Development of dynamic model of variable refrigerant flow cooling system based on moving boundary method

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Problem definition

Set value of compressor inlet pressure is constant

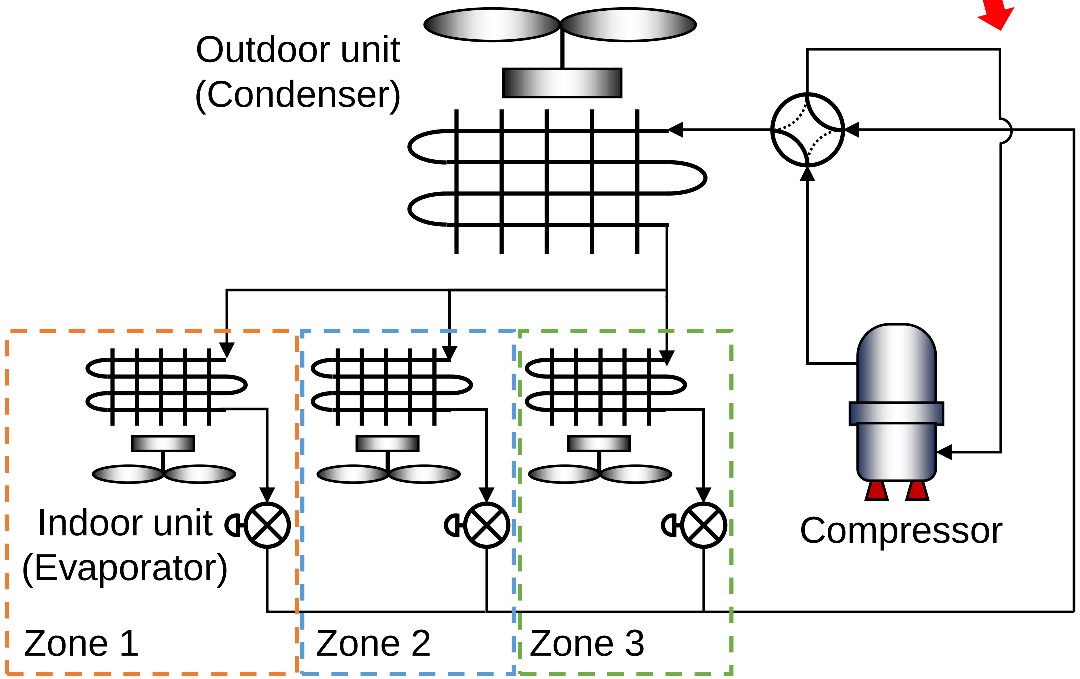
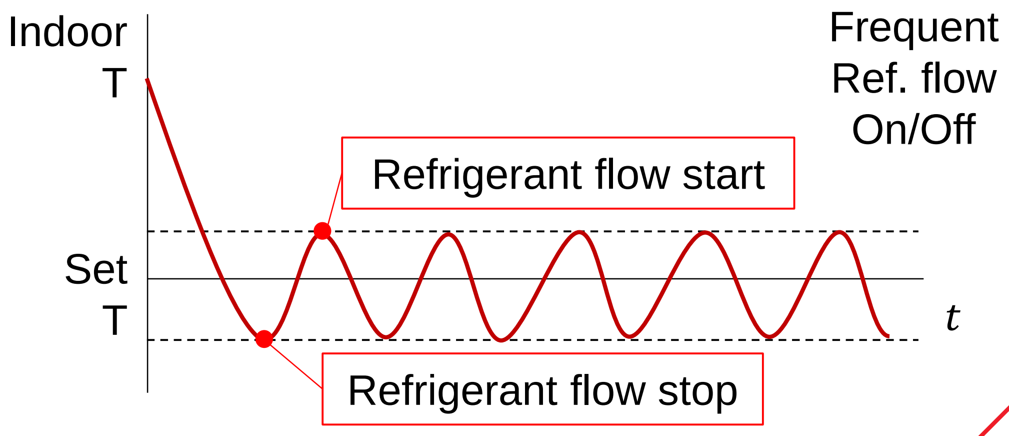


Fig. Schematic diagram of a VRF cooling system

Cooling load is smaller than cooling capacity



- Currently, the evaporation pressure of the VRF system is controlled so that the compressor inlet pressure is constant regardless of its cooling load.
- This results in frequent on/off of refrigerant flow in the indoor unit when the cooling load is small

Problem definition

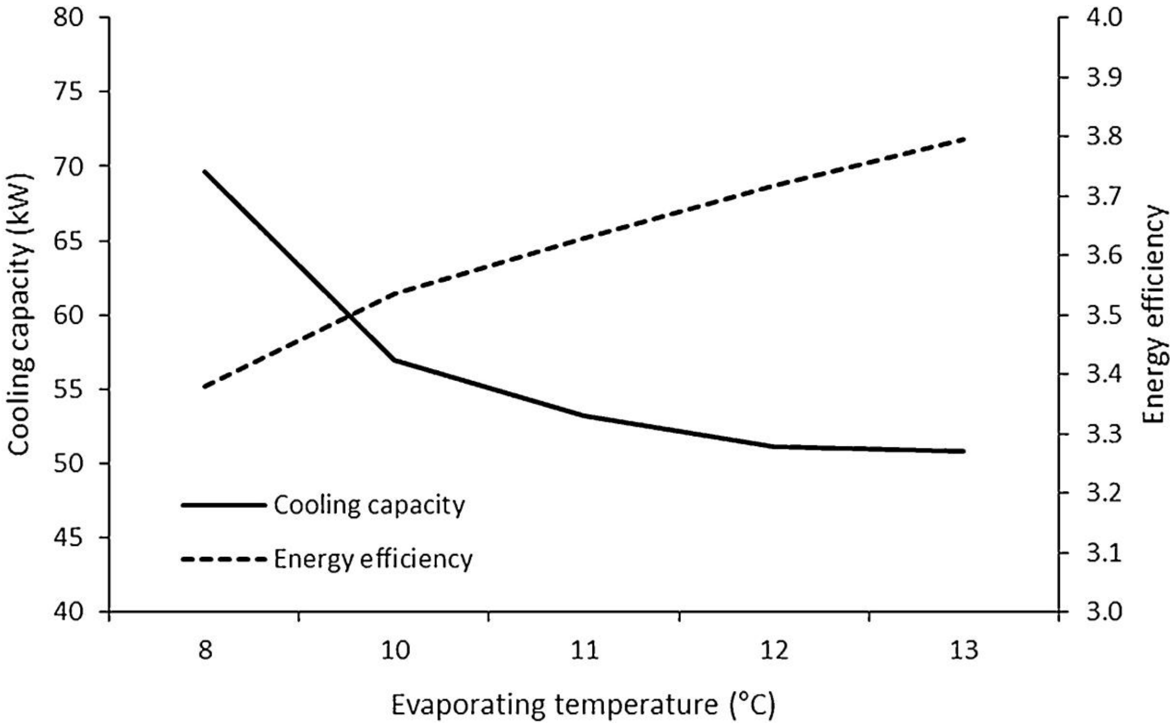
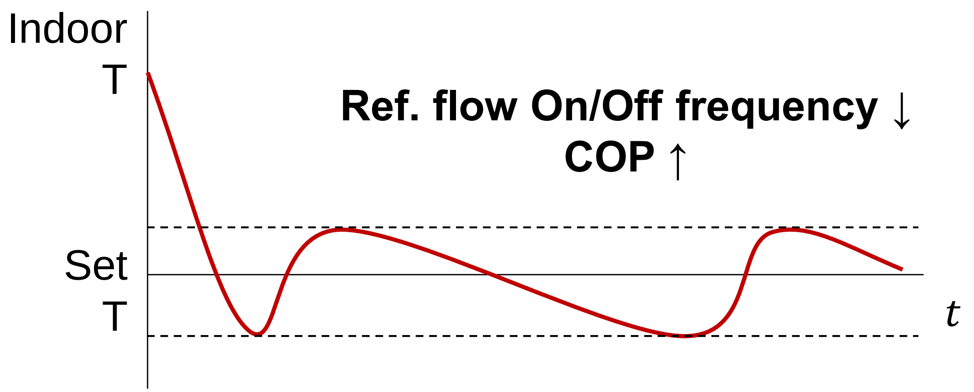


