Case Studies in Thermal Engineering 43 (2023) 102775

Contents lists available at ScienceDirect

Case Studies in Thermal Engineering

journal homepage: www.elsevier.com/locate/csite

Comparative analysis of sizing procedure for cooling and water heating cascade heat pump applied to a residential building

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A R T I C L E I N F O

Handling Editor: Huihe Qiu

Keywords:
Cascade heat pump
Multi-functional system
Sizing procedure
Residential building
Intermediate heat exchanger
Comparative analysi

A B S T R A C T

An appropriate design strategy suitable for the purpose of a building should be considered to apply a cascade heat pump system for multiple functions to buildings. The purpose of this study is to propose a sizing process and evaluate a capacity calculation simulation of air-to-air/water cascade heat pump systems applied in residential buildings that supply indoor cooling and hot water. A simulation study was performed by dividing the capacity calculation method suitable for the building conditions into two cases when this multi-functional cascade heat pump system was applied to a residential building. Case 1 was a sizing method that considered a low-cycle heat pump for indoor cooling. The existing heat pump sizing method for the hot-water supply used in the previous study was set to Case 2. The heat pump simulation and heat calculations were conducted based on thermodynamic theoretical equations. A simulation tool capable of dynamic thermal load analysis was used for model building. Detailed building information was entered in accordance with design standards. The results showed that Case 1 completely satisfies both the cooling and hot-water supply performance of the heat pump compared to Case 2. Based on Case 1, when the temperature of the intermediate heat exchanger was 10 °C higher than the outdoor temperature, the air volume of the indoor unit and the outdoor unit was found to be 941 m^3/h at an appropriate air volume rate. In addition, the energy consumption was also the lowest at 2.29 kW h, and the maximum system COP of 6.0 was achieved.

Nomenclature

COP	coefficient of performance [−]
h	Enthalpy [kJ/kg]
$\dot{m}$	Mass flow rate [kg/s]
P	power [kW]
PLR	part load ratio [−]
ΔP	pressure [Pa]
$\dot{Q}$ Heat rate [kW]T Heat rate [kW]TTemperature [°C]	
$\dot{V}$	volume flow rate [m^3/s]

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https://doi.org/10.1016/j.csite.2023.102775
Received 15 November 2022; Received in revised form 9 January 2023; Accepted 23 January 2023
Available online 24 January 2023

$\dot{W}$ power [kW]Greek Symbols

ε	efficiency [−]
η	effectiveness [−]
ρ	density [kg/m^3]

Superscripts

C	condenser
E	evaporator

Subscripts

comp	compressor
cond	condenser
evap	evaporator
HC	high cycle
hw	hot water
hx	heat exchanger
IDU	indoor unit
IHX	intermediate heat exchanger
LC	low cycle
ref	refrigerant
sa	supply air
tw	tap water

Abbreviations

AC	air-conditioning
COP	coefficient of performance
EEV	electric expansion valve
HW	hot water supply
LPH	liter per hour
ODU	outdoor unit

1. Introduction

With increasing fossil fuel usage in building air-conditioning systems, air-source heat pump systems are in the spotlight as an alternative HVAC system to reduce carbon emissions [1]. An air-source heat pump system can treat zone air conditions with high energy efficiency using electricity. In addition, boilers that supply hot water to buildings are also actively being studied using a heat pump system to supply high-temperature heat without using fossil fuels [2,3]. Therefore, when an eco-friendly renewable energy generation system is combined with a heating and cooling heat pump, it is possible to supply indoor air-conditioning and hot water with its own power generation system [4]. This potential application has led to increased research interest in multi-functional heat pumps capable of simultaneously solving heating, cooling, and hot water supply problems in buildings.

A multi-functional heat pump is a system that supplies indoor cooling, heating, and hot water using an evaporator and condenser in the heat pump system. Ji et al. [5] proposed a multi-functional single heat pump system in which an evaporator absorbs heat from indoor air and a condenser releases heat to the indoor air or tap water for zone and water heating. The system proposed in this study has three operational modes. The hot water supply and indoor heating functions are divided into separate operation modes—hot water supply only and indoor heating only—because the condenser function overlaps. As the evaporator is responsible for zone cooling, it can be supplied with hot water in the condenser simultaneously. Liu et al. [6,7] proposed a multi-functional heat pump system capable of performing indoor cooling, heating, and hot water supply functions that can utilize wastewater sources and air-source heat pumps in a single stage. An experimental study was conducted on four cases using either the air source or the wastewater source alone or in series and parallel to the heat sources, and the coefficient of performance (COP) of the system was improved. Cai et al. [8,9] proposed various operating methods for a multi-functional heat pump system combining a single-stage heat pump with a solar source, and numerical analysis and experimental validation were conducted. These studies used a single heat pump system for multi-functional performance. This system improved COP results compared to the existing single-purpose heat pump system because the heat pump systems simultaneously utilized both heat absorbed in the evaporator and heat dissipated in the condenser. However, heating tap water using a single-stage heat pump requires more compression energy from the compressor to supply a high temperature of 60 °C, leading to excessive energy consumption by the compressor, reducing the COP of the single heat pump system and deteriorating system performance.

A heat pump system with cascade-type has been investigated to supply hot water while overcoming the limitations of the single-stage heat pump. In a cascade heat pump, two independent heat pump cycles are connected in series through an intermediate heat exchanger. As the temperature is raised in two cycles to supply high-temperature heat from a low-temperature heat source, the load of the compressor in each cycle can be reduced compared with a single heat pump—a major feature of energy-saving performance and

COP improvement. A previous study experimentally revealed that cascade heat pump water heating systems have superior energy efficiency compared to a single heat pump for supplying hot water [10–14]. Many studies have been conducted on applying a cascade heat pump system capable of supplying sufficient heat as a multi-functional system. Jung et al. [10] conducted a comparative analysis of a single heat pump for a hot water supply and a cascade heat pump, experimentally, and found that the performance of the proposed system was superior. Kim et al. [11,12] investigated the temperature increase effect of a cascade heat pump for supplying domestic hot water and aimed to determine the optimum temperature value for the intermediate heat exchanger. Wu et al. [13] performed a performance analysis of a cascade heat pump for high-temperature hot-water supply using a multi-objective optimization methodology. Xu et al. [14] performed an economic and experimental performance analysis by applying a cascade heat pump system in a high latitude. All these studies focused on a heat pump system with a single purpose, such as hot water supply, for a cascade heat pump system. A previous study experimentally revealed that the cascade heat pump water heating system has superior energy efficiency compared to a single heat pump for supplying hot water supply heat.

In contrast, several studies have been conducted to install one more heat exchanger on the low-cycle side of the cascade heat pump and apply it as a multi-functional heat pump. These systems provide heating and cooling in a low cycle and hot-water supply in a high cycle. Boahen et al. [15,16] experimentally analyzed the performance of a water-to-water cascade heat pump system using a geothermal heat pump during cooling-hot water supply mode and heating-hot water supply mode operations in summer and winter. In this study, cold and hot water supply performances were analyzed under various water temperature conditions. Jung et al. [17] proposed a cascade heat pump system that connected a high-cycle to an air source heat pump system for hot water supply and conducted a performance optimization study based on the experiment. In this study, the heating, cooling, and hot water supply performance and COP analysis were performed by controlling the opening rate of the electric expansion valve and rotation speed of the compressor. However, previous studies have only focused on simple performance analysis or system applicability. For the actual application of this cascade heat pump to a building, an appropriate capacity calculation process for the system for application in the building unit should be performed. Residential buildings are constantly subjected to a load of hot water supply, followed by a load of cooling in summer. Therefore, a more realistic system capacity calculation and system sizing design can only be performed when a design process that considers various design conditions in an actual residential building is established. However, studies on cascade heat pump applications in actual building units are insufficient.

