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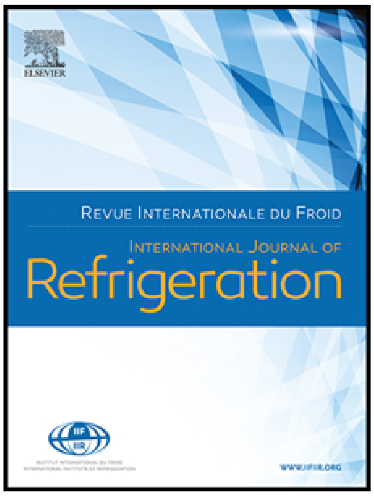

Experimental study and thermo-economic analysis of a novel radiant-convective cooling system

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Highlights

- A novel refrigerant-cooling radiant terminal system is proposed.
- The thermal comfort, energy and economic performance are experimentally studied.
- The proposed system has better indoor thermal comfort than split air-conditioners.
- The proposed system has higher energy efficiency ratio than split air-conditioners.

Experimental study and thermo-economic analysis of a novel radiant-convective cooling system

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Abstract

Split air-conditioners have limitations in terms of energy efficiency and local comfortability. Radiant cooling systems (RCSs) cannot be used in most residential buildings, due to its lack of space for a central plant room to achieve secondary heat exchange. To overcome these defects, a novel radiant-convective cooling system with the refrigerant-cooling radiant terminal (RCRT) was proposed, which took the advantages of both split air-conditioners and RCSs. Experiments were conducted in a climate chamber to investigate the thermal comfort performance, energy efficiency ratio and economic performance of the proposed system, in comparison with split air-conditioners. The results indicated that the RCRT cooling system could improve the indoor set temperature from 26 ^{o}C to 28 ^{o}C, under the same indoor thermal comfort as split air-conditioners. Additionally, the proposed system had higher energy efficiency ratio than split air-conditioners, and the energy saving potential of this system was 8.4%. The initial cost of the RCRT cooling system was higher than split air-conditioners, however, its life cycle cost was lower. The payback period of using the proposed system to replace split air-conditioners was 2.4 years, indicating that adopting the RCRT cooling system was more economical.

Keywords: refrigerant-cooling radiant terminal; thermal comfort performance; energy efficiency ratio; economic performance

Nomenclature

d	discount rate (%)
G	refrigerant mass flow rate (kg·s^{-1})

H	enthalpy ($J \cdot kg^{-1}$)
$\overline{LP}$	mean sound pressure level (dB(A))
L_{pi}	sound pressure level at point i (dB(A))
OC_n	operating cost of the nth year (\$)
P	pressure (Pa)
Q	total cooling capacity (W)
t	life time (year)
T	temperature (^{o}C)
V_a	air velocity ($m \cdot s^{-1}$)
W_s	input power (W)
Subscripts	
co	copper pipe
cond	conduction
conv	convection
in	inlet
m	mean
n	indoor air
out	outlet
r	refrigerant
rad	radiation
st	steel panel
wa	water

Abbreviