UNSTEADY STATE SIMULATION OF VRF SYSTEMS

Keisuke, OHNO, Research associate, Waseda Univ., 3-4-1 Okubo Shinjuku Tokyo, JAPAN;
Kiyoshi, SAITO, Professor, Waseda Univ.;
Hokuto, NAKAMURA, Mr., P.T.Morimura & Associates,Ltd.;
Hiromichi, MURATA, Mr., P.T.Morimura & Associates,Ltd.;
Yukihiro JINNO, Mr., The Kansai Electric Power Co.,Inc.;
Katsuhiro KONISHI, Mr., The Kansai Electric Power Co.,Inc.;
Yasuhisa NAKASO, Mr., The Kansai Electric Power Co.,Inc.;

Abstract: Due to worldwide demand for energy saving, many new technologies have been developed in the air conditioning field. VRF (Variable Refrigerant Flow) system is a recently developed air conditioning system that has some indoor units. This system has advantage for space and cost saving. In recent year, we have to improve the annual performance of this system. In the annual performance, intermittent driving and defrost operation have to be considered. The study based on experiment has limitations to grasp the system performance well because there are many driving and control patterns. Therefore, theoretical approach with unsteady state simulation is required. However, there are few reports concerning the simulation for VRF systems. In this study, we constructed unsteady state simulation model and carried out unsteady simulation for VRF system. Moreover, we verified its validity by comparing the simulation results with the experimental ones using the actual system. As a result, we made sure that the constructed simulation model could predict actual behavior of systems.

Key Words: Unsteady state simulation, Compression type heat pump, VRF system

1 INTRODUCTION

VRF system is an air conditioning system, which has many indoor units against a single outdoor unit such as multi air conditioners for buildings. Compared to single type air conditioners, the system performs as well and enables us to save space and energy, even though there are several indoor units.
But because there are several indoor units each placed in different rooms, driving conditions for each unit differs by its usage conditions. Thus, not only when driving condition are maximum full load or minimum light load, unbalance of loads occur at pauses and extreme loads.
So, in order to improve the performance and efficiency of this system without losing reliability and usability, accurate simulation is vital. This simulation must include conditions where each indoor unit are driving in various ways and furthermore, unsteady state during mode change.

In this research, we constructed a simulation model based on a device containing 4 indoor units and, verified its validity by running performance tests on the device.

2 SYSTEM

The refrigerator we are dealing with is a compression type VRF system, which is composed of one outdoor unit and four indoor units. The outdoor unit is loaded with two compressors and a heat exchanger for sub cooling. The indoor units are a cassette type and are a set in the ceiling. The cooling performance for each indoor unit is controlled by expansion valves, which are set in indoor units. Rated cooling capacity is 28.0kW, and rated cooling input for outdoor unit is 7.19kW. For refrigerant we used R410A. Fig.1 shows the system flow.