Fig. Effects of the evaporating temperature on the cooling capacity and COP [1]

When appropriate target pressure controller is applied



- When the evaporation temperature(pressure) is increased, the cooling capacity decreases but the energy efficiency increases.
- In order to achieve high energy efficiency, **a load responsive controller that controls the evaporation pressure according to the cooling load** is being developed.

[1] Yun et al., 2016, Energy Build., Development and application of the load responsive control of the evaporating temperature in a VRF system for cooling energy savings

Flow of research

Component Model Development → VRF System Model Development → Validation of the Model

Development of dynamic model of VRF system

↓

Evaluation ← Performance Simulation ← Controller Design

Development of Load Response Set Pressure Control algorithm

Major components of a dynamic model

2. Dynamic Model Development

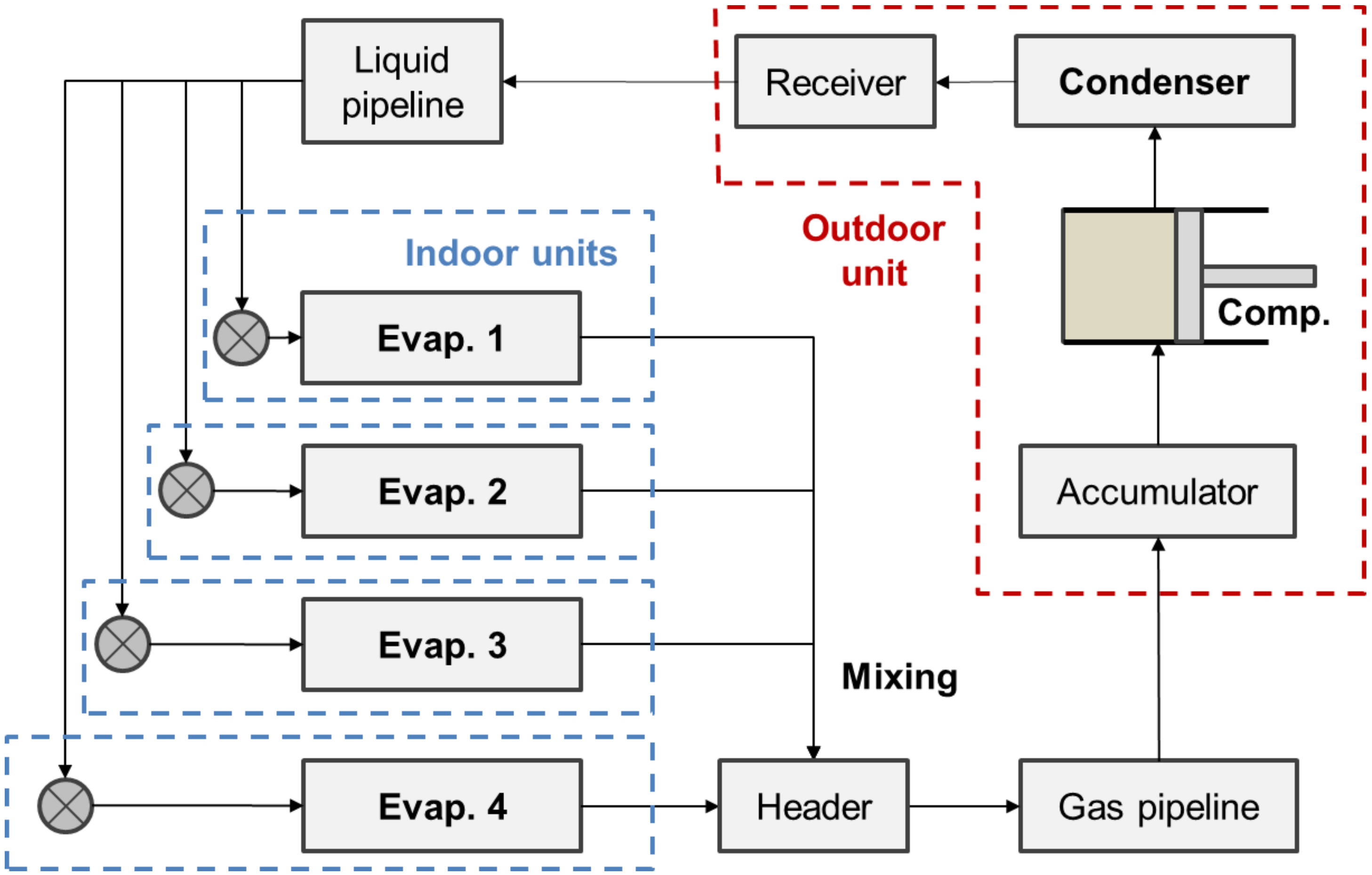


Fig. A schematic diagram of a dynamic model developed by simplifying a actual VRF system

- In this study, the dynamic model was developed by combining major component models of **compressor, condenser, receiver, accumulator, refrigerant pipes, multiple evaporators, and refrigerant distribution header.**

Heat exchanger model - Governing equations

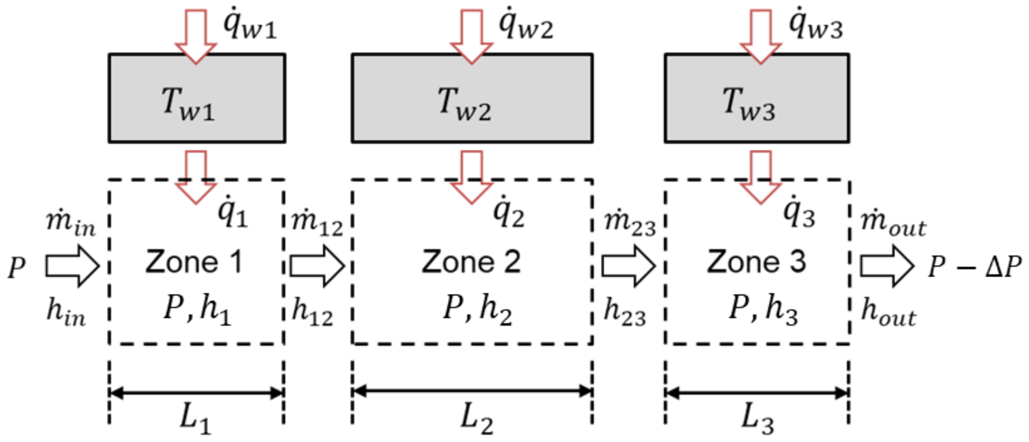


Fig. Schematic of a general heat exchanger including subcool-two phase-superheat section [2]

Refrigerant mass conservation

$$A\frac{d}{dt}\int_{z_A}^{z_B}\rho dz + A\rho_A\dot{z}_A - A\rho_B\dot{z}_B = \dot{m}_A - \dot{m}_B$$

Refrigerant energy conservation

$$A\frac{d}{dt}\int_{z_A}^{z_B}\rho h dz - A(z_B - z_A)\dot{P} + A\rho_A h_A\dot{z}_A$$
$$-A\rho_B h_B\dot{z}_B = \dot{m}_A h_A - \dot{m}_B h_B + \dot{q}(z_B - z_A)$$

Tube wall energy conservation

$$A_w\rho_w c_w(z_B - z_A)\dot{T}_w$$
$$+A_w\rho_w c_w(T_w(z_A) - T_w)\dot{z}_A$$
$$-A_w\rho_w c_w(T_w(z_B) - T_w)\dot{z}_B$$
$$= \dot{q}_w(z_B - z_A) + \dot{q}(z_B - z_A)$$

$z: position\ of\ boundary$

$A: inlet\ \ B: outlet$

- The moving boundary method is to model each section as a control volume that has variable boundaries and lumped properties and track the length of the different sections dynamically.
- For each control volume, three conservation equations can be obtained.
- Then, dynamic behavior of the heat exchanger can be expressed with $\dot{P}, \dot{L}_1, \dot{L}_2, \dot{L}_3,$ $\dot{h}_1, \dot{h}_2, \dot{h}_3, \dot{\rho}_1, \dot{\rho}_2, \dot{\rho}_3, \dot{T}_{w1}, \dot{T}_{w2}, \dot{T}_{w3}$.

[2] Qiao et al., 2016, Int. J. Refrig., An advanced switching moving boundary heat exchanger model with pressure drop

Heat exchanger model - Governing equations

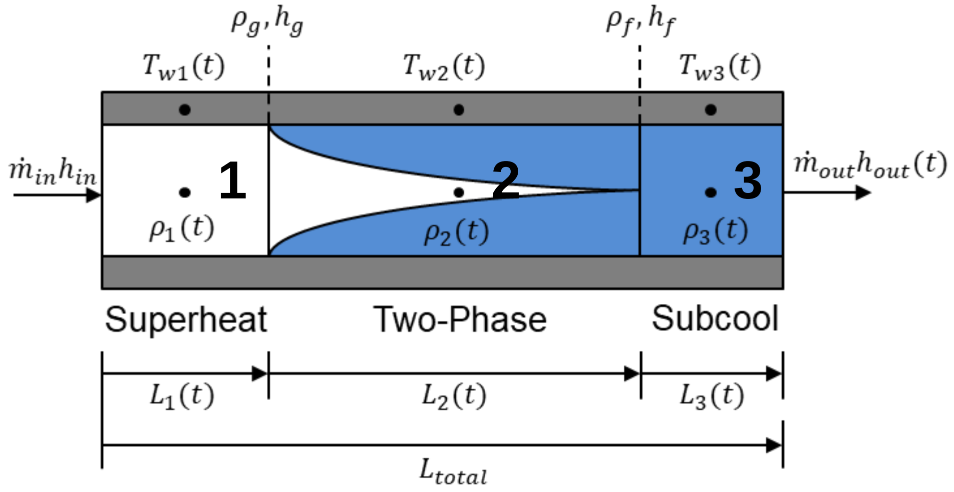


Fig. 3-section condenser analysis with moving boundary method

Mean void fraction equation

$$\rho_2 = \bar{\gamma}\rho_g + (1-\bar{\gamma})\rho_f$$

$$\rho_2 h_2 = \bar{\gamma}\rho_g h_g + (1-\bar{\gamma})\rho_f h_f$$

& $$\dot{\bar{\gamma}} = \frac{\partial \bar{\gamma}}{\partial P}\dot{P}$$

Equation of state

$$\rho_i = \frac{\partial \rho_i}{\partial P}\dot{P} + \frac{\partial \rho_i}{\partial h_i}\dot{h}_i \quad \text{for } i = 1,3$$

Linear enthalpy distribution

$$h_1 = \frac{h_{in} + h_g}{2}, h_3 = \frac{h_{out} + h_f}{2}$$

Total tube length is constant

$$\dot{L}_1 + \dot{L}_2 + \dot{L}_3 = 0$$

Boundary conditions

$$[\dot{m}_{in}, \dot{m}_{out}, h_{in}]^T$$

⇒ Dynamic behavior can be expressed with

$$\dot{\mathbf{x}} = [\dot{L}_1, \dot{L}_2, \dot{P}, \dot{h}_{out}, \dot{T}_{w1}, \dot{T}_{w2}, \dot{T}_{w3}, \dot{\bar{\gamma}}]^T$$

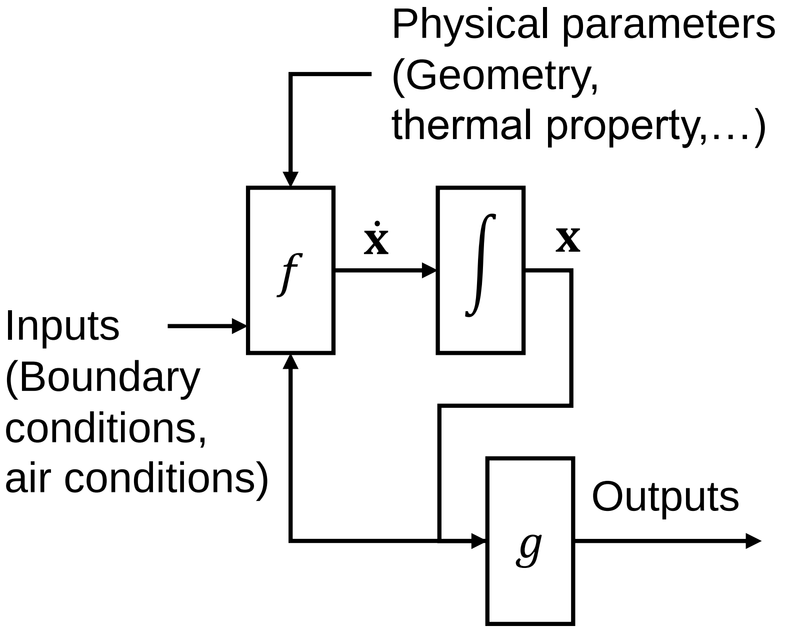
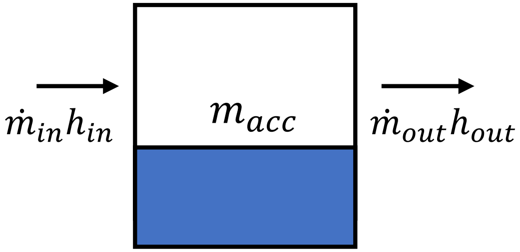


Fig. Mathematical model [3]

[3] McKinley and Alleyne, 2008, Int. J. Refrig., An advanced nonlinear switched heat exchanger model for vapor compression cycles using the moving-boundary method

Other component models

Accumulator & Receiver



Accumulator

$$h_{out}\ \begin{cases} h_{out} = h_g (\text{if } \rho_{acc} \geq \rho_g) \\ h_{out} = h_{acc}\ (\text{if } \rho_{acc} < \rho_g) \end{cases}$$

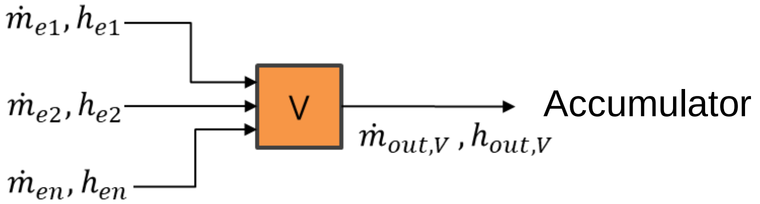
Mass Conservation

$$\dot{m}_{acc} = \dot{m}_{in} - \dot{m}_{out}$$

Energy Conservation

$$\left(\frac{\partial h}{\partial P} m_{acc} - V_{acc}\right)\dot{P} + \left(h_{acc} + \rho_{acc}\frac{\partial h}{\partial \rho}\right)\dot{m}_{acc}$$
$$= \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}$$

Refrigerant mixing in header [4]



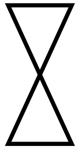
$\dot{h}_v \approx 0$ at header,

$$\rightarrow \begin{cases} V\dfrac{\partial \rho_V}{\partial P_V}\dot{P}_V = \sum \dot{m}_{e,k} - \dot{m}_{out,V} \\ h_{out,V} = \sum \dot{m}_{e,k}h_{e,k} \,/ \sum \dot{m}_{e,k} \end{cases}$$

Pressure drop in pipes and heat exchangers

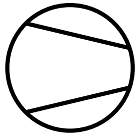
$$\Delta P = f_D \frac{L}{D}\frac{\rho V^2}{2}$$ (Darcy-Weisbach equation)

EEV



$$\dot{m}_{EEV} = C_V\sqrt{\rho_{in}(P_{cond} - P_{eva})}$$
$$h_{out} = h_{in}$$

Compressor



$$\dot{m}_{comp} = \rho_{in}V_{disp}\omega_{comp}\eta_V$$
$$W_{comp} = \dot{m}_{comp}\frac{(h_{comp,in,is} - h_{comp,in})}{\eta_{isen}}$$

[4] Shah et al., 2003, Dynamic modeling and control of single and multi-evaporator subcritical vapor compression systems

Integrated dynamic model

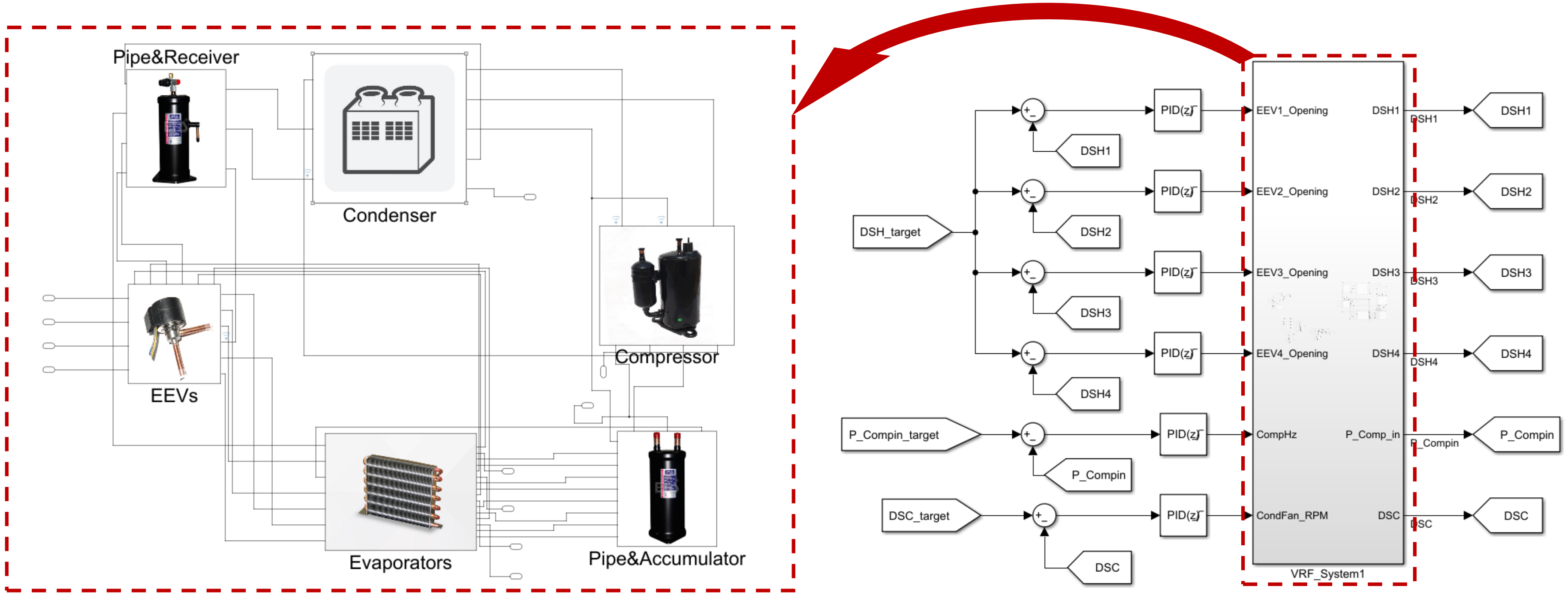


Fig. Dynamic model and control signal block diagram of VRF system implemented on Simulink

- The entire system dynamic model was implemented on Simulink by combining each component model. In addition, the control logic of the actual VRF system is simulated and applied to the dynamic model.

