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Energy saving potential of centralized heat pump-based air-conditioning system for an office building

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Abstract

This study explores a hybrid air-conditioning configuration for office buildings in which a centralized air-source heat pump (ASHP) supplies moderately conditioned water to multiple distributed water-to-air terminal heat pumps. The system is intended to mitigate the significant performance fluctuations commonly observed in decentralized air-to-air ASHP units when exposed to varying outdoor temperatures. To examine the potential benefits, a detailed thermodynamic cycle evaluation and annual performance simulations were performed using TRNSYS 18 and Engineering Equation Solver (EES). A representative office building model was adopted to quantify heating and cooling demands under realistic operating conditions. The simulation results demonstrate that the proposed system achieves noticeably higher compressor isentropic efficiencies compared with both standalone and fully centralized reference configurations. Individual terminal units in the hybrid system reached an average isentropic efficiency of approximately 0.66, while the central unit achieved around 0.80 due to the reduced compression ratio resulting from stabilized heat-source temperatures. Correspondingly, the total annual energy consumption of the proposed configuration was calculated to be about 99.4 MWh, a 16% and a 1.18% reduction relative to a conventional central ASHP system and a conventional individual ASHP system, respectively, when the energy use of the central heat pump and circulation pumps is included. These findings indicate that supplying a moderated water-based heat source to individual units can substantially enhance system stability, reduce compressor work, and yield significant annual energy savings.

Keywords: Building electrification; Heat pump; Isentropic efficiency; Energy consumption;

1. Introduction

Growing global commitments to energy transition and carbon neutrality have intensified efforts to improve the efficiency of building HVAC systems. Because heating, cooling, and domestic hot-water systems account for a large proportion of total building energy use, improving their performance is fundamental for reducing greenhouse-gas emissions [1]. Among various high-efficiency technologies, heat pumps have attracted extensive attention as they are able to draw low-grade thermal energy from ambient sources and upgrade it with relatively low electricity input [2]. Their compatibility with renewable energy systems further enhances their appeal for future electrified building systems [3].

Recent studies have investigated hybrid configurations that integrate heat pumps with diverse thermal sources, including air, ground, solar, and waste heat. Despite this progress, typical office buildings still commonly employ either decentralized air-to-air heat pumps installed in each zone or centralized systems that distribute chilled/hot water from a central ASHP. Decentralized systems offer installation convenience and reduced initial cost, but their performance is highly sensitive to outdoor air temperature. During periods of extreme weather, compression ratios rise sharply, leading to lower COP and increased energy consumption.

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Heating performance, in particular, deteriorates significantly in cold climates [4]. Furthermore, office buildings typically require high ventilation rates and operate for extended hours, amplifying the performance instability of decentralized units.

To address these limitations, hybrid concepts combining a central thermal source with multiple distributed terminal heat pumps have been proposed [5, 6]. In such systems, a central heat pump delivers moderately conditioned water that reduces the workload of individual units, leading to improved part-load performance and stable operation. When the central heat source is supplemented by renewable technologies such as solar thermal or geothermal systems, further energy savings can be achieved.