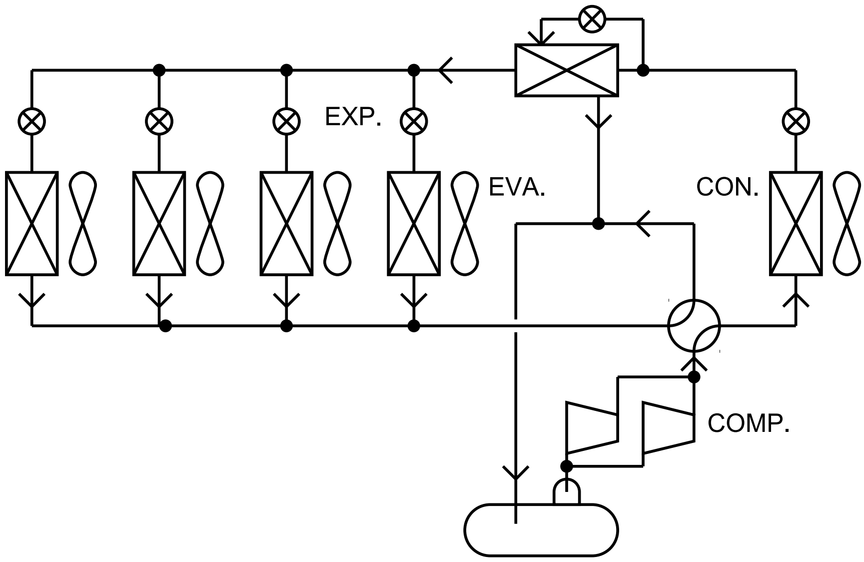


Figure 1: Schematic flow of VRF system

3 SIMULATION

3.1 Simulation Method

Following the modular analysis method, first each analyzing model is discrerized. In doing so, we adopted finite one dimensional distributed parameter system using upwind for adjective term. For discretization, we used staggered grid to stabilize calculation. The modules are connected based on the mass flow rate, pressure and specific enthalpy of the points where the fluids flow in and out. Then we will state boundary conditions at the points between modules are the same, thus the system dominant equation and boundary conditions are complete [1][2][3] [4]. These equations are nonlinear, so we adopted a convergence algorithm to obtain the nonlinear solution.

3.2 Simulation Models

3.2.1 Compressor

Compressor model equations are as follows.

$$\frac{d\rho}{dt}V = G_I - G_O \tag{1}$$

$$\frac{d\rho u}{dt}V = G_I' h_I' - G_O h_O \tag{2}$$

$$\eta = \frac{h_{Iad}^{'} - h_I}{h_I^{'} - h_I} \qquad (3)$$

$$s_{Iad}^{'} = s_I \qquad (4)$$

$$G_I = \frac{n \rho_I \eta V}{60} \qquad (5)$$

$$G_I^{'} = G_I \qquad (6)$$

$$G_I^{'} h_I^{'} = G_I h_I + W \qquad (7)$$

$$P_O = P \qquad (8)$$

$$h_O = h \qquad (9)$$

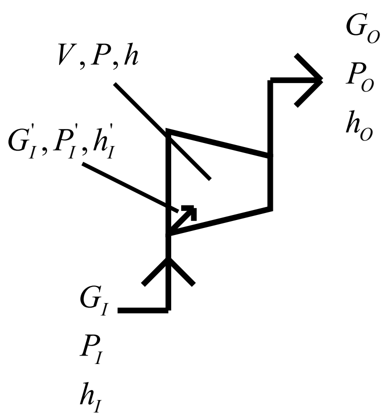


Figure 2: Compressor

3.2.2 Heat Exchanger

Heat exchanger model equations are as follows.

$$\frac{\partial \rho_R}{\partial t} S_R = -\frac{\partial G_R}{\partial x} \qquad (10)$$

$$\frac{\partial P_R}{\partial x} = 0 \qquad (11)$$

$$\frac{\partial \rho_R u_R}{\partial t} S_R = -\frac{\partial G_R h_R}{\partial x} + \pi D_{M\_in} q \qquad (12)$$

$$\frac{\partial \rho_M u_M}{\partial t} S_M = \pi D_{M\_out} q_M - \pi D_{M\_in} q \qquad (13)$$

$$g\left(h_O - h_I\right) = -\pi D_{M\_out} q_M \qquad (14)$$

$$q = \alpha\left(T_M - T_R\right) \tag{15}$$

$$q_M = \alpha_M\left(T_A - T_M\right) \tag{16}$$

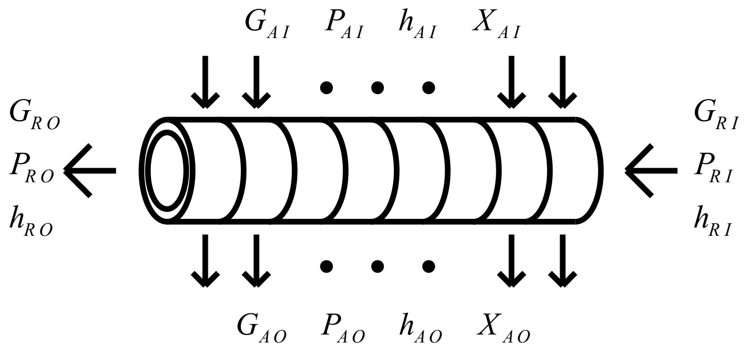


Figure 3: Heat Exchanger

3.2.3 Connecting Tube

Connecting tube model is based on the heat exchanger model. In this model, heat transfer of outside is neglected. Equation is as follows.

$$q_M = 0 \tag{17}$$

3.2.4 Sub Cool Heat Exchanger

Sub cool heat exchanger model is based on heat exchanger model. It is treated as opposed flow and discrerized eq. (10), (11) and (12).