VRF system performance experiments

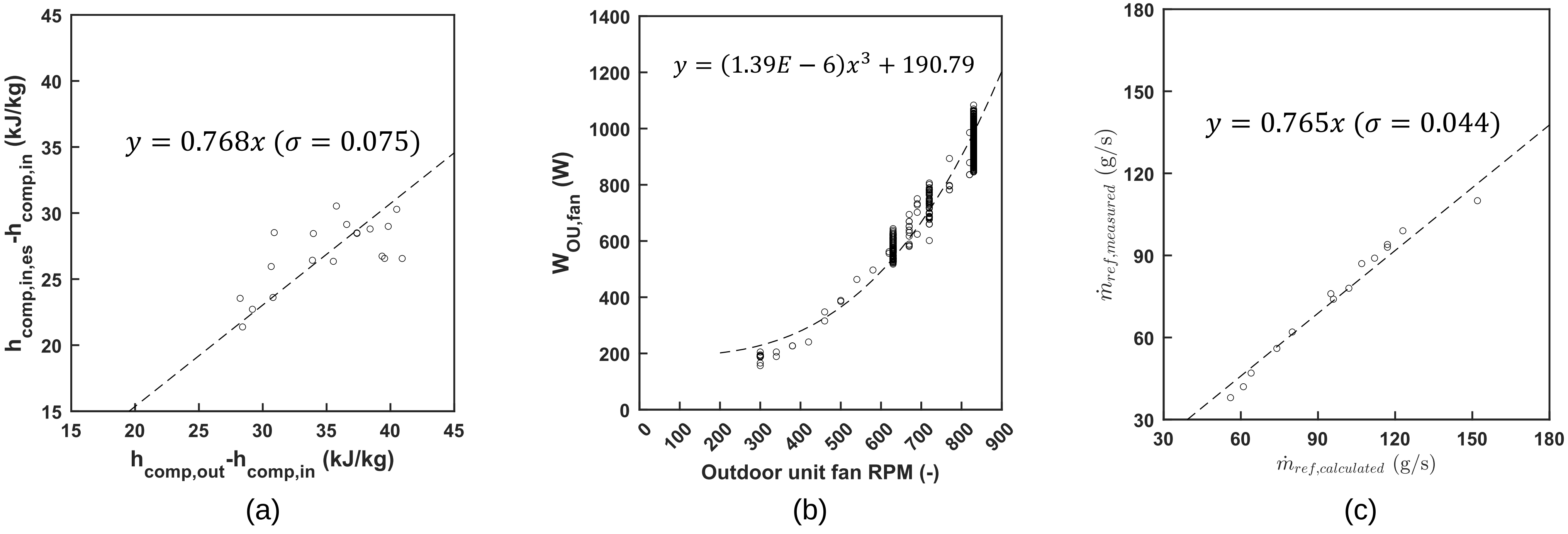


Fig. (a) Isotropic efficiency, (b) fan performance curves, and (c) volumetric efficiency derived from experimental results

- In order to increase the model's accuracy, a VRF system performance test was conducted in a calorimetric chamber. From the experimental data, isentropic efficiency, fan performance curve, and other parameters could be obtained.

Set pressure controller

Governing equation

$$C_{room,sens}\frac{dT_{room}}{dt} = Q_{load,sens} - Q_{cool,sens}$$

$$Q_{cool,sens} = \text{Overall cooling capacity of indoor unit} \times \text{Sensible heat Factor (SHF)} = Q_{cool,total} \times \mathrm{SHF}$$

To maintain zone temperature constant ($\frac{dT_{room}}{dt} = 0$), the sensible cooling capacity matching sensible cooling loads must be supplied.

Required algorithms

1. Sensible cooling load estimating algorithm
2. Set pressure finding algorithm to match the cooling load and capacity
3. Error feedback algorithm

1) Cooling load estimation

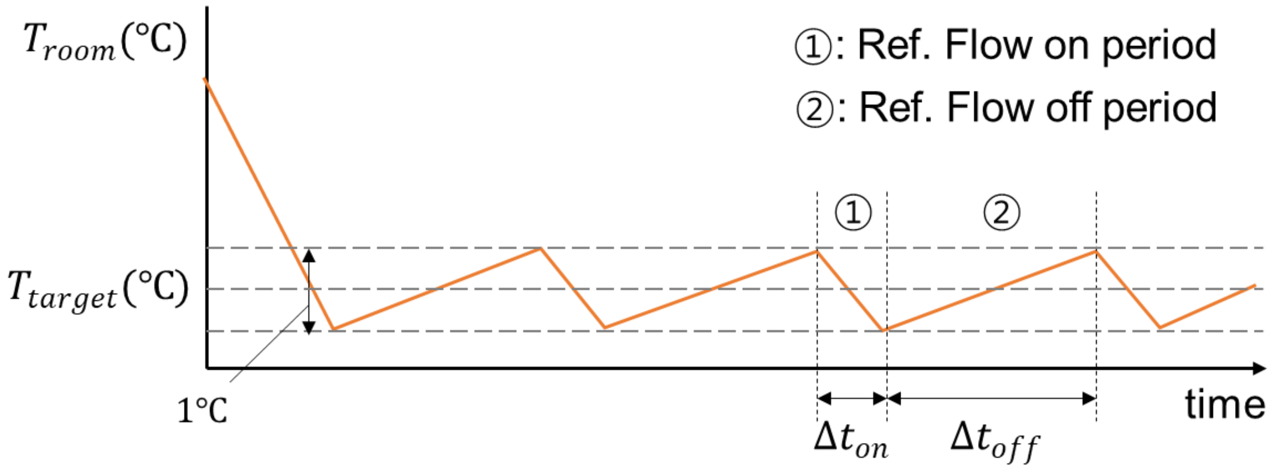


Fig. Temperature profile assuming a linear change in room temperature

For period ①, $C_{room}\frac{dT_{room}}{dt} = Q_{load,sens} - Q_{cool,sens}$

➡ $slope ① = \frac{-1}{\Delta t_{on}} = \frac{Q_{load,sens} - Q_{cool,sens}}{C_{room}}$

For period ②, $C_{room}\frac{dT_{room}}{dt} = Q_{load,sens} \cancel{- Q_{cool,sens}}$

➡ $slope ② = \frac{1}{\Delta t_{off}} = \frac{Q_{load,sens}}{C_{room}}$

Ref. flow on Ratio (ROR) $= \frac{\Delta t_{on}}{\Delta t_{on} + \Delta t_{off}} = \frac{Q_{load,sens}}{Q_{cool,sens}}$

- It is possible to know the ratio of the cooling load to the cooling capacity from the ratio of the ref. flow on and off time (only for sensible heat).

1) Cooling load estimation

Overall cooling capacity calculation

$$\boldsymbol{Q_{cool,total} = \dot{m}_{ref}(h_{eva,out} - h_{eva,in})}$$

① $\dot{m}_{ref} = f(Opening, P_{comp,out}, P_{sat}(T_{eva,in}))$

② $h_{eva,out} = f(P_{sat}(T_{eva,in}), T_{eva,out})$

③ $h_{eva,in} = f(P_{sat}(T_{eva,in}), T_{LiquidPipe})$

SHF calculation

$$\mathbf{SHF} = \frac{\boldsymbol{Q_{cool,sens}}}{\boldsymbol{Q_{cool,total}}} = \frac{\boldsymbol{Q_{cool,sens}}}{\boldsymbol{Q_{cool,sens} + Q_{cool,latent}}}$$

$$Q_{cool,sens} = \dot{m}_a C_{pa}(T_{a,inlet} - T_{wall})\epsilon$$

$$Q_{cool,latent} = \dot{m}_a \Delta h_{fg}(\omega_{a,inlet} - \omega_{wall,sat})\epsilon$$