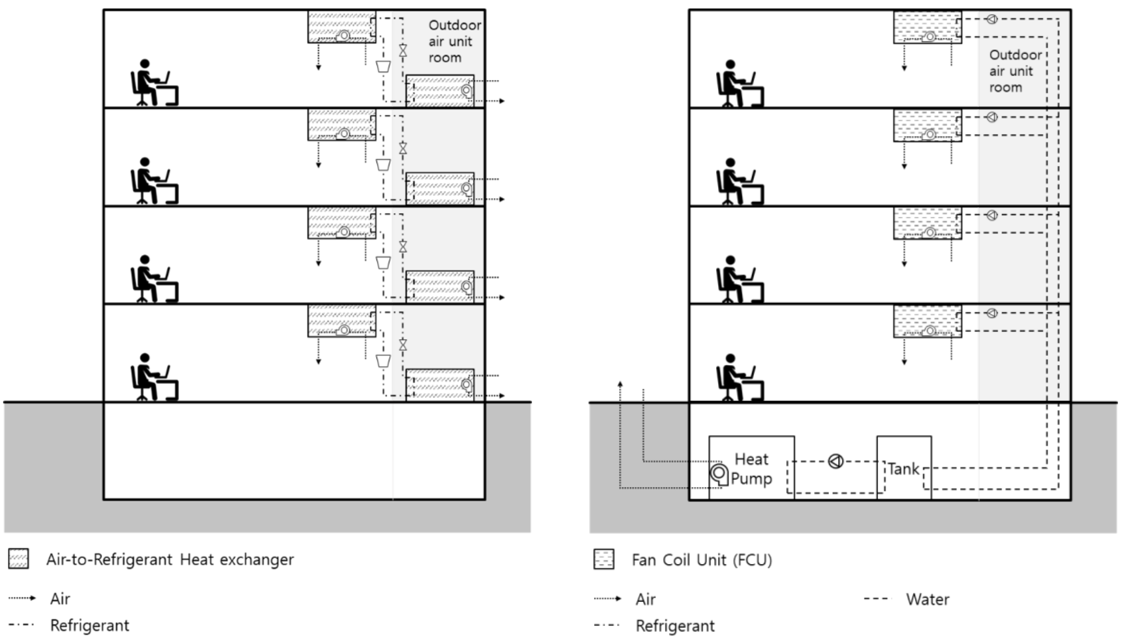
This study therefore focuses on the development and evaluation of a hybrid system that provides a centralized ASHP as a shared water-based heat source for distributed terminal heat pumps. Three system configurations were analyzed for a building of identical size: (1) decentralized air-to-air heat pumps, (2) a fully centralized ASHP with fan-coil units, and (3) the proposed hybrid system combining a central heat source with individual water-to-air heat pumps. Through thermodynamic analysis and annual simulation, the study quantifies improvements in COP, compressor performance, and energy consumption. The goal is to demonstrate that the proposed system can provide meaningful efficiency gains while maintaining compatibility with standard office building layouts.

2. System Overview

2.1. Reference system: conventional air-source heat pump systems

The first reference configuration consists of decentralized air-to-air heat pumps placed in each thermal zone. Each system is composed of an indoor unit (IDU) and outdoor unit (ODU), operating independently according to cooling or heating demand. Outdoor air serves as the heat source or sink, and R32 refrigerant is used throughout. Although easy to install, these units are directly exposed to outdoor temperature fluctuations, which strongly affect compressor operation.

The second reference configuration represents a typical central ASHP system used in medium-sized office buildings. A central water-type air-source heat pump produces chilled or hot water stored in a buffer tank. Water is then supplied to fan-coil units (FCUs) installed in each zone. Chilled-water temperature for cooling operation is maintained at approximately 10°C, while hot-water temperature for heating is maintained at about 45°C. For the building considered, 48 FCUs or individual ASHP units are distributed across four identical floors.



(a) Decentralized air-to-air HP system (Reference system 1)

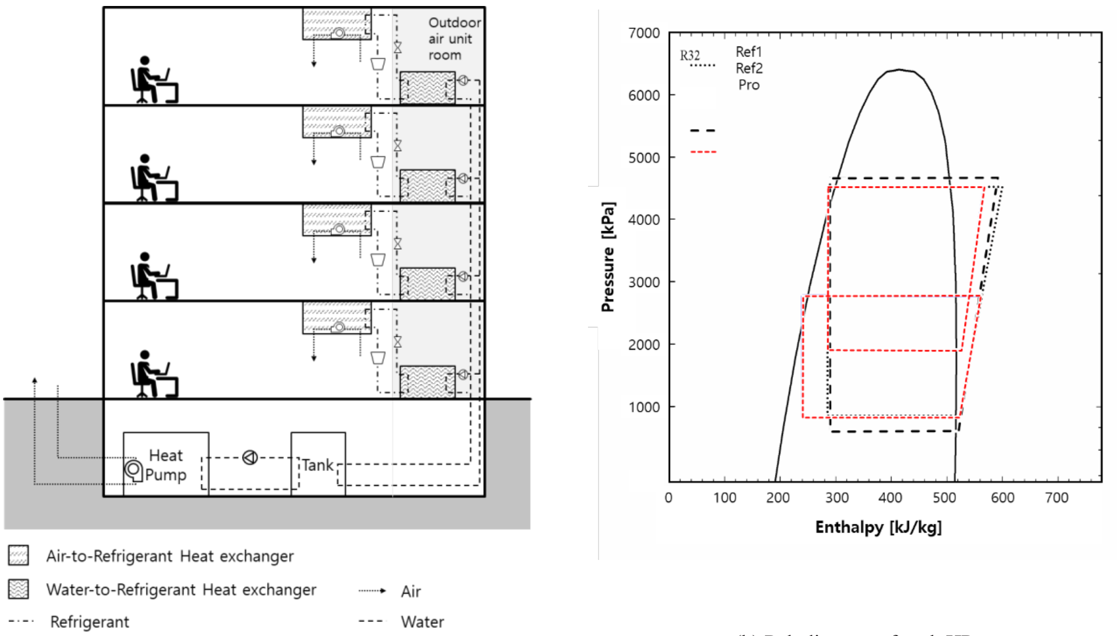
(b) Centralized air-to-water HP system with fan-coil units (Reference system 2)