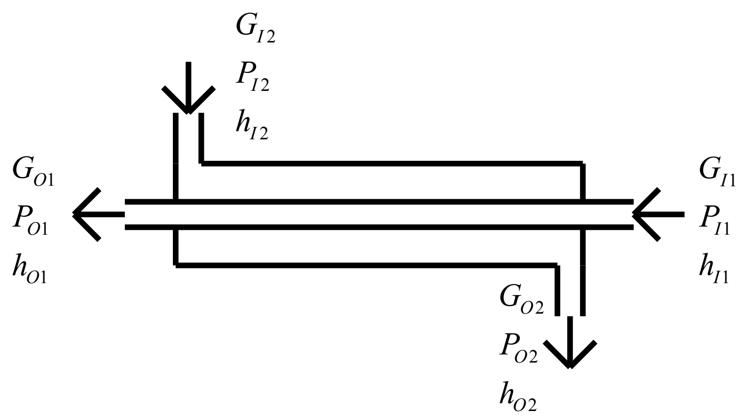


Figure 4: Sub Cool Heat Exchanger

3.2.5 Reversing Valve
Reversing valve model equations are as follows.

$$\frac{d\rho_1}{dt}V_1 = G_{I1} - G_{O1} \tag{18}$$

$$\frac{d\rho_1 u_1}{dt}V_1 = G_{I1}h_{I1} - G_{O1}h_{O1} \tag{19}$$

$$P_{I1} = P_{O1} \tag{20}$$

$$\frac{d\rho_2}{dt}V_2 = G_{I2} - G_{O2} \tag{21}$$

$$\frac{d\rho_2 u_2}{dt}V_2 = G_{I2}h_{I2} - G_{O2}h_{O2} \tag{22}$$

$$P_{I2} = P_{O2} \tag{23}$$

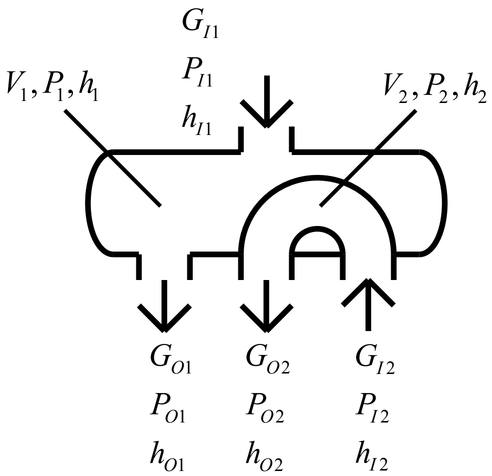


Figure 5: Reversing Valve

3.2.6 Accumulator
Accumulator model equations are as follows.

$$\frac{d\rho}{dt}V = G_I - G_O \tag{24}$$

$$\frac{d\rho u}{dt}V = G_I h_I - G_O h_O \tag{25}$$

$$P_I = P \tag{26}$$

$$P_O = P \tag{27}$$

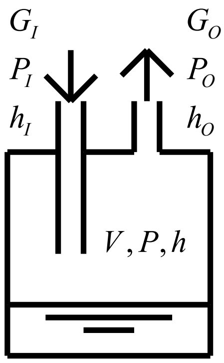


Figure 6: Accumulator

3.2.7 Expansion Valve

Expansion valve model equations are as follows.

$$\frac{d\rho}{dt}V = G_I - G_O \tag{28}$$

$$\frac{d\rho u}{dt}V = G_I h_I - G_O h_O \tag{29}$$

$$G_I = \rho_I Ca\sqrt{2\frac{(P_I - P_O)}{\rho_I}} \tag{30}$$

$$P_O = P \tag{31}$$

$$h_O = h \tag{32}$$

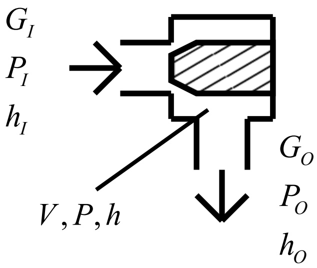


Figure 7: Expansion Valve

3.2.8 Mixer

Mixer model equations are as follows.