Consequently, this study aims to propose a design process for a multi-functional cascade heat pump system to provide cooling and hot water supply to a residential building. The detailed building cooling load and heating load for hot water were determined based on a transient thermal calculator program and standards. A thermodynamic analysis was conducted to build the system sizing process based on air-conditioning and hot-water supply constraints. From the proposed design process, energy consumption analyses and system performance are conducted, and the system is compared with the existing sizing analysis process. Using the analyzed design method, suitable refrigerant temperature settings were discussed with a parametric study of the energy and coefficient of performance.

2. System description and methodology

2.1. Air-to-air/water cascade heat pump system

A cascade heat pump system can provide multiple functions such as indoor cooling, heating, and hot water supply for residential buildings [17]. The proposed cascade heat pump system is configured such that two heat pump cycles are connected in a half-series by an intermediate heat exchanger. The low-cycle (LC) of the proposed system consists of an indoor unit (IDU), an outdoor unit (ODU), a compressor, an electric expansion valve (EEV), and a 4-way valve. The LC heat pump cooled the zone air for indoor air-conditioning. A high-temperature and high-pressure refrigerant was discharged from the compressor, and the intermediate heat exchanger (IHX) was configured in parallel with the ODU to use the refrigerant in this state as a heat source on the high-cycle (HC) heat pump. This

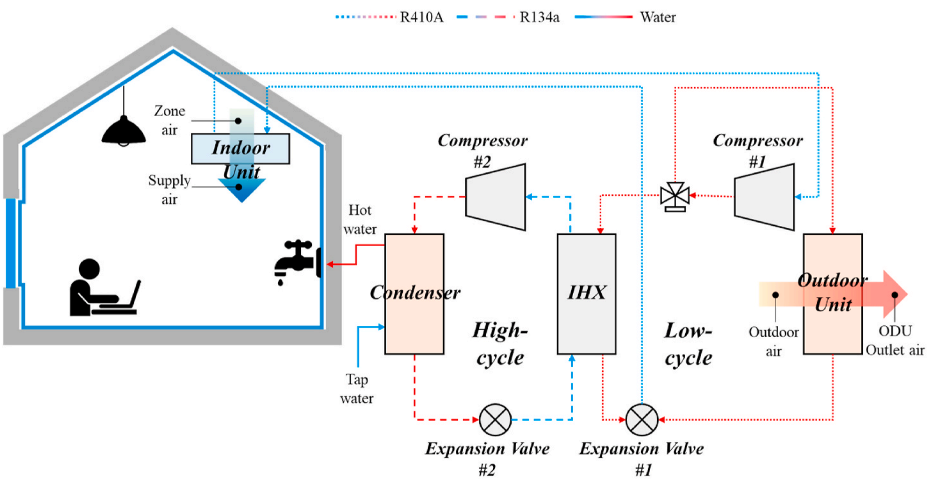


Fig. 1. Schematic diagram of air-to-air/water cascade heat pump system.

high-temperature and high-pressure refrigerant flows into the IHX, and heat exchange occurs between the refrigerants of the HC and LC. The HC heat pump system includes a compressor, EEV, and condenser, similar to the LC heat pump system. Because the HC system supplies hot water [10–14], the cycle is operated only as a one-way flow. In the HC condenser, the required hot water was supplied to the target building zone through refrigerant–water heat exchange. Thus, the entire system operates as the air-to-air/water heat pump system for air cooling, air heating and water heating. The cooling and heating modes can be switched by operating a 4-way valve. However, only the cooling mode was considered in this study.

2.2. System components

2.2.1. Indoor unit

Fig. 1 shows the cooling and hot water supply modes of the proposed cascade heat pump system. When a cooling load occurs in a building, the LC heat pump is operated for indoor cooling. At this time, the IDU operates as an evaporator that absorbs heat by cooling the indoor air. When designing a heat pump system for indoor cooling, the main parameters for sizing the system are the temperature and relative humidity of the zone air and supply air [18]. The ASHRAE recommendation provides a standard to satisfy the thermal comfort of residents in summer [19,20], and the indoor conditions in summer are usually maintained at 26 °C and 50% relative humidity. To maintain the zone temperature at 26 °C in summer, cold air must be supplied, and many air conditioning standards and previous literatures set the supply air temperature and humidity conditions to 15 °C and 80% [21–23]. To obtain the mass flow rate ($\dot{m}_{IDU.sa}$) for air cooling, the design supply air volume was calculated using the zone heating load ($\dot{Q}_{zone}$) and enthalpy difference between the zone air (h_{zone}) and supply air ($h_{IDU.sa}$) (Equation (1)). Meanwhile, the refrigerant temperature at evaporator was set to 15 °C for cooling the supply air.

$$\dot{m}_{IDU.sa} = \frac{\dot{Q}_{load.zone}}{h_{zone} - h_{IDU.sa}} \tag{1}$$

2.2.2. Outdoor unit

For continuous operation of the heat pump, the heat absorbed by the evaporator must discharge heat outside the cycle [24]. An outdoor unit dissipates heat outdoors during the cooling operation, and the outdoor unit is operated as a condenser. The main parameter for designing the outdoor unit air volume is the outlet temperature condition, which is 10 °C higher than the ambient air temperature for releasing enough heat as the condenser setting temperature [25,26]. The outdoor air volume was calculated using a heat exchange equation between the outdoor temperature and condenser set temperature, and it was assumed that the heat exchanger used a finned-tube heat exchanger with an efficiency of 0.8.

2.2.3. Condenser for hot water supply

Residential buildings require a continuous hot water supply daily, and high-temperature heat sources are required to heat the hot water. For a high-temperature heat supply with low energy, the proposed system receives a low-cycle refrigerant heat source and heats tap water by raising the temperature during the high cycle. A high-cycle heat exchanger for heating hot water was used as a condenser. The parameters necessary for the design of hot water heating are the target water temperature, the water temperature, and the supply of hot water. According to the ASHRAE standards [27], the target temperature for hot water is 45 °C or higher, which is sufficient for indoor hot water supply (e.g., hand washing, showers and tubs, residential dish washing, and laundry). To suppress the generation of bacteria during hot water supply, water may be heated to 60 °C [28]. In contrast, the water temperature in summer is maintained at a certain value because of thermal interference from the surrounding area as water moves along the underground water pipe or wall. In this study, to block the generation of bacteria, the hot water heating temperature was assumed to be 60 °C, and the water temperature was assumed to be 15 °C [29]. Regarding the hot water supply standard, the hot water supply schedule necessary for residential buildings was also applied according to the ASHRAE standards [30]. Accordingly, the required hot water supply load was calculated using the following equation (Equation (2)):

$$\dot{Q}_{load.hw} = \dot{V}_{hw}\rho_{tw}(T_{hw} - T_{tw}) \tag{2}$$

2.2.4. Fan

The equation for calculating the energy consumption of each fan is as follows: (Equations (3)–(5)). The fan efficiency was set to 50% [31]. At this time, 0.25 kPa in the coil and 0.01 kPa in the duct were applied to the pressure difference generated while operating the fan [32].

$$PLR_{fan} = \frac{\dot{V}_{zone.current}}{\dot{V}_{zone.design}} \tag{3}$$

$$P_{fan.design} = \frac{\dot{V}_{zone.design} * (\Delta P_{coil} + \Delta P_{duct})}{\eta_{fan}} \tag{4}$$

$$P_{fan} = \left(0.0013 + \left(0.147 * PLR_{fan}\right) + \left(0.9506 * PLR_{fan}^2\right) - \left(0.0998 * PLR_{fan}^3\right)\right)P_{fan.design} \tag{5}$$

2.3. Methodology

To calculate the proposed system capacity when the system is applied to a residential building, the capacity must be determined by considering not only the system but also various conditions applied to the residential building. To evaluate the cooling and hot water supply performance and COP, a detailed simulation study was conducted to evaluate the system capacity and performance by applying the following methodology. The research methodology applied in this study was as follows.