Figure 1. Schematic of reference air-source heat pump (ASHP) systems for the office building

2.2. Proposed System: Central Heat-Source-Assisted ASHP System

The proposed system introduces a central air-to-water heat pump that supplies moderately conditioned water to individual water-to-air heat pumps located in each zone. Although this central heat pump can be linked with renewable heat sources, the present analysis focuses on an air-source unit in order to isolate the effects of system configuration. The central unit maintains chilled water at roughly 15–25°C during cooling and hot water at around 10–20°C during heating.

This moderate water temperature significantly lowers the condensing temperature for the terminal heat pumps during cooling and increases the evaporating temperature during heating, enabling improved COP. All heat pumps in the system use R32 refrigerant. A representative P-h diagram for system operation is provided to illustrate the thermodynamic cycle of both central and terminal heat pumps.



(a) System configuration integrating central and terminal units (Proposed system)

(b) P–h diagram of each HP

Figure 2. Conceptual diagram of the proposed central heat-source-assisted air-source heat pump (ASHP) system

3. Simulation Methodology

3.1. Building model

A mid-size office floor with a conditioned area of 900 m² and a ceiling height of 2.7 m was modeled. Envelope heat-transfer coefficients followed national energy-saving design standards. Internal heat gains included occupants (75 W/person), lighting (15 W/m²), and equipment (12 W/m²). Indoor design conditions were set to 26°C and 50% RH for cooling, and 20°C and 40% RH for heating. Ventilation loads were calculated according to ASHRAE Standard 62.1 [7]. TRNSYS 18 was used to generate dynamic space load profiles.

3.2. Heat pump model

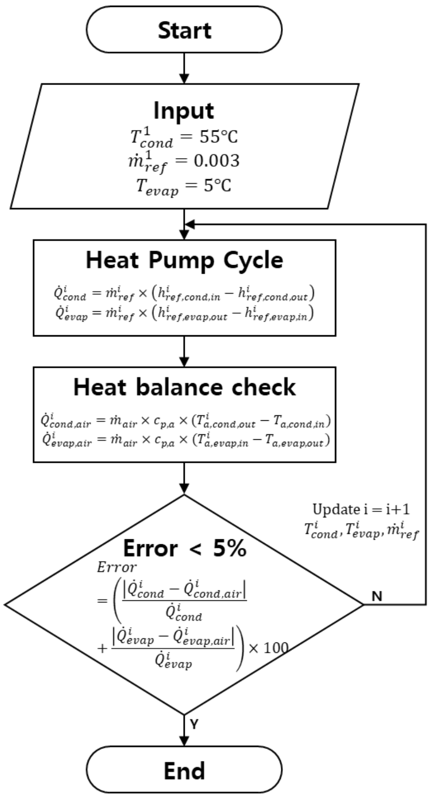
The heat pump model included compressor, condenser, evaporator, and expansion valve components. Compressor performance was represented using isentropic efficiency, which accounts for irreversibility in the compression process (Eq. (1)). A rotary compressor model was used for terminal units, and a scroll compressor model for the central heat pump. Correlations from previous literature [8] were adopted to estimate efficiency based on pressure ratio and volumetric flow rate (Eq. (2)). Actual discharge enthalpy was calculated using the predicted isentropic efficiency.

$$\eta_{comp} = \frac{h_{comp,ideal} - h_{comp,in}}{h_{comp,out} - h_{comp,in}} \quad (1)$$