Assuming that $T_{wall} = T_{eva} + 3$,

$$\therefore \mathrm{SHF} = \frac{C_{pa}(T_{a,inlet} - T_{eva} - 3)}{C_{pa}(T_{a,inlet} - T_{eva} - 3) + \Delta h_{fg}(\omega_{a,inlet} - \omega_{eva,sat})}$$

$$\boldsymbol{Q_{cool,load} = \frac{Q_{cool,load}}{Q_{cool,sens}} \times \frac{Q_{cool,sens}}{Q_{cool,total}} \times Q_{cool,total}} = \mathbf{ROR \times SHF \times} \boldsymbol{Q_{cool,total}}$$

2) Finding set pressure

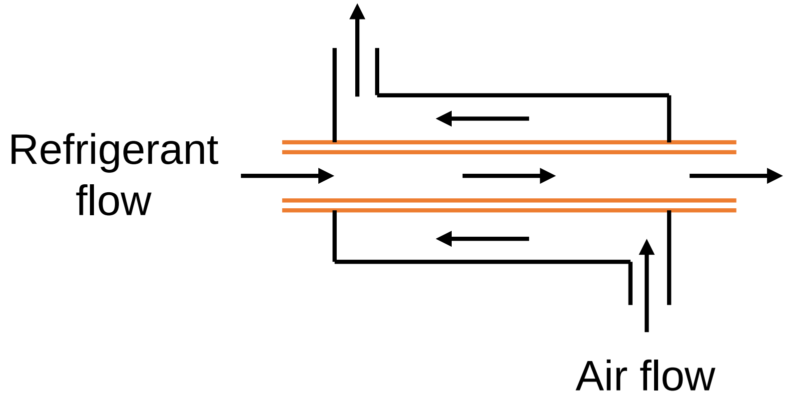


Fig. Finned tube heat exchanger simplified as counterflow heat exchanger

From $Q_{cool,sens} = \dot{m}_a C_{pa}(T_{a,in} - T_{wall})\epsilon$ and

assuming $T_{wall} = T_{eva} + 3$,

$$\boldsymbol{Q_{cool,sens} = k_1 \dot{m}_a C_p (T_{a,in} - T_{eva} - 3)\left(1 - exp\left(-\frac{U_a A_a}{\dot{m}_a C_{pa}}\right)\right)}$$

$Q_{cool,sens}$: from cooling load estimation algorithm

$\dot{m}_a C_{pa}$: from indoor airflow settings (user settings)

$T_{a,in}$: from indoor unit sensor data

U_a: from geometry and airflow data

A_a: from geometry

$\boldsymbol{k_1}$: Correction factor (acquired by performing experiments in advance. 0.93 in this study)

$\} \; T_{eva}$

➡ Target T_{eva} can be found

2) Finding set pressure

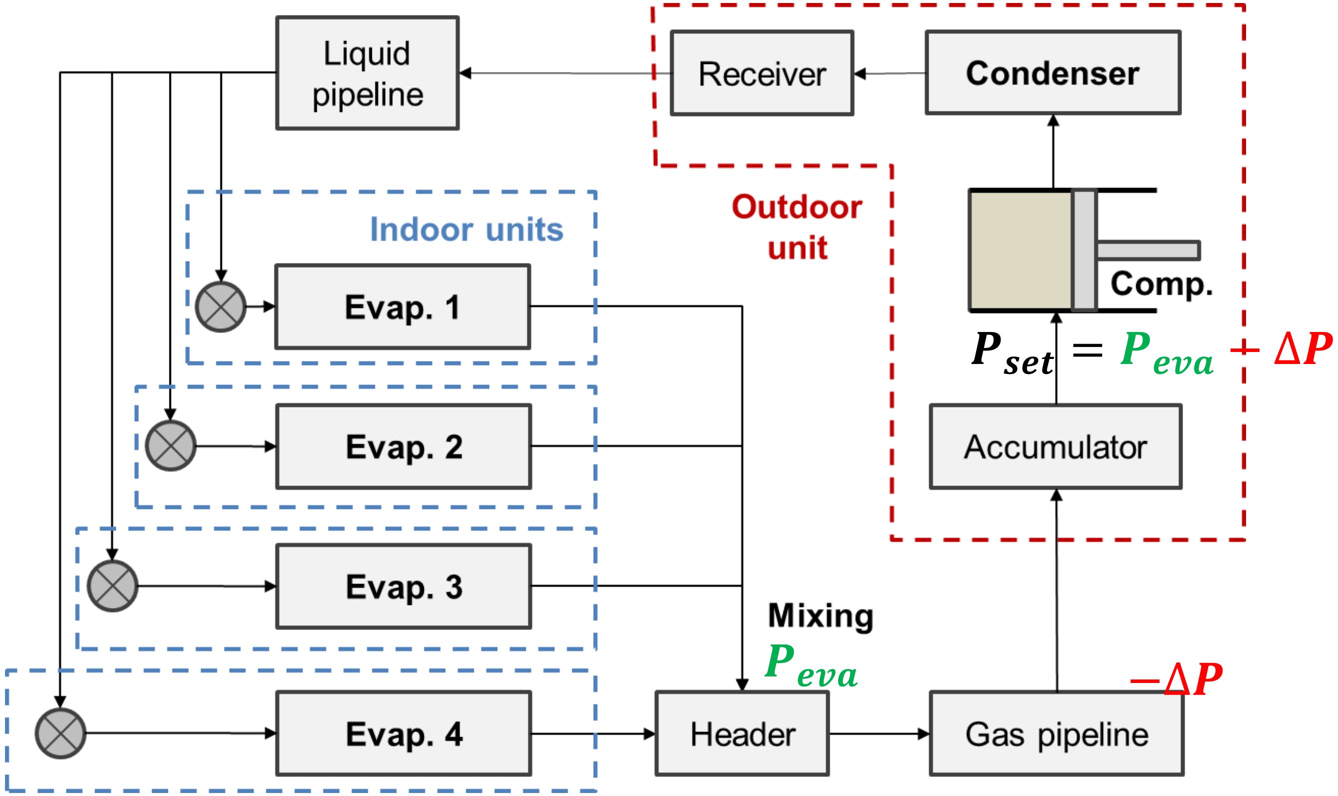


Fig. A schematic diagram of a dynamic model developed by simplifying a actual VRF system

1. T_{eva} is the evaporator saturation temperature, so it can be converted to saturation pressure P_{eva}

$$P_{eva} = P_{eva}(T_{eva})$$

2. Derive differential pressure ΔP to compressor inlet using the overall expected cooling capacity

$$\Delta P = f(k_2, \sum Q_{cool}) = k_2 \times (\sum Q_{cool})^2$$

3. $P_{set} = P_{eva} - \Delta P$

➡ Target P_{set} can be found

3) Feedback algorithm

From $Q_{cool,target,i} = \dfrac{Q_{load,sens,i}}{\mathrm{Margin}}$

Initial value of Margin is 0.75

$\Leftrightarrow$ Let the ROR be 0.75

If ROR < 0.75
: cooling capacity is still excessive
$\Rightarrow$ increase Margin $\Rightarrow$ decrease $Q_{cool,target}$

If ROR > 0.75
: cooling capacity is insufficient
$\Rightarrow$ decrease Margin $\Rightarrow$ increase $Q_{cool,target}$

$\therefore$ $error = \mathrm{ROR} - 0.75$ should be fed back

$$\mathrm{Margin} = \frac{K_I}{\Delta t}\int_{t-\Delta t}^{t} (\mathrm{ROR} - 0.75)\, dt + 0.75 + K_2$$

I term: Set Δt to 1,200 seconds to reflect
the average of the errors for 20 minutes

K_I : Gain value. $K_I = -0.5$ in this study

K_2 : Conditional control input. The initial value is 0.