(1) Application of the heat flow rate criteria for each area where building simulations are possible.
(2) Application of indoor temperature and humidity conditions and hot-water schedules.
(3) Use of a building load simulation tool capable of calculating the transient to derive the summer building heat load.
(4) Calculation of the cooling and hot water supply performance of the cascade heat pump system through a thermodynamic analysis.
(5) Capacity calculation using the proposed strategy and a comparative simulation with a previous study after establishing the design conditions of the cascade heat pump.
(6) Investigation of the satisfaction achieved by the design condition and calculation of the COP according to the changes in the system setting conditions.

3. Sizing procedure

3.1. Building thermal load determination

3.1.1. Building information

In this study, a building that requires both an indoor cooling load and a hot water supply load was required. The factors that affect the heat load of a building include climate outside the building, indoor occupancy schedules, and heat generation [30]. The building used in this study was assumed located in Seoul, South Korea. The in-room schedule and hot water supply schedule were set to be applied regularly every week during the simulation period, after applying the weekly schedule. The residential schedule and required hot water supply schedule for residential buildings were applied. The model building information based on government statutes is summarized in Table 1, circumstantially [33].

3.1.2. Thermal load calculation

To calculate the heat load of the building, the thermal peak loads for a model residential building were estimated using a transient simulation calculation program [34]. The annual heat load generation was calculated for the load derivation period. The system design process was performed based on the building load that occurred on the relevant day by determining the design-day in which the cooling load and the hot water load were the highest during this period.

3.2. System sizing process

In the previous section, the heat pump sizing strategy and building thermal load calculation were described. The process of applying the cascade heat pump to a building must be as follows to properly size the system. First, building information was determined. In this step, the volume of the yarn, performance information of the wall and window heat flow rate, and number and schedule of occupants were determined. Based on the determined building input value, the required indoor cooling load and hot water supply load were calculated. Subsequently, a heat pump sizing simulation was performed based on the derived cooling load and hot water supply load, and sizing was performed by dividing cases 1 and 2. The evaluation criterion for selecting the two cases is to first check whether the indoor cooling and hot water supply can be sufficiently provided (Fig. 2). Subsequently, the COP and energy consumption of the heat pump were compared with those of the derived sizing.

3.3. Determination of sizing strategy

Two conditions were proposed to determine the appropriate capacity during the cooling-water supply operation of the A-A/W cascade heat pump. The condition of the two cases involves determining which of the two cycles constituting the cascade heat pump is

Table 1
Information of the residential building for determining the thermal load.

Category	Parameters	Values
Number of floors		3
Floor area		100 m^2
Occupants		15
Infiltration		0.6 1/h
Zone condition	Temperature	26 °C
	Relative humidity	50%
Supply air condition	Temperature	15 °C
	Relative humidity	80%
U-value	Outdoor wall	0.17 W/m^{2}K
	Floor	0.17 W/m^{2}K
	Roof	0.15 W/m^{2}K
	Window	1.12 W/m^{2}K
Heat gain	People	130 W/person (sensible, latent)
	Lights	7 W/m^2
	Equipment	0.4 W/m^2

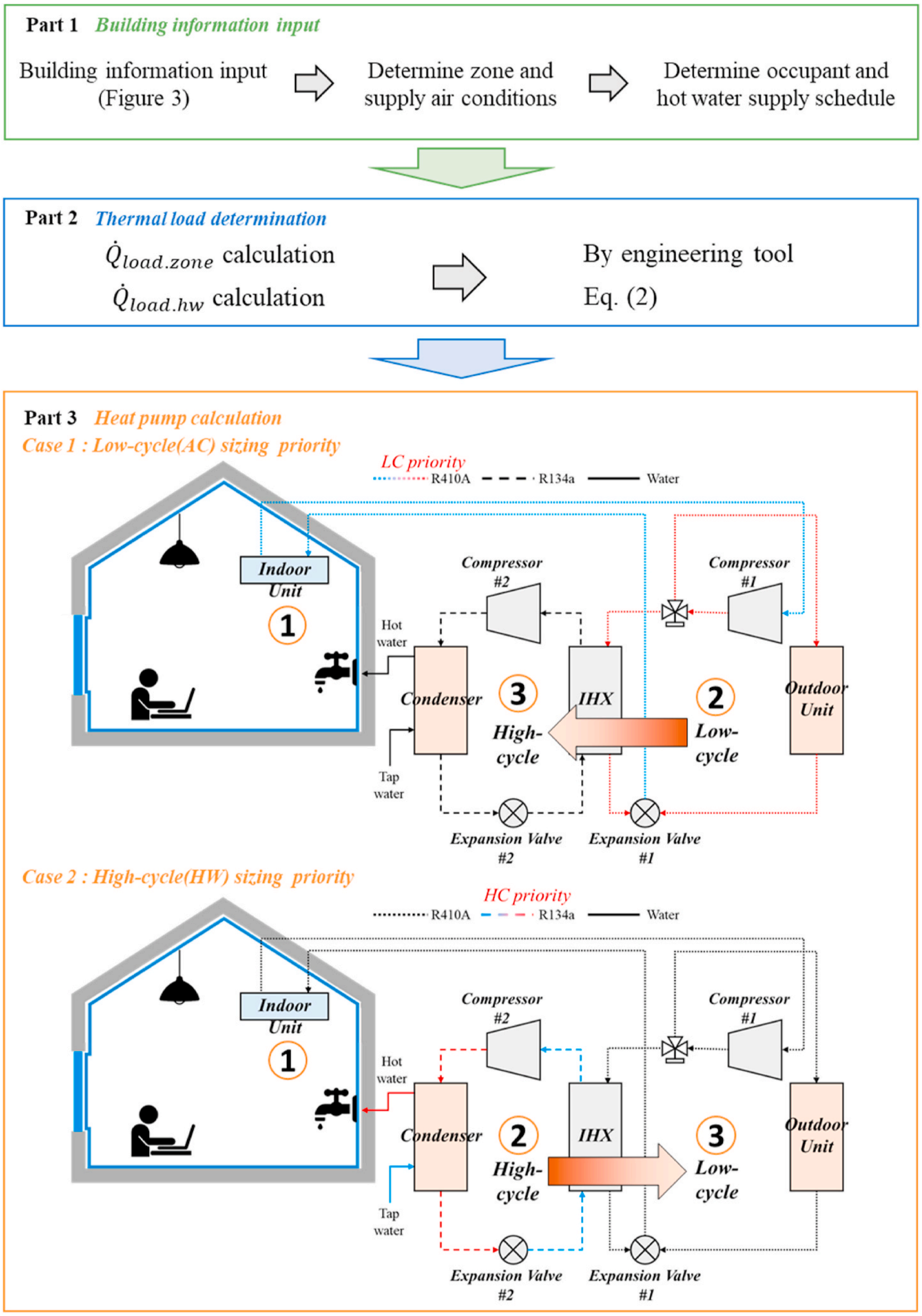


Fig. 2. Sizing procedure of the proposed system

preferentially calculated. This condition is applied is to obtain the function to be considered first when applying a cascade heat pump that uses both air cooling and hot water simultaneously to a building.

3.3.1. Case 1: low-cycle (AC) sizing priority

Case 1 is the process of preferentially sizing the cooling function in charge of the low cycle of the cascade heat pump. The overall process is as follows: First, the door unit that is the target of the low-cycle is calculated, and then, low-cycle heat pump analysis is performed. Then, the heat dissipated in a low-cycle condenser is calculated, and a heat source is supplied from the IHX to the high cycle. After performing the high-cycle heat pump analysis, the amount of heat supplied to the hot water load is calculated. A flow chart is shown in Fig. 3.