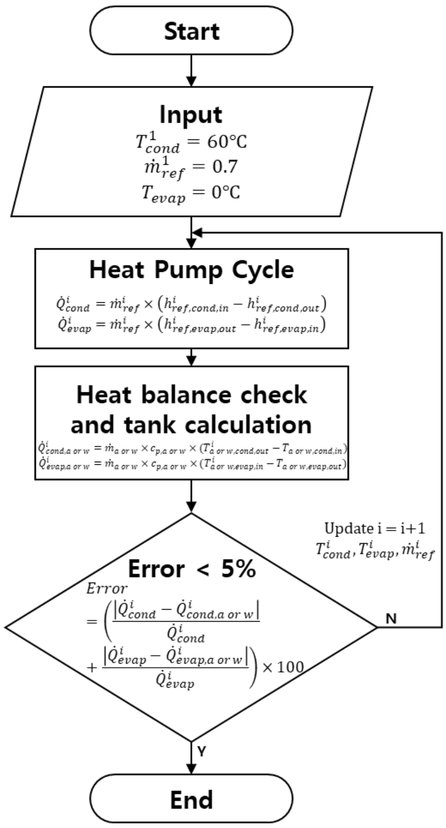
$$\eta_{comp} = a_1 + a_2 r + a_3 \dot{V}_{ref} + a_4 r^2 + a_5 r \dot{V}_{ref} + a_6 \dot{V}_{ref}^2 \quad (2)$$

The heat pump cycle was calculated based on the compressor model, as shown in Figure 3. The calculation procedure follows a sequential thermodynamic state-point approach, iteratively determining the refrigerant properties at the inlet and outlet of each component until convergence is achieved.

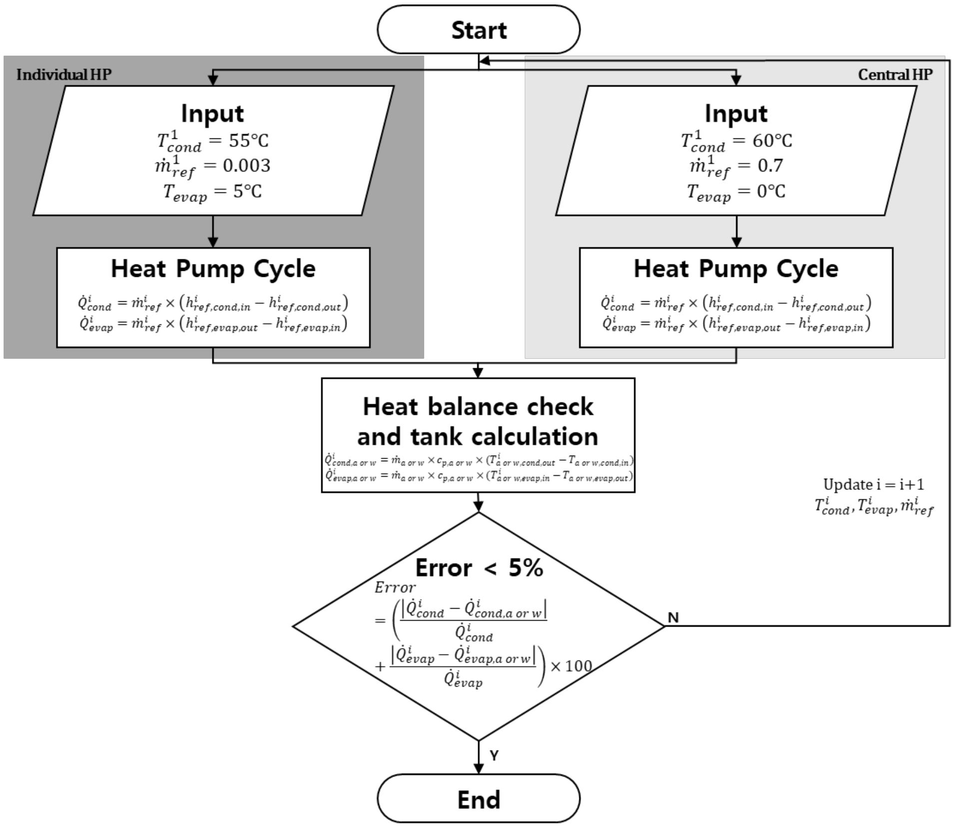
The cycle begins with the specification of indoor and outdoor air conditions, which define the evaporating and condensing temperatures of the heat pump. Based on these air-side boundary conditions, the saturation temperatures at the evaporator and condenser are estimated by applying fixed temperature differences that represent realistic heat exchanger approach values. These saturation temperatures are then used to determine the corresponding refrigerant pressures. At the compressor inlet, the refrigerant is assumed to be superheated vapor at the evaporating pressure. The inlet enthalpy and entropy are obtained from refrigerant property tables using the assumed superheat degree 5°C. An isentropic compression process is first calculated to determine the ideal discharge enthalpy at the condenser pressure. The actual compressor outlet enthalpy is then computed by applying the compressor isentropic efficiency, which accounts for mechanical and thermodynamic irreversibility. The evaporator outlet condition is assumed to be superheated degree 5°C, completing the thermodynamic cycle. The refrigerant mass flow rate is determined such that the evaporator heat transfer rate matches the zone cooling or heating demand. This matching is achieved through an iterative loop that adjusts the mass flow rate until the calculated heat transfer rate converges to the required load within a prescribed tolerance.