$\mathbf{if}\ T_{room} > T_{target} + 1\ \mathbf{or}\ \mathrm{RH}_{room} > 80\%$

$K_2 = K_2 - 0.1$

$P_{set} = 900\ \mathrm{kPa}$

Simulation conditions

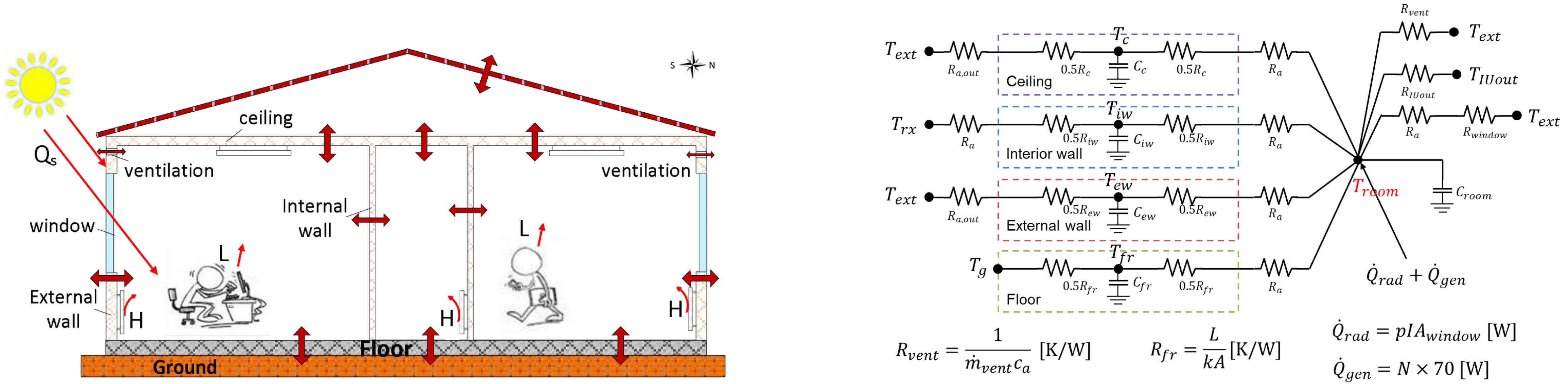


Fig. Heat flow in a multi-room building and thermal resistance circuit [5]

- **Four office rooms arranged side by side** was set as a simulation target.
- An **indoor space cooling load model was developed** and combined with the VRF model. Thermal load models include heat conduction from walls, floors, ceilings, and windows, solar radiation, heat and mass transfer by ventilation, occupancy, and **cooling by VRF (2.5 HP cooling unit in each room).**

[5] Ciprian Lapusan et al., 2016, Energy Procedia, DEVELOPMENT OF A MULTI-ROOM BUILDING THERMAL MODEL FOR USE IN THE DESIGN PROCESS OF ENERGY MANAGEMENT SYSTEMS

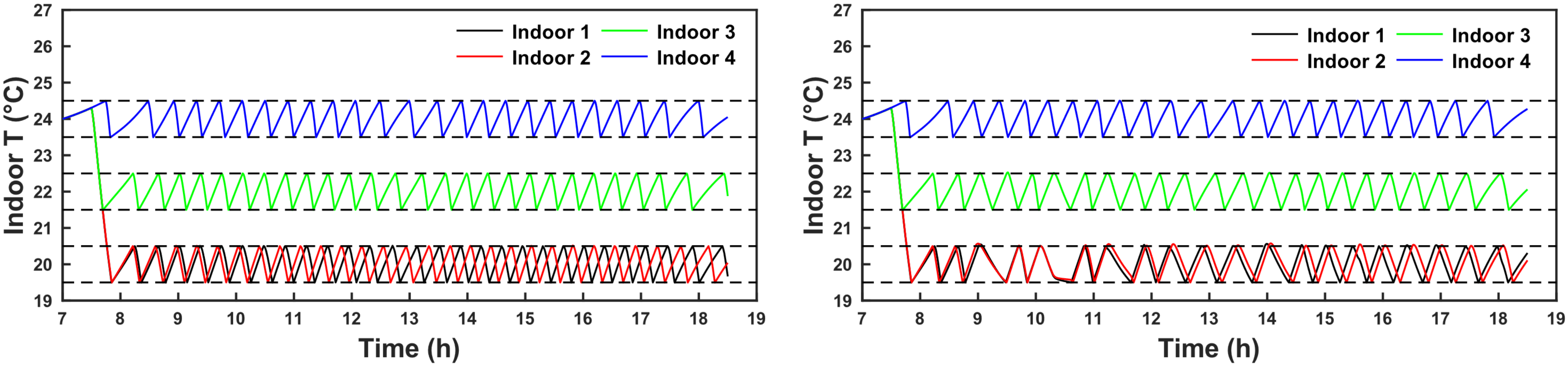
Simulation conditions

Table. Controller performance simulation conditions

	Outdoor air	Occupancy	Indoor set temperature [℃]	Note
1	DB 35.1℃ WB 24.7℃	6	For all the indoor zone, 21℃	Small cooling load case
2	DB 35.1℃ WB 24.7℃	6	20 / 20 / 22 / 24℃	Different indoor unit set temperatures
3	DB 35.1℃ WB 24.7℃	$4+\alpha^*$	21 / 22 / 23 / 24℃	Cooling load varies from room to room
4	DB 29.6℃ WB 25.5℃	$8+\alpha^*$	21 / 22 / 23 / 24℃	Large cooling load case

α^* : 1 to 2 kW of cooling load was added to each indoor zone by time (room 2: 13:00-15:00, room 3: 9-11:00, room 4: 11:00-13:00)

Simulation results
- Different set temperature

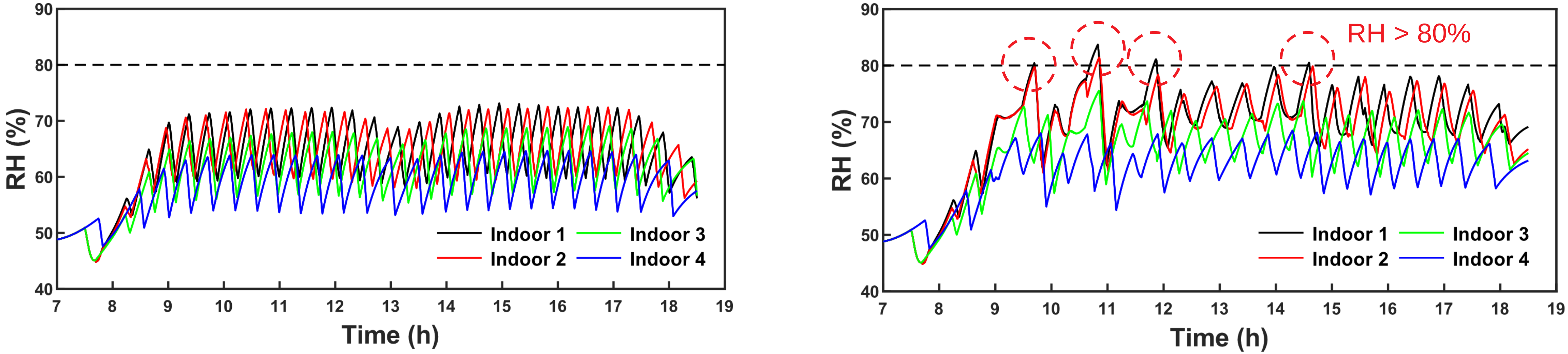


(a) (b)