3.3.1.1. Zone thermal load determination. First, the air volume of the door unit is calculated. At this time, the air volume is calculated based on the enthalpy using Equation (1), introduced in Section 2.2.1. When the inlet air meets the cold evaporator during indoor

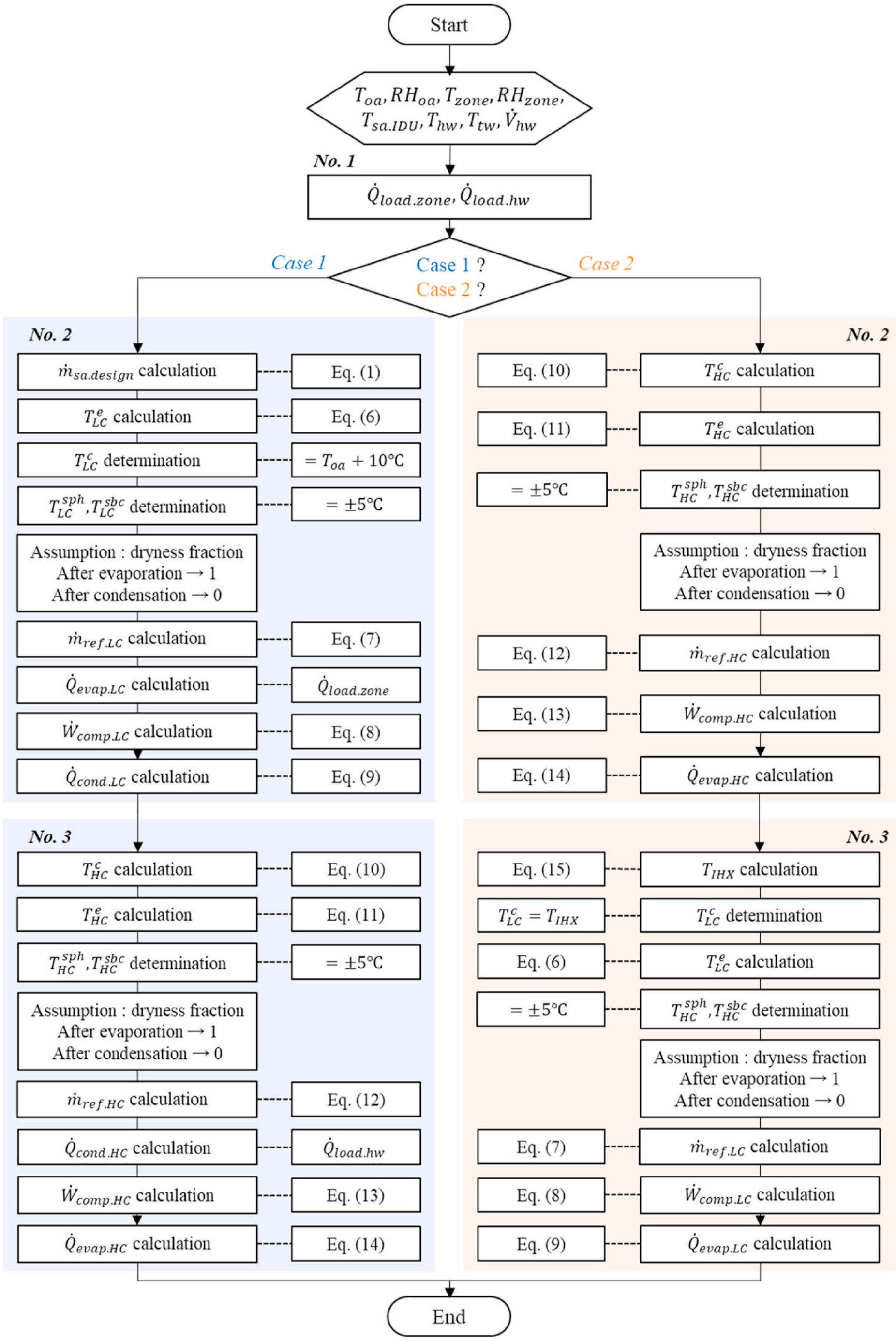


Fig. 3. Flow chart of the cascade heat pump capacity size calculation for residential buildings.

cooling, the air meets the dew point, and condensation humidity can occur. Therefore, the reason for calculating enthalpy is that both sensible and latent heat must be considered. Thereafter, the refrigerant temperature is calculated based on the calculated air volume (Equation (2)):

$$T_{LC}^{e} = T_{zone} - \frac{T_{zone} - T_{sa}}{\varepsilon_{hx}} \tag{6}$$

The heat-exchange efficiency of the heat exchanger is assumed to be 0.8 [35]. As a refrigerant undergoes a phase change in an evaporator [36], the evaporation temperature of the refrigerant is calculated by the flow rate, specific heat, indoor heat load, and indoor inlet air temperature.

3.3.1.2. Low-cycle heat pumps. The interpretation of the system cycle is based on thermodynamic theory. As an assumption for analysis, the refrigerant temperature (T^{c}_{LC}) setting value required for the outdoor unit was set to be 10 °C above the ambient air dry-bulb temperature condition because the outdoor unit of the system must dissipate heat smoothly when operating with a condenser. Second, the superheating and subcooling were set to 5 °C, respectively [37–39]. Finally, when the refrigerant is in a phase change, it is assumed that the phase change occurs as a complete gas in the evaporator and as a complete liquid in the condenser. Based on the above assumptions, the amount of refrigerant required for indoor cooling in the low cycle was calculated using the enthalpy difference in the evaporator compared to the amount of indoor load (Equation (7)).

$$\dot{m}_{ref.LC} = \frac{\dot{Q}_{load.zone}}{\Delta h_{evap.LC}} \tag{7}$$

After the refrigerant amount was calculated, the energy consumption of the compressor and heat dissipation of the condenser were calculated using thermodynamic theoretical equations (Equations (8) and (9)) because the required refrigerant temperature of the evaporator and the required refrigerant temperature of the condenser are determined, and the states during evaporation and condensation of the refrigerant are assumed; thus, the enthalpy difference can be identified by displaying it on the P-h line. Therefore, the heat pump can be calculated based on the aforementioned order.

$$\dot{W}_{comp.LC} = \dot{m}_{ref.LC} \Delta h_{comp.LC} \tag{8}$$

$$\dot{Q}_{cond.LC} = \dot{m}_{ref.LC} \Delta h_{cond.LC} \tag{9}$$

3.3.1.3. High-cycle heat pumps. The interpretation of the high-cycle heat pump is similar to that of the low-cycle heat pump. First, the condenser temperature was calculated as the target temperature of the high-cycle heat pump (Equation (10)). The condenser temperature for the hot water supply is a temperature value capable of satisfying the hot water supply load by heat exchange with time water. Meanwhile, the IHX, which can directly exchange heat between the high-cycle and low-cycle, serves as an evaporator for the high-cycle. As low-cycle heat should be received, the set temperature on the evaporator side of the high cycle should be set 5 °C lower than the condensation temperature of the low cycle to ensure that the heat moves from the low-cycle to the high-cycle side (Equation (11)) [40].

$$T^{c}_{HC} = T_{tw} + \frac{T_{hw} - T_{tw}}{\varepsilon_{hx}} \tag{10}$$

$$T^{e}_{HC} = T_{IHX} - 5^{\circ}C \tag{11}$$

In addition, when the refrigerant is in phase change, it is assumed that the phase change occurs as a complete gas in the evaporator and as a complete liquid in the condenser. Based on this assumption, the amount of refrigerant required in a high cycle to circulate the amount of refrigerant by the amount of heat that may receive all heat dissipation in a low cycle is calculated. In other words, the high-cycle evaporation heat is equal to the low-cycle condensation heat, and the refrigerant amount is calculated by dividing the evaporation enthalpy difference by the corresponding heat amount (Equation (12)). After the refrigerant amount is calculated, the energy consumption of the compressor and heat dissipation in the condenser are calculated using thermodynamic theoretical equations (Equations (13) and (14)).

$$\dot{m}_{ref.HC} = \frac{\dot{Q}_{load.hw}}{\Delta h_{cond.HC}} \tag{12}$$

$$\dot{W}_{comp.HC} = \dot{m}_{ref.HC} \Delta h_{comp.HC} \tag{13}$$

$$\dot{Q}_{evap.HC} = \dot{m}_{ref.HC} \Delta h_{evap.HC} \tag{14}$$

3.3.2. Case 2: high-cycle (HW) sizing priority

Case 2 is a process of preferentially sizing the hot-water supply function in charge of the high-cycle cascade heat pump, contrary to the order of Case 1. The overall process is as follows: first, the parameter for the hot water supply, which is the target of the high-cycle, is calculated, and then, high-cycle heat pump analysis is performed. Subsequently, the heat absorbed by the high-cycle evaporator is calculated, and the heat source required in the low-cycle of the IHX is calculated. After performing low-cycle heat pump analysis, the amount of heat that could be supplied to the interior unit is calculated.