(a) Reference system 1



(b) Reference system 2



(c) Proposed system

Figure 3. Flowchart of the heat pump cycle and system calculation

Once convergence is achieved, the heat pump cooling or heating capacity, compressor power consumption, and coefficient of performance are calculated. Energy consumption included compressor power and additional pump or/and fan power as represented in Eq. (3). Energy efficiency ratio (EER) and COP were computed for each operating period by dividing cooling or heating output by total electrical input (Eqs. (4) and (5)) every hour.

$$\dot{P}_{HP} = \dot{P}_{comp} + \sum \dot{P}_{fan\ or\ pump} \tag{3}$$

$$EER_{HP} = \frac{\dot{Q}_{\ cooling}\ [Btu/hr]}{\dot{P}_{HP}\ [W]} \tag{4}$$

$$COP_{HP} = \frac{\dot{Q}_{\ heating}}{\dot{P}_{HP}} \tag{5}$$

3.3. Thermal storage tank

Because the storage tank influences the evaporating and condensing conditions of all heat pumps, a transient thermal model was used to track temperature changes [9]. At each time step, the net heat addition or removal from the tank was divided by its thermal capacity to update the temperature as shown in Eq. (6). An initial temperature of 15°C and a simulation time step of 1 second were used for the analysis.

$$T^{i+1}_{tank} = T^i_{tank} + \frac{\sum Q_{gain} - \sum Q_{loss}}{m^{i+1}_{tank} \times C_p} \tag{6}$$

3.4. Fan and pump

Every system has two fans: one for the heat exchanger fan for indoor heating and cooling, and one for outside air to dissipate and absorb heat from the heat pump. The required design fan power was obtained from Eq. (7), assuming a fan efficiency of 0.5 and applying the maximum air flow rate as the design volumetric flow. The fan power corresponding to part-load operation was derived from the generic fan performance curve expressed in Eq. (8).

$$\dot{P}_{fan,design} = \frac{\dot{V}_{air,design} \times \Delta P}{\eta_{fan}} \tag{7}$$

$$\dot{P}_{fan} = (b_1 + b_2 PLR + b_3 PLR^2 + b_4 PLR^3\ ) \times \dot{P}_{fan,design} \tag{8}$$

The pump power for each system was determined using Eq. (9), adopting a pump efficiency of 0.6. In the proposed system, the constant-speed pump provides circulation of the central heat pump and tank, and the variable-speed pump provides flow from tank to the individual heat pump. Similar to the proposed system, the water circulates from the central heat pump and tank by the constant-speed pump, and the other loop is from tank to the individual FCU using variable-speed pump.

$$\dot{P}_{pump} = \frac{\dot{m} \times H \times g}{\eta_{pump}} \tag{9}$$

4. Results and Discussion

4.1. Compressor isentropic efficiency

The proposed configuration showed improved compressor efficiency for both individual and central heat pumps. The average isentropic efficiency was approximately 0.66 for terminal units and 0.80 for the central unit. These values were higher than those obtained from both reference systems. The improvement can be attributed to the moderate water temperatures reducing the compression ratio and stabilizing operating conditions throughout the year.