$$\frac{d\rho}{dt}V = G_{I1} + G_{I2} - G_O \tag{33}$$

$$\frac{d\rho u}{dt}V = G_{I1}h_{I1} + G_{I2}h_{I2} - G_O h_O \tag{34}$$

$$P_{I1} = P \tag{35}$$

$$P_{I2} = P \tag{36}$$

$$P_O = P \tag{37}$$

$$h_O = h \tag{38}$$

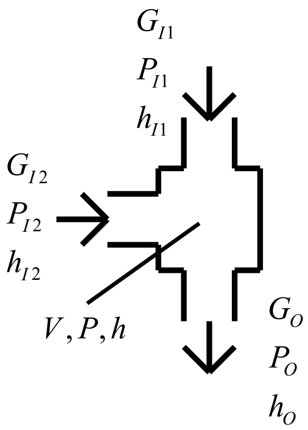


Figure 8: Mixer

3.2.9 Separator

Separator model equations are as follows.

$$\frac{d\rho}{dt} V = G_I - G_{O1} - G_{O2} \tag{39}$$

$$\frac{d\rho u}{dt} V = G_I h_I - G_{O1} h_{O1} - G_{O2} h_{O2} \tag{40}$$

$$P_I = P \tag{41}$$

$$P_{O1} = P \tag{42}$$

$$P_{O2} = P \tag{43}$$

$$h_{O1} = h \tag{44}$$

$$h_{O2} = h \tag{45}$$

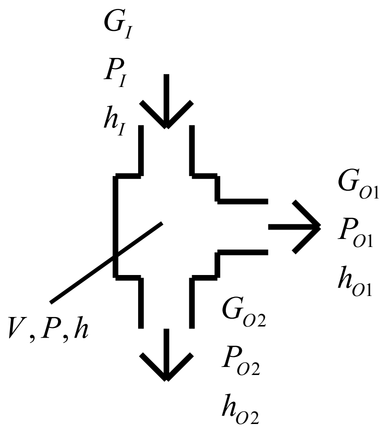
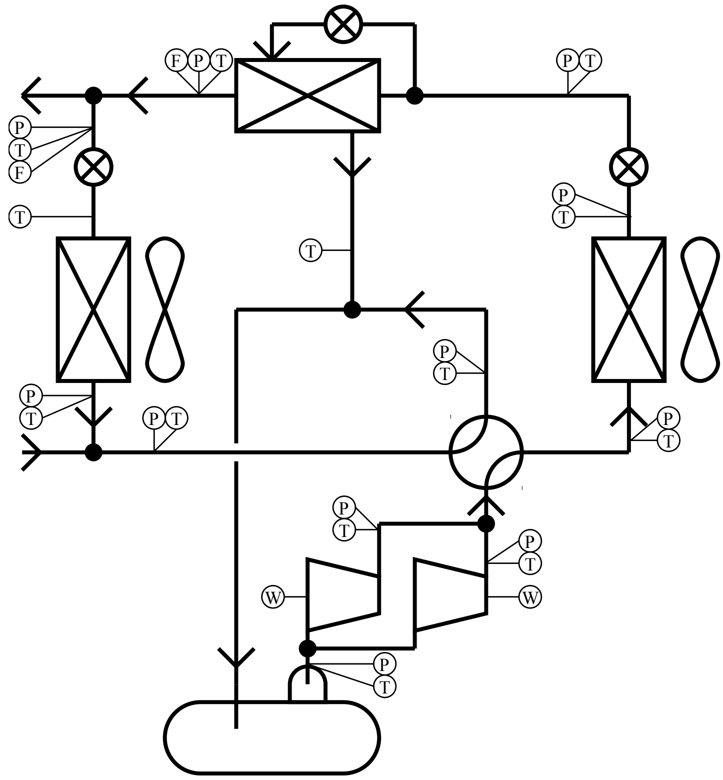


Figure 9: Separator

4 Experiment Apparatus

In our research, we have confirmed the validity of our unsteady simulation model by performance tests from our actual equipments. Measurement terms are shown on Fig. 10. On Fig.10, only one indoor unit is shown, but the other three indoor units also posses the same measuring instruments.