3.3.2.1. Zone thermal load determination. Indoor water heating rate can be determined by multiplying the required hot water supply, specific heat of hot water, and temperature difference between the hot water and tap water (Equation (15)). The required hot water supply was set based on the standard hot water supply schedule for residential buildings with four residents as per the ASHRAE Standard. As described in Section 2.2.3, hot water was set to supply 60 °C water in which all *Legionella* bacteria were sterilized [28].

3.3.2.2. High-cycle heat pumps. The high-cycle heat pump analysis was interpreted based on thermodynamic theory in the same way as in Case 1. In Case 2, because enough heat to handle the hot water load must be supplied to the hot water side, the amount of refrigerant required in the high cycle is calculated through the enthalpy difference of the condenser relative to the hot water load (Equation (12)). The calculation of the required energy requirement and heat absorption amount of the compressor and evaporator are the same as those in Case 1.

3.3.2.3. Low-cycle heat pumps. Case 2 is a process that first solves the hot water supply load and then solves the heat absorption requirement in a high cycle with the amount of heat dissipated in a low cycle. Therefore, the condenser heat amount in the low cycle is the same as the required heat absorption amount in the high cycle, and heat exchange occurs in the IHX. The main purpose of Case 2 is a cascade hot water supply, so the system operates under a temperature condition that can save the most energy to achieve the hot water supply goal. Previous studies have investigated the optimal IHX temperature of an A-W cascade heat pump using R410A and R134a [11].

$$T_{IHX} = \sqrt{T^{c}_{HC} T^{e}_{LC}} \tag{15}$$

Therefore, in this study, simulation was conducted assuming that the temperature of IHX was 30 °C during the hot water priority sizing. Therefore, the condenser set temperature of the low cycle was 30 °C, and the evaporator of the high cycle was 5 °C higher than that of the low cycle temperature, so that heat exchange occurred. The set temperature of the evaporator was calculated using Equation (9) to determine the temperature for indoor cooling. Condenser, compressor, and evaporator calories in a low cycle were interpreted based on thermodynamic theory, as shown in Case 1. At this time, the amount of refrigerant circulating in the cycle was circulated by the amount of heat exchanged in the IHX.

3.4. Coefficient of performance

The coefficient of performance (COP) was derived for the entire system, including the LC heat pump, HC heat pump, total heat pump, and fan energy. The equations for the COP determination are as follows:

$$COP_{LC} = \frac{\dot{Q}_{evap.LC}}{\dot{W}_{comp.LC}} \tag{16}$$

$$COP_{HC} = \frac{\dot{Q}_{cond.HC}}{\dot{W}_{comp.HC}} \tag{17}$$

$$COP_{total.HP} = \frac{\dot{Q}_{evap.LC} + \dot{Q}_{cond.HC}}{\dot{W}_{comp.LC} + \dot{W}_{comp.HC}} \tag{18}$$

$$COP_{total.system} = \frac{\dot{Q}_{evap.LC} + \dot{Q}_{cond.HC}}{\dot{W}_{comp.LC} + \dot{W}_{comp.HC} + P_{fan,IDU} + P_{fan.ODU}} \tag{19}$$

4. Simulation result

4.1. Design load of each component

To derive the heat load value of the building, the heat load of the building model in the summer was calculated using a transient simulation tool capable of calculating the heat load. The simulation results showed that the maximum cooling load of the building assumed in this study was 3.58 kW, which is the sum of the results derived as 2.72 for the sensible load and 0.86 for the latent load. The load absorbed as much heat as the maximum cooling load from the evaporator in the IDU located indoors. The air volume required for the indoor evaporator to remove the heat load from the difference between the zone temperature and supply air set temperature was calculated to be 941.7 m^3/h. This result falls within the 800–1000 m^3/h category, which is the supply air volume of a general indoor air conditioner or DXA/C [41,42].

In contrast, 1.41 kW of hot water was derived, which is the amount of heat required to heat water by 15 °C of to the reference temperature of 60 °C based on 27 l/h of the residential building proposed by ASHRAE Standard [30].

Therefore, 3.58 kW of indoor cooling load was removed from the LC, and 1.41 kW of hot water load was supplied to the HC. The design loads of the building are listed in Table 2.

4.2. System performance and size determination

4.2.1. Refrigerant temperature

Indoor air cooling was performed at the IDU to handle the thermal load of the building. Therefore, as a result of calculating the required refrigerant temperature for cooling indoor air, the temperature on the evaporator side of the LC was found to be 12.3 °C. In contrast, as a result of calculating by applying the sizing strategy methods of Case 1 and Case 2 during system simulation, the condenser temperature of the LC heat pump in Case 1 was found to be 41.3 °C, which allows heat to move from the inside of the LC heat pump to the outside air by forming heat that is hotter than the outside air and discards condensation heat. In Case 2, the condenser temperature

Table 2
Result of building design load and stream flow rate.

Cooling load [kW]	Hot water load [kW]	Indoor air flow rate [m^3/h]	Hot water flow rate [l/h]
3.58	1.41	941.7	27

of the LC heat pump was 29.5 °C because the operation of the LC in Case 2 preferentially considers the supply of the HC heat source. Eventually, the LC was operated under optimal conditions to supply heat to the HC. Therefore, the condenser temperature of the LC was determined as the optimal temperature using thermodynamic theory.

Tap water is heated to handle the hot water load of the building on the condenser side of the HC. The required refrigerant temperature of the condenser of the HC was found to be 71.3 °C. Because the refrigerant used in the HC heat pump was R134a, it was possible to generate a higher temperature than that in the LC. The evaporator temperature in the HC heat pump was 5 °C lower than the LC condenser refrigerant temperature because it needs to be heated by the LC heat pump. Table 3 lists the refrigerant temperature results for each sizing strategy.

4.2.2. T-s graph

The refrigerant characteristics of each of the derived heat pumps are shown in the T-s diagram [43–46]. Fig. 4 shows the saturation curves of R410A, an LC refrigerant, and R134a, an HC refrigerant, in one figure simultaneously. The temperature and entropy calculated at each component point of cases 1 and 2 are shown in the corresponding saturation curves. The HC indicated by the red line was located above the LC, indicated by the blue line. According to the second law of thermodynamics, heat can be expressed as a product of the temperature and entropy difference, and the heat moves from high to low temperatures when a temperature difference occurs. Considering this basic thermodynamic law, in Case 1, the heat remaining after heat transfer from the condenser of the LC to the HC side may be radiated to the ODU side; however, in Case 2, the condenser refrigerant temperature of the LC may be lower than the outside temperature, and thus, it is impossible to radiate heat to the ODU side. Therefore, when operating and sizing in Case 2, the heat source supply at the LC should be calculated to provide only the amount of heat absorbed by the HC evaporator after performing the HC's hot water load treatment first for the system to run while maintaining heat balance.

In contrast, the temperature difference, entropy difference, and amount of heat transferred can be seen in the T-s diagram when the temperature of the LC evaporator is raised from 12.3 °C, which is the temperature of the LC evaporator, to 71.3 °C, which is the HC condenser temperature. However, the amount of load required for the compressor in each cycle, the amount of refrigerant flowing inside the cycle, and the enthalpy difference at each point must be calculated for heat pump sizing.

4.2.3. Refrigerant flow rate

Table 4 lists the enthalpy difference generated for each case and each component to calculate the amount of refrigerant flowing inside the heat pump. The enthalpy value was calculated using the temperature and entropy values for each point of each component. The enthalpy was calculated based on the physical property data of R410A and R134a embedded in the EES program.