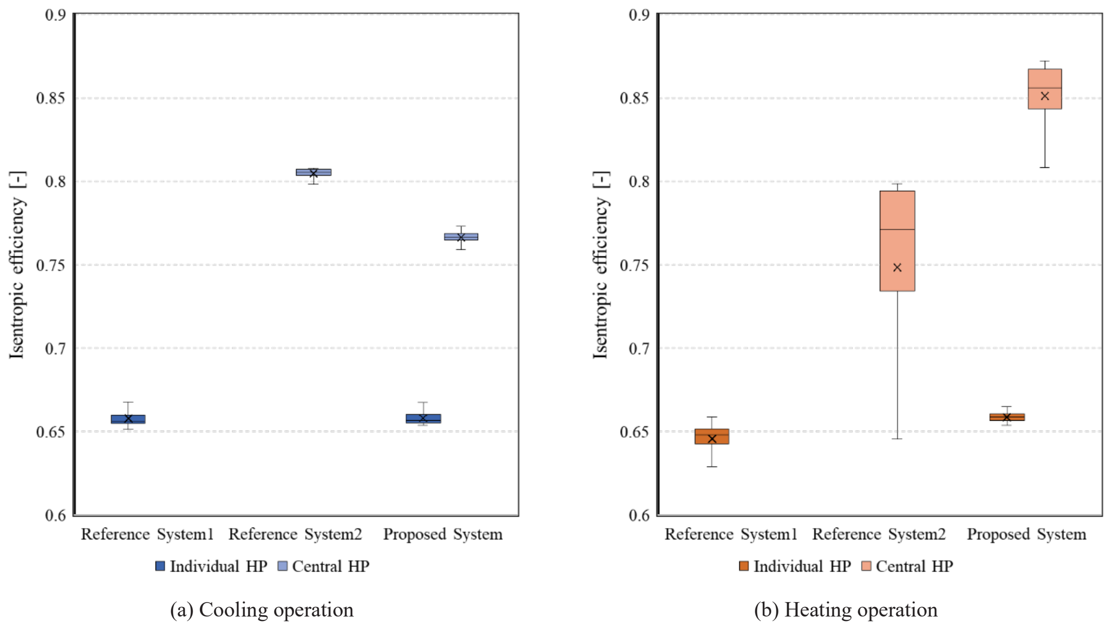
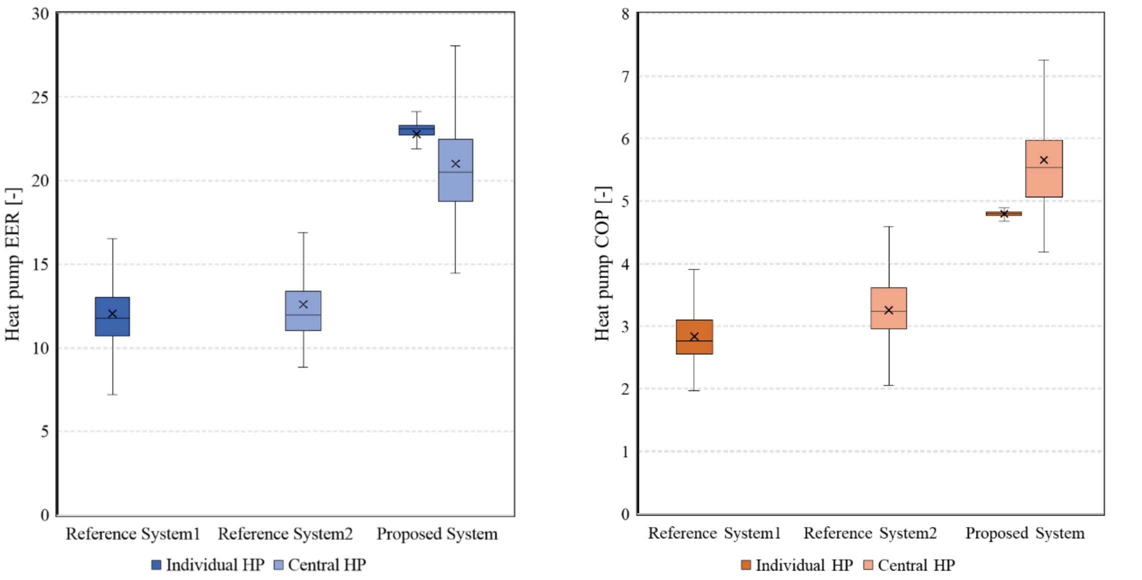


Figure 4. Comparison of compressor isentropic efficiencies under cooling and heating operation for the reference and proposed systems

4.2. Coefficient of Performance (COP)

The COP comparison showed clear performance benefits. Terminal heat pumps in the proposed system attained an average COP of 5.77, substantially higher than the 3.19 observed in the decentralized air-to-air configuration. Although the central heat pump in the proposed system operated at relatively high part-load ratios—which generally lowers COP—its performance still surpassed that of the fully centralized reference configuration. The enhanced operating conditions provided by the central tank led to reduced compressor work and improved efficiency for all units.