P: Pressure transducer
T: Thermo couple
F: Flow meter

Figure 10: Measurement Points

5 RESULTS and DISCUSSION

5.1 Steady State Operation

In our research, we obtained the amount of heat transfer coefficient, flow coefficient, isentropic efficiency and volumetric efficiency from experiments and regarded them as constant. These values are obtained by driving the testing machine at rated state, which follows JIS B 8616. On Fig.11, the comparison between the results from the rated state test and the simulation is shown. The plots show the test results, and the line shows the simulation results. The two results are in good agreement.

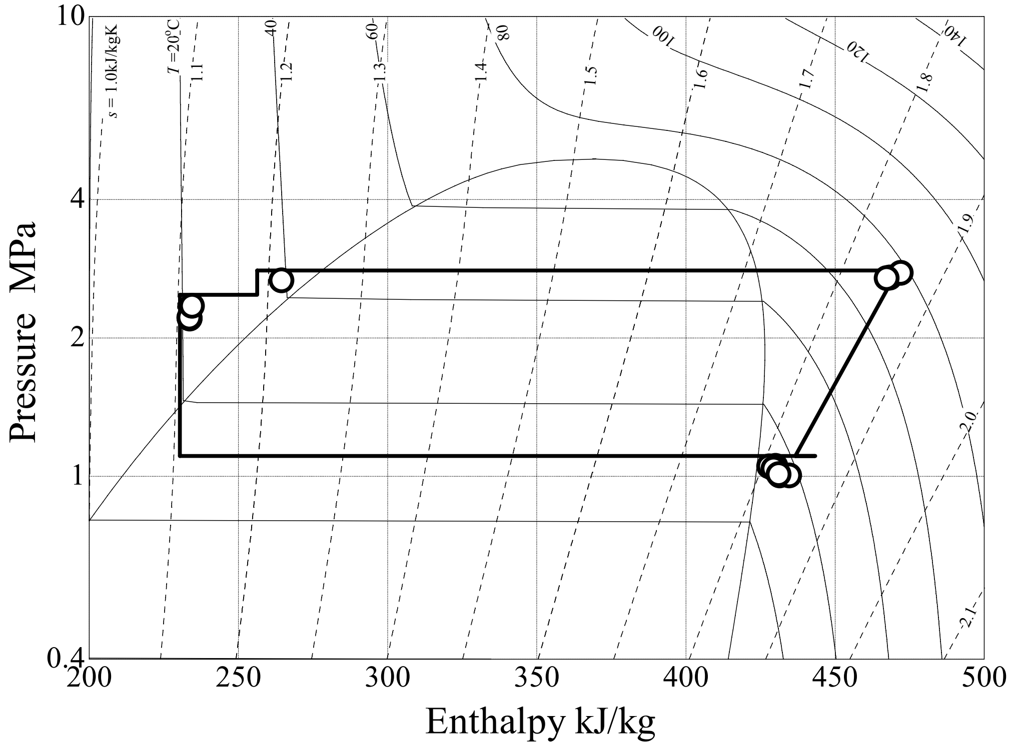


Figure 11: Steady Simulation Result on p-h Diagram

5.2 Unsteady State Operation

Next, we will verify the unsteady state simulation. For unsteady state conditions, we will deal with a situation where the numbers of driving indoor units change. On Fig.12, the results of a situation where the numbers of driving units change from 1 to 4 are shown. On Fig. 13, the results of a situation where the numbers of driving units change from 4 to 1 are shown. On the first situation, cooling capacities and compressor inputs are in good agreement. Despite the fact that we are regarding the heat transfer rate a constant value and not considering pressure drop, we can say that it is possible to predict performances very accurately. On the second situation, the response after changing the cooling capacities and the lower pressure differed. We are searching for the cause for this at the moment. As a whole, the result for both the simulation and the experiment coincided very well.