The amount of refrigerant in each cycle was calculated by dividing the difference in the enthalpy value derived from the calorific value for each point calculated in the T-s diagram (Equation (7)). At this time, Case 1 had the purpose of preferentially treating the indoor cooling load of the LC, so the amount of refrigerant in the LC was calculated based on the difference between the heat value and enthalpy of the evaporator of the LC heat pump. Next, the amount of refrigerant flowing through the HCH heat pump was calculated based on the difference between the heat amount and enthalpy of the condenser, which supplies hot water. The purpose of system sizing in Case 2 is to prioritize the supply of hot water from the HC, so it was calculated through the difference between the heat amount and enthalpy of the condenser used to supply hot water from the HC heat pump, in contrast to Case 1. In Case 2, the amount of refrigerant in the LC should maintain a heat balance with the amount of heat to be absorbed by the evaporator in the HC; therefore, it was calculated based on the difference between the required heat of the IHX and the condenser enthalpy of the LC. The results for the amount of the derived refrigerant are shown in Fig. 5. Thus, the amount of refrigerant for supplying the heat of hot water in the HC is the same in both cases, but Case 1 requires approximately 3.5 times more in the case of LC.

4.2.4. Size of heat pump compressor

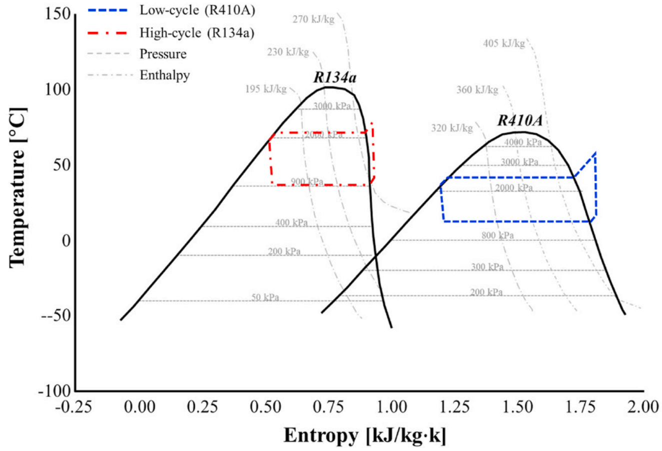
The required workload for the compressors of the LC and HC was calculated using the derived refrigerant amount. Table 5 showed results of the compressor size. To convert the calculated compressor load result into horsepower, the load in kW was multiplied by 0.7457. Thus, the size of the heat pump for each cycle to handle the indoor cooling load and the hot water load of the model building was 0.53 horsepower for LC and 0.21 horsepower for HC in Case 1. In Case 2, 0.06 horsepower was derived for LC and 0.33 horsepower for HC. The results of capacity calculation show that the LC calculation method (Case 1) requires a capacity 8.8 times larger in LC than the HC calculation method (Case 2), and in the case of HC, hot water could be supplied with a capacity of 63%.

4.3. Capacity of each sizing strategy

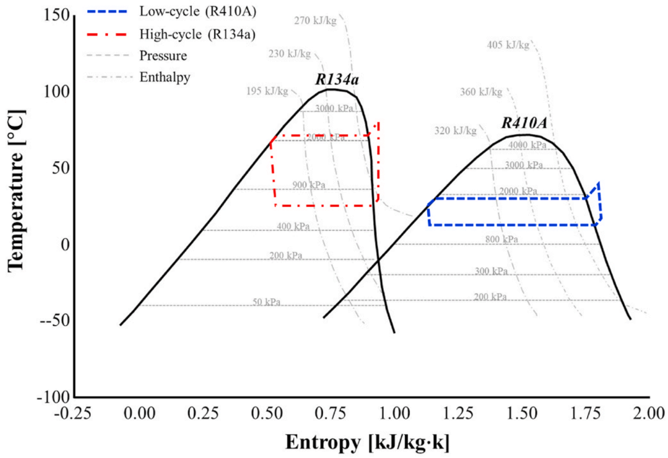
Fig. 6 shows the heat supply and heat-exchange performance of each component of the cascade heat pump for the indoor cooling and hot water supply. The calorific value of each component is a calorific value derived based on the conditions set for capacity calculation in this study and the indoor heat load and hot water load derived through the building simulation. Design conditions are indicated in bold. In addition, Case 1 is marked in blue and Case 2 is marked in orange to distinguish the capacity calculation strategies

Table 3
Temperature of the refrigerant at each heat pump heat exchanger.

	Low-cycle heat pump		High-cycle heat pump	
	Evaporator	Condenser	Evaporator	Condenser
Case 1	12.3 °C	41.3 °C	36.3 °C	71.3 °C
Case 2	12.3 °C	29.5 °C	24.5 °C	71.3 °C



(a) Case 1: Air-conditioning priority.



(b) Case 2: Hot water priority.

Fig. 4. T-s diagram of the simulation result for each case.

Table 4
Enthalpy of each heat pump component.

	Low-cycle heat pump			High-cycle heat pump		
	Δh_{evap} [kJ/kg]	Δh_{comp} [kJ/kg]	Δh_{cond} [kJ/kg]	Δh_{evap} [kJ/kg]	Δh_{comp} [kJ/kg]	Δh_{cond} [kJ/kg]
Case 1	171.4	18.9	190.2	125.7	15.6	141.2
Case 2	191.7	8.42	200.1	119.6	25.4	145.1

of the cascade heat pump. In Case 1, the LC absorbs the heat equivalent to the indoor cooling load from the IDU, and the refrigerant is distributed to the IHX and ODU sides after the compressor. The IHX side supplies 1.26 kW heat, which is the amount of heat required to transfer heat to the HC. Finally, the condenser of the HC supplies 1.41 kW heat. The remaining condenser heat from the LC is exchanged with external air on the ODU side to dissipate the remaining 2.71 kW of heat. In contrast, in Case 2, the hot water supply load of the HC is preferentially treated, and the amount of heat supplied to the condenser of the HC is the same as that in Case 1. However, considering the IDU side of the LC, unlike in Case 1, only 1.11 kW of heat was absorbed, implying that the set temperature value of the IHX is 29.5 °C, and the corresponding condenser temperature cannot be heat-exchanged with the outside air on the ODU side. The corresponding regions are marked in red. Therefore, the LC in Case 2 circulates only the amount of refrigerant that can supply heat to the HC, so that the indoor IDU absorbs less heat than in Case 1. Therefore, it can be observed that the system sizing strategy satisfies all the design conditions of the system is Case 1.

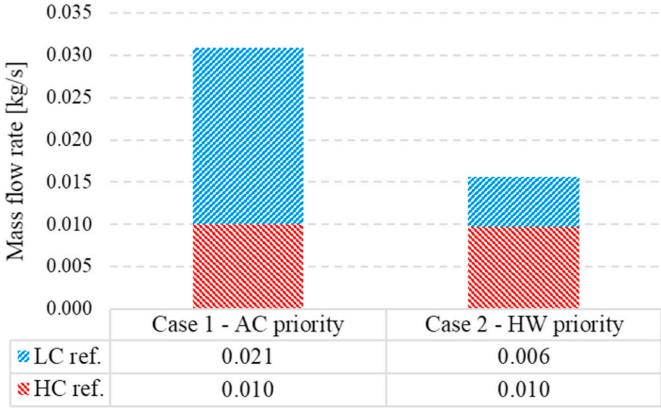


Fig. 5. Mass flow rate of the refrigerant for each heat pump cycle.

Table 5
Results of compressor size.