(a) Cooling operation (b) Heating operation

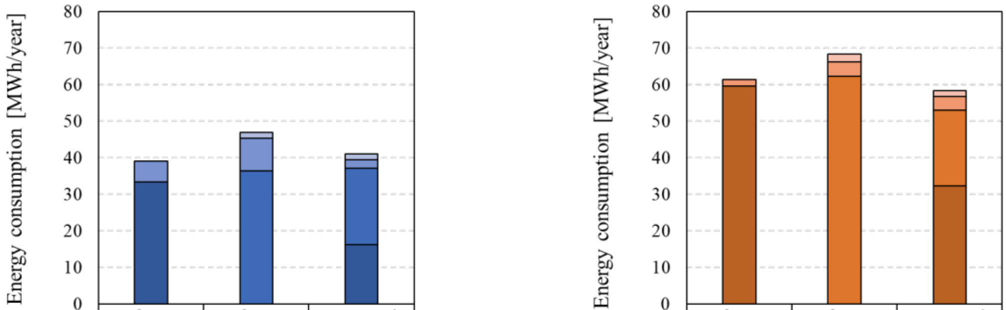
Figure 5. Comparison of EER and COP for individual and central heat pumps under different system configurations

4.3. Annual energy consumption

The proposed configuration demonstrated the lowest annual energy use among the three systems evaluated. Terminal units consumed approximately 48.39 MWh annually, representing a reduction of nearly 48% compared with decentralized units. When central heat pump and circulation pump consumption were included, the total system energy use was roughly 99.4 MWh. This represents an 16% improvement relative to the fully centralized reference system, which consumed 115.29 MWh. These results confirm that stabilizing the temperature of the heat source can significantly reduce electricity consumption and improve the overall cost-effectiveness of the HVAC system.

5. Conclusion

This study examined the performance of a proposed HVAC configuration that supplies moderately conditioned water from a central ASHP to distributed water-to-air heat pumps. The simulation results show that the proposed system can substantially enhance both compressor isentropic efficiency and COP relative to standard decentralized or centralized systems. The annual energy consumption was reduced by approximately 18%, demonstrating that the proposed approach effectively balances the advantages of centralized thermal stabilization and decentralized zone-level control. These improvements suggest that the proposed configuration offers a viable pathway for advancing electrified HVAC systems in office buildings. Integration with renewable heat-source technologies in future developments may provide additional opportunities for energy savings and carbon-emission reductions.



	Reference system1	Reference system2	Proposed system
Pump	0	1.57	1.59
Fan	5.7	8.93	2.28
Central Compressor	0	36.42	20.91
Individual Compressor	33.42	0	16.2

(a) Cooling season

	Reference system1	Reference system2	Proposed system
Pump	0	2.08	1.63
Fan	1.84	3.95	3.81
Central Compressor	0	62.34	20.79
Individual Compressor	59.61	0	32.19

(b) Heating season

Figure 6. Annual energy consumption of the reference and proposed air-conditioning systems

Nomenclature

$a_1 - a_6$: coefficients for compressor isentropic efficiency correlation [-]
$b_1 - b_4$: coefficients for fan energy consumption correlation [-]
C_p : specific heat [kJ/kg°C]
g : acceleration of gravity [m/s^2]
H : head loss [m]
h : enthalpy [kJ/kg]
$\dot{m}$: mass flow rate [kg/s]
m : mass [kg]
$\dot{P}$: power consumption [kJ/s]
P : pressure drop [kPa]
PLR : part load ratio [-]

$\dot{Q}$: load [kJ/s]
Q : heat capacitance [kJ]
r : compression ratio [-]
T : temperature [°C]
$\dot{V}$: volume flow rate [m^3/s]
η : efficiency [-]

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