Fig. Indoor air temperature change in (a) no control (b) controlled case

- With control, 1) the room temperature was maintained within a certain range 2) Ref. flow On/Off frequency decreased

Simulation results
- Different set temperature



(a) (b)

Fig. Indoor air RH change in (a) no control (b) controlled case

- Except for a few points, RH remained below 80%.

Simulation results
- Different set temperature

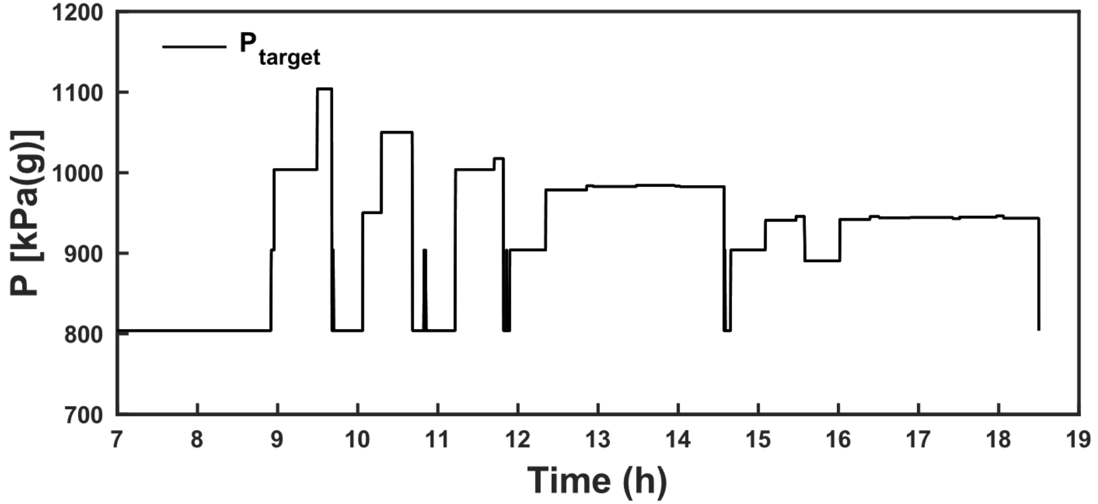


Fig. Change in set pressure value (default is 900 kPa)

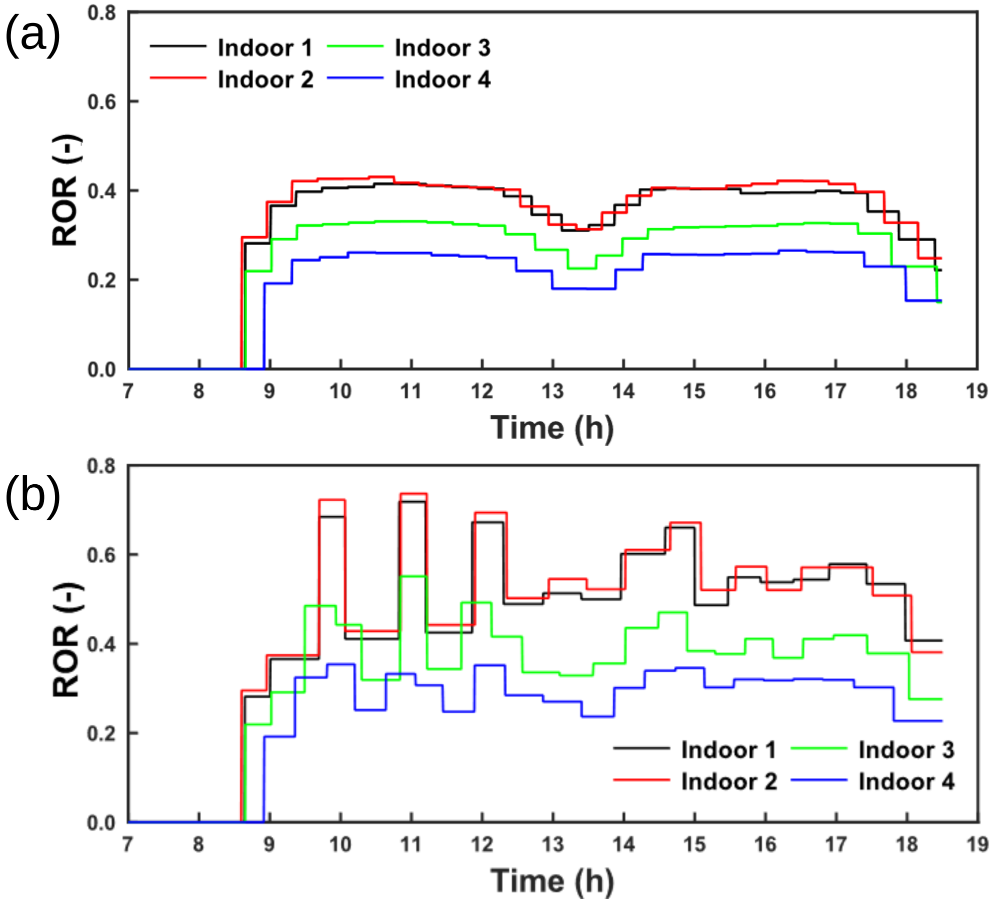


Fig. ROR value change in
(a) no control (b) controlled case

Simulation results

Table. Controller performance simulation results

#		Changes with Controller Application			
	Note	Q_{cool}	$(W_{Comp} + W_{fan})$	COP	ΔRH
1	Small cooling load case	-7%	-22%	+22%	+8.3%p
2	Different indoor unit set temperatures	-5%	-29%	+31%	+4.3%p
3	Cooling load varies from room to room	-6%	-27%	+29%	+5.6%p
4	Large cooling load case	-1%	-2%	+1%	+0.8%p

Conclusions

- **Dynamic Model Development**

  - A dynamic model for the VRF system was developed on Simulink by the moving boundary method
  - Modeling parameters based on actual system performance experiments were derived to increase model accuracy

- **Set Pressure Controller Development**

  - This study suggested 1) cooling load finding algorithm 2) evaporation pressure adjusting algorithm, and 3) feedback algorithm
  - The proposed control algorithm is expected to reduce energy consumption by up to about 30% depending on operating conditions and reduce the frequency of Ref. flow on/off.