	Low-cycle compressor [HP]	High-cycle compressor [HP]
Case 1	0.53	0.21
Case 2	0.06	0.33

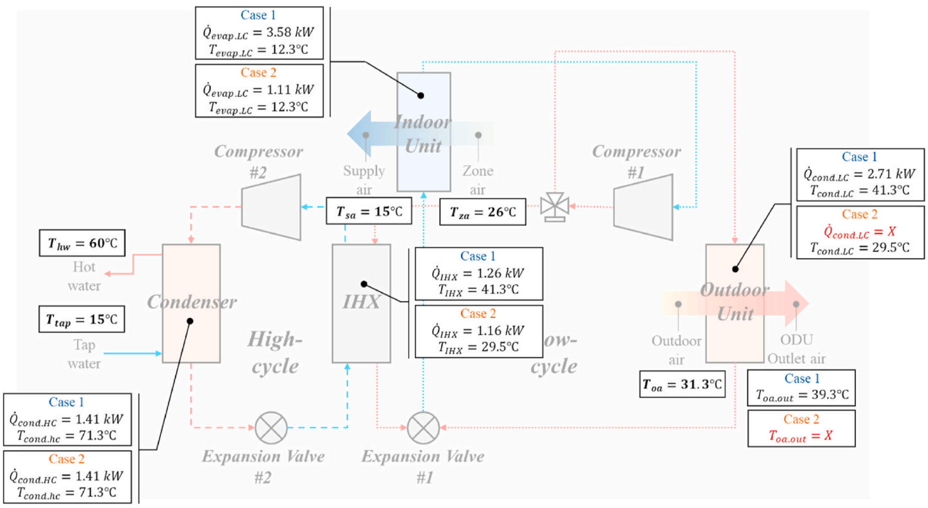


Fig. 6. Heat supply and heat-exchange performance of each component of the cascade heat pump

5. Discussion

5.1. Parametric study for refrigerant temperature

The results in the previous section are those of simulating cases 1 and 2 by applying the sizing strategy of the system differently, and accordingly, the largest difference is the temperature set value of the IHX. Case 1 prioritized the driving of LC, so 41.3 °C, which is a condenser temperature that can be radiated from ODU, was also applied to IHX. In contrast, Case 2 prioritized the driving of HC, so 29.5 °C, which is the condenser temperature of LC for optimal heat supply based on the theoretical formula, was applied to IHX. If so, system sizing was performed using the strategy of Case 1 based on the summer peak day, but a parametric study was conducted to check the change in system performance when the set temperature condition of the IHX was changed. The independent variable was the IHX temperature. The dependent variables are the IDU and ODU air volumes, which can be used to check the performance of the system, heat consumption of each unit, primary energy consumption, and COP. The temperature range of the intermediate heat exchanger was set from 25 °C close to the IHX temperature derived in Case 2 to 50 °C, which is 10 °C lower than the hot water supply temperature.

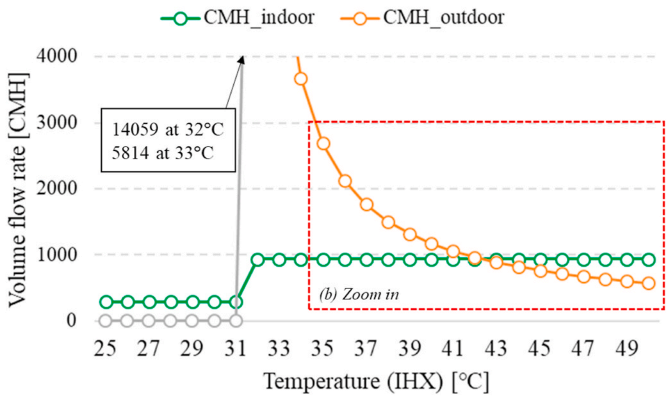
During the parametric study, the temperature of the outside air is 31.3C on the peak day. When the condenser temperature of the LC is between 25 and 31 C, the temperature of the condenser is lower than that of the outside air. Because of this, condenser heat cannot be dissipated from the ODU. On the other hand, as a result of the simulation in Section 4, the amount of absorbed heat required by the evaporator on the HC side is not large compared to the amount of heat generated by the condenser in the LC in Case 1. Therefore, if the Case 1 method is operated in the corresponding temperature range, even if the condenser heat of the LC is given to the HC side through the IHX, the heat cannot be dissipated to the ODU side because the temperature is lower than the outside air. So the remaining condensation heat cannot be treated. Therefore, Case 2 method which consideres preferentially to HC was operated from 25 °C to 31 °C, and Case 1 was operated to conduct a parametric study above 32 °C.

5.1.1. Air flow rate of units

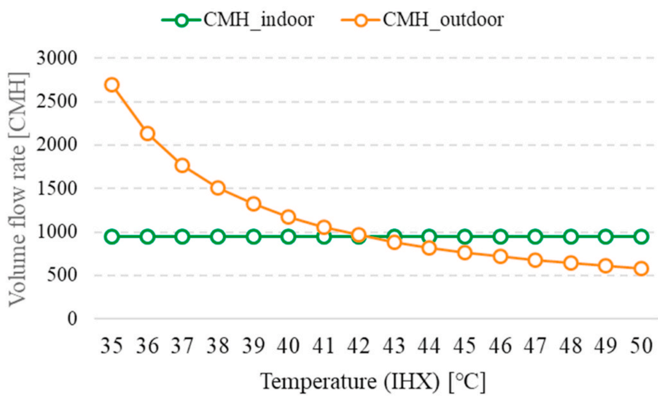
Fig. 7 shows the change in the air volume of the IDU and ODU when the temperature of the intermediate heat exchanger was changed. Fig. 7 (a) shows that a zero value of the ODU wind volume was derived up to 31 °C because the condenser temperature of LC is lower than 31.3 °C, which is the outside temperature, so the ODU does not operate until the temperature of the IHX is between 25 °C and 31 °C. A positive value is manifested at a temperature of 32 °C, where the condenser temperature was higher than the outside temperature, and an abnormally large amount of air was required to release LC's condenser heat to the outside air, calculated as a realistic air volume value as the temperature difference gradually increased. As shown in Fig. (b), as the IHX temperature increased, the air volume continuously decreased, and when it exceeded 42 °C, the air volume required was less than the indoor design air volume, 941 CMH.

5.1.2. Amount of heat at each component

Fig. 8 shows the thermal load of each heat exchanger unit of the entire system when the set temperature of the IHX, which plays the same role as the condenser of the LC, is changed. Based on the previous results, when operated in Case 2, there was no significant difference between the evaporator of the LC and the condenser temperature. Therefore, there was no significant difference in the amount of heat between the IDU and the IHX of the LC. ODU converges to zero because it cannot be operated. In contrast, when the set temperature of the IHX increased, the amount of heat increased slightly as the amount of work required for the LC compressor



(a) Volume flow rate of units in LC.



(b) Magnified image in Figure (a).

Fig. 7. Air volume flow rate of the IDU and ODU in various IHX temperatures.

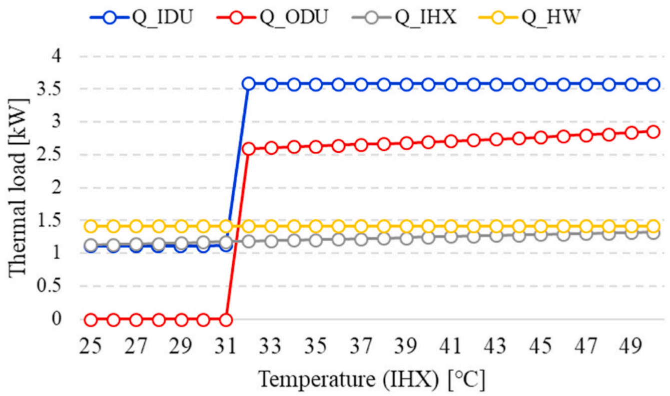


Fig. 8. Thermal load of each unit in various IHX temperatures.

increased.

In contrast, in the case of HC, when the condenser temperature of the LC was low, there is a large difference in the heat quantity between the evaporator and condenser of the HC. In addition, as the temperature of the IHX increased, the temperature difference between the evaporator and condenser of the HC decreased, and the difference in the heat value gradually decreased. Therefore, when the IHX temperature was 32 °C or higher, the IDU was 3.58 kW, which is equal to the indoor heat load, and the HW heat value was 1.41 kW, which is equal to the hot water supply heat. The IHX heat flux was 1.18–1.32 kW, and the ODU was the value obtained by subtracting the IHX heat flux from the total LC conditioner heat flux, and a heat flux of 2.60–2.9 kW was derived.