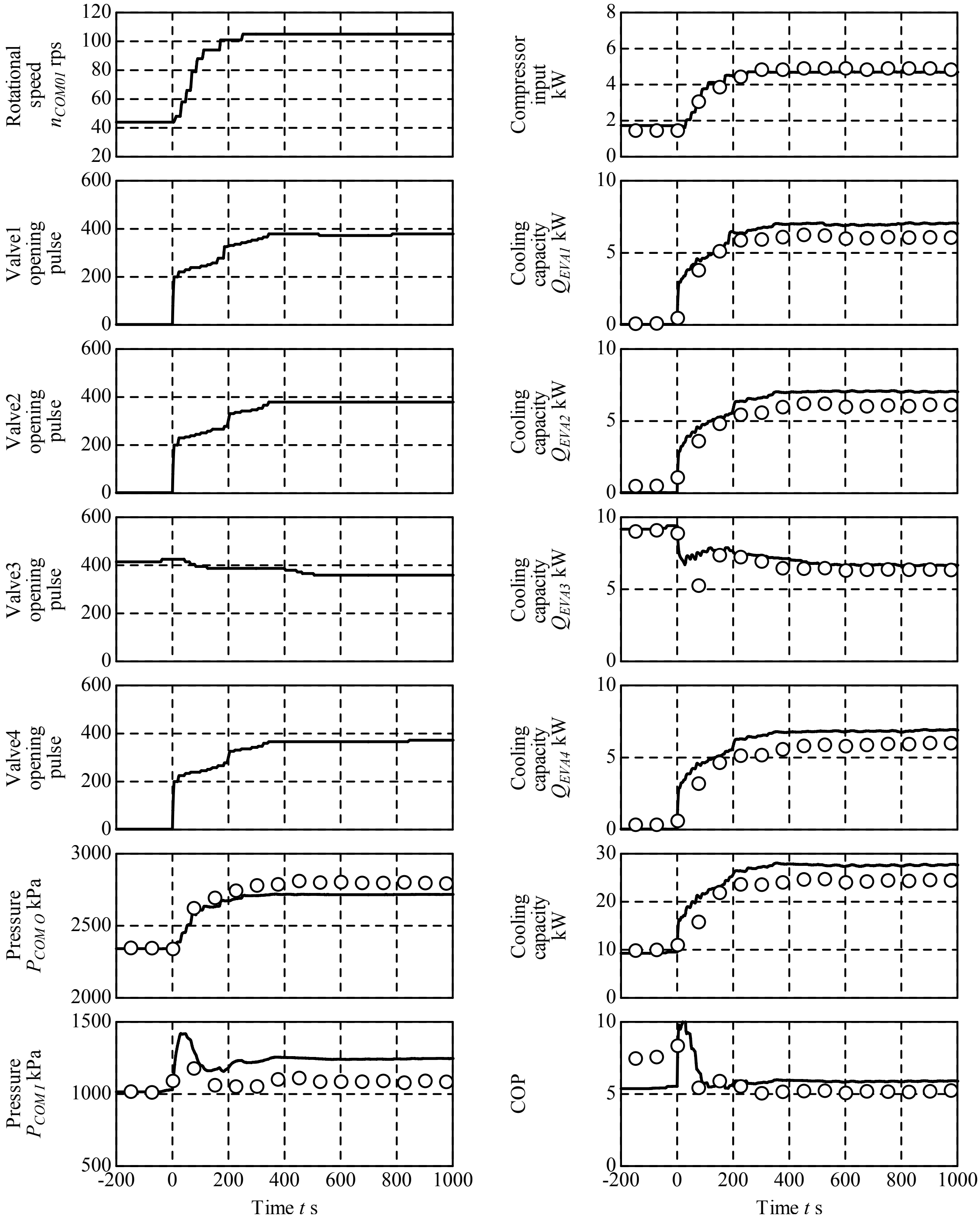


Figure 12: Unsteady Simulation Result, Units change from 4 to 1

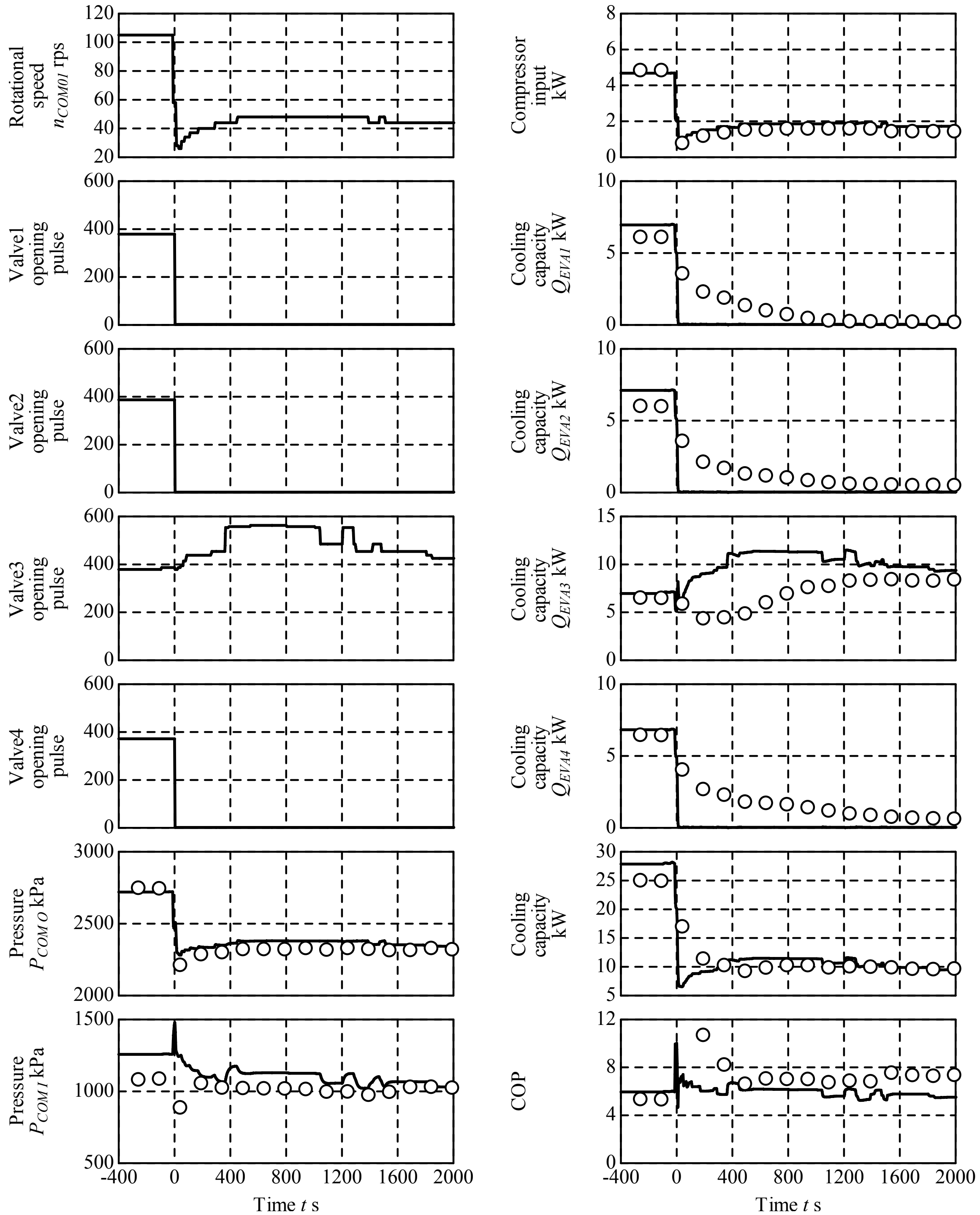


Figure 13: Unsteady Simulation Result, Units change from 1 to 4

6 CONCLUSION

In this study, unsteady state simulation model of VRF system is constructed. Simulation results and experimental results are in good agreements. More accuracy simulation model is constructed and control method for intermittent driving is considered in the future.

7 NOMENCLATURE

a : Cross section area, m^2
C : Flow coefficient, -
D : Diameter, m
G : Mass flow rate, $kg \cdot s^{-1}$
g : Mass flow rate per unit length, $kg \cdot m^{-1} \cdot s^{-1}$
h : Specific enthalpy, $kJ \cdot kg^{-1}$
P : Pressure, kPa
Q : Heat exchange rate, kW
q : Specific heat flux, $kW \cdot m^{-2}$
S : Area, m^2
T : Temperature, ^{o}C
t : Time, s
u : Specific internal energy, $kJ \cdot kg^{-1}$
V : Volume, m^3
W : Work, kW
X : Humidity, $kg \cdot kg(DA)^{-1}$
α : Heat transfer coefficient, $kW \cdot m^{-2} \cdot K^{-1}$
ρ : Density, $kg \cdot m^{-3}$

8 SUBSCRIPTION

A : Air
COM : Compressor
EVA : Evaporator
I : Inlet
In : Inside
M : Tube
O : Outlet
Out : Outside
R : Refrigerant

9 REFERENCES

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