and pressure transducer, heat loss of heat exchangers and error caused by bypass short cut between indoor heat exchangers.

The difference between performance calculated by C.C was small compared to A.E, but considering the fact that performances where calculated excessively by C.C, we can say that effects by the bypass was large. The bypass between sub cooling heat exchangers exchanges heat between refrigerants thus, has no effects on the performance. So the remaining main causes for error are factors such as oil tempering bypath and refrigerant leakage at four way valves.

In aiming to obtain further accuracy for C.C, we must consider these causes for each device and correct them.

Table 3: Test condition

Data No.	SHF	Load%	In Temp. Dry / Wet	Out Temp. Dry / Wet
C1	–	rated	27/19	35/24
C2	–	middle	27/19	35/24
C3	0.85	75	27/19	35/24
C4	1	75	27/19	35/24
C5	0.85	100	27/19	35/24
H1	–	rated	20/15	7/6
H2	–	middle	20/15	7/6
H3	–	25	20/15	7/6
H4	–	75	20/15	7/6
H5	–	100	20/15	7/6
H6	–	50	20/15	12/11

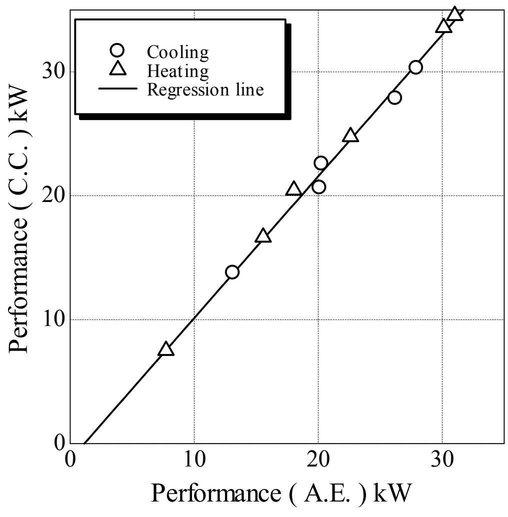

Figure 7: Comparison C.C. and A.E. method

5.3 Correction method

In case of production models, it is difficult to add measurement devices excessively because of cost issue. In this section, we will roughly calculate performances with our C.C. using minimum data for controlling, obtained by built in sensors in the devices. For input values,

compressor inlet/outlet pressures and temperatures are used, outlet temperatures of each heat exchangers and rotational speed of compressors. We measured compressor outlet pressures at the single point where the two flows joins together and also, corrected delicate bypass values of oil tempering and leakage of four way valves as a function of condensing/evaporating pressure.
Fig.8 shows calculations results. In case of using built in sensors, the refrigerant flow rate of indoor heat exchangers where corrected by considering refrigerant bypass. And under large differential driving regions, cooling capacity was predicted with relative error less than 10%.

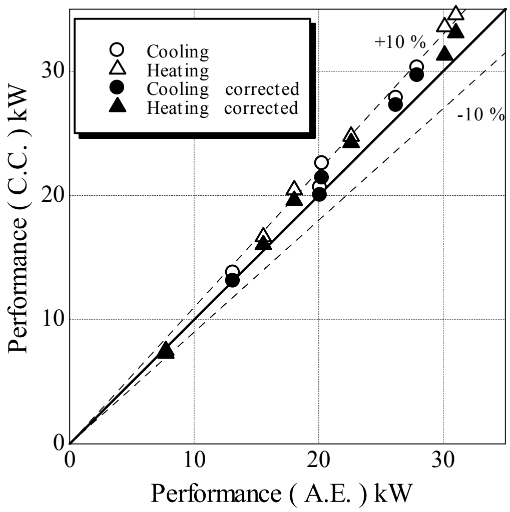

Table 8: Comparison C.C. and A.E. method
(Collected results)

6 CONCLUSION

In this research, we have introduced a method to calculate performances called the C.C. and, confirmed that performances for even complex systems such as the VRF could be predicted by compressor curves and few measured data. Although maximum relative error is 15% compared to A.E. because of refrigerant leakage and bypass, error was under 4% and proved to be accurate for both cooling and heating. Furthermore, by adding the above corrections, we proved that it is possible to predict performances with data acquired by minimum number of sensors set on the air conditioners.
With C.C., it becomes possible to easily predict air conditioning performances with small investments. This is because recent high efficient air conditioners contain various sensors inside.
With VRF, if you set in a reliable C.C measuring system as a part of the device, you can avoid excess designing from driving records. Thus, reduce driving time under low load and uplift periodic efficiency. We would like to consider these effects and report them from here on.

7 NOMENCLATURE

$freq$: Frequency, Hz
G : mass flow rate, $kg \cdot s^{-1}$
h : specific enthalpy, $kJ \cdot kg^{-1}$
P : pressure, kPa
Q : heat quantity, kW
T : temperature, K

8 SUBSCRIPTION

com	: compressor
con	: condenser
eva	: evaporator
fan	: fan
I	: inlet
O	: outlet

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