5.1.3. Primary energy consumption

An energy performance analysis was performed to confirm the amount of energy consumed (Fig. 9). The components that consume energy in the system are LC compressor, HC compressor, IDU fan, and ODU fan. It was assumed that the flow rate of the hot water supply was obtained by the water pressure when the water was turned on from the faucet, and the pump energy was not considered. All the components use electrical energy. Therefore, the primary energy consumption was calculated by multiplying the calculated load by the coefficient corresponding to electricity (2.75) when the primary energy coefficient was applied [47]. This condition is expressed as a bar chart according to the temperature setting value of the intermediate heat exchanger, and the energy consumption pattern according to the temperature change is confirmed (Fig. 9). As a result, the absolute energy consumption was the lowest at 25 °C to 31 °C. However, this temperature range is an abnormal condition, in which the system cannot operate as designed. Therefore, the energy consumption of the intermediate heat exchanger temperature was compared in the range of 32–50 °C. As the temperature increased, the energy consumption of the LC compressor gradually increased, whereas that of the HC gradually decreased, thus increasing the amount of energy required by the compressor because the temperature difference between the evaporator and condenser of the LC increased. Conversely, the HC temperature difference between the evaporator and condenser gradually decreased; thus, the amount of energy required by the compressor decreased. However, the fan energy of the IDU did not change because the design air volume for processing the indoor heat load was maintained. The fan energy of the ODU was changed by the difference between the outside and condenser temperatures. As the wind volume changed, the amount of energy consumed by the fan continued to decrease.

When the energy consumptions of all four components were added, the smallest energy consumption value was 2.29 kW h, which was derived when the intermediate heat exchanger set temperature reached 41 °C.

5.1.4. Result of COP

Finally, the COP of the entire system was derived for the temperature change in the IHX. As a result of the COP calculation in section 3.4, the COPs of LC and HC tended to follow opposite trends as the temperature increased because the temperature difference between the evaporator and the condenser temperature of the LC increased and those of the HC decreased, as only the IHX temperature changes, while the condenser temperatures of the LC and HC remain unchanged. Therefore, similar to the energy consumption results, the COP also shows the opposite tendency. In contrast, when the LC COP was less than 32 °C, the amount of energy consumed by the compressor was extremely small; thus, it seems that a high COP value was derived. However, the actual significant COP result value should be confirmed at 32 °C or higher, where the IHX temperature is above the outdoor air temperature.

As a result of checking the COP of the entire heat pump by combining LC and HC, the highest value of 11.5 was derived at 32 °C, after which it gradually decreased to 7.2 when the temperature was 50 °C. As a result of calculating the total system COP added up to the fan energy consumption, the smallest value of 1.9 was derived at 32 °C because of the dramatic decrease in ODU fan energy, whereas the COP value gradually increased as the temperature increased to reach 41 °C and 42 °C, resulting in a maximum value of 6.0 (Fig. 10).

6. Conclusions

This study examined the system sizing method to apply an air-to-air/water cascade heat pump to a building and conducted simulation research on two possible capacity calculation methods. The cascade heat pump system explored in this study has two

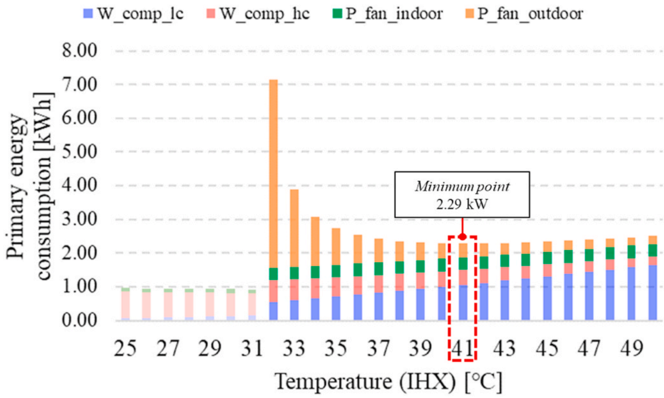


Fig. 9. Primary energy consumption in various IHX temperatures.

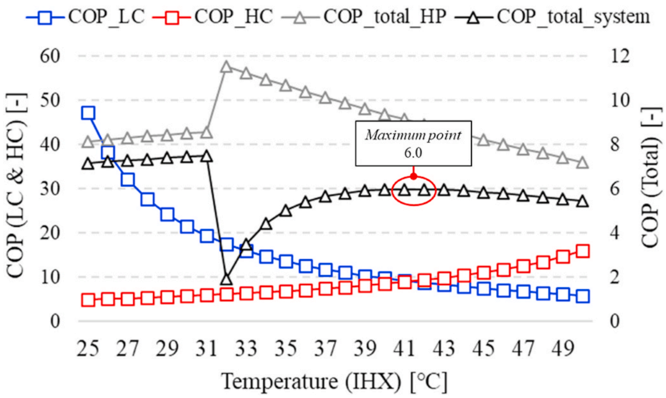


Fig. 10. Coefficient of performance for air-to-air/water cascade heat pump system in various IHX temperatures.

purposes: cooling indoor air in LC and heating domestic hot water in HC. The system design methods were classified into Case 1, which preferentially considers indoor cooling in LC, and Case 2, which prioritizes domestic hot water heating in HC, which was dealt with in previous studies. The building thermal load was calculated using the transient analysis tool, and based on this, the two sizing methods were applied for comparative analysis on the summer peak day. As a result of simulation according to each system sizing method, the refrigerant temperature in the IHX, which is the condenser of LC and connects LC and HC, was 41.3 ° C in Case 1 and 29.5 ° C in Case 2. The sizing method of Case 1, which designed the LC first, was able to sufficiently transfer the necessary heat to the HC among the condensation heat after processing all the indoor cooling loads in the summer, and the remaining condensation heat could be sufficiently dissipated due to the temperature difference (i.e., 10 ° C) from the outside air in the ODU. However, the method of Case 2, in which HC was designed first, had an issue in that it could not handle all of the cooling load in charge of the LC to handle the domestic hot water load of the residential building. This is because ODU cannot be utilized because the condenser temperature of the LC is lower than that of the outside air in the Case 2 method. For this reason, it was found that the heat dissipation amount of the LC side condenser, which is determined only for heat transfer to the HC side, is insufficient to handle the cooling load during operation of the LC heat pump.

Therefore, The results showed that the sizing method of Case 1 can derive an appropriate capacity calculation that can satisfy both indoor cooling and hot water supply because, when the system is designed using the method of Case 1, the heat of the condenser of the LC can be supplied from the IHX to the HC side, while the remaining heat can be radiated to the outdoor condition; thus, the heat balance of the system can be sufficiently satisfied. On the other hand, as a result of analyzing the energy consumption and COP of the entire system while changing the IHX set temperature value, the IHX set temperature of about 41 ° C showed the lowest energy consumption of the system at 2.29 kW, and the system COP was 6.0 was found to be the best.

In this study, a sizing study was conducted in the cooling-hot water mode. Owing to the nature of the air-to-air/water cascade heat pump system, the heat exchanger unit must be operated with a condenser for both indoor cooling and water heating. Therefore, as a sizing method different from the one considered in this study should be applied, a system design study on the heating-water supply mode is needed in future studies. In addition, when manufacturing a cascade heat pump system for future building applications, the design process and condition values derived from the results of this study can be applied.

CRediT author statement

Beom-Jun Kim: Conceptualization, Methodology, Data curation, Writing original draft preparation. Su-Young Jo: Methodology, Data curation, Jae-Weon Jeong: Supervision, Validation, Reviewing and Editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

Acknowledgements

This work was supported by grans form the Korea Institute of Energy Technology Evaluation and Planning (KETEP) (No. 20202020800360) and the National Research Foundation of Korea (NRF) (No. 2022R1A4A1026503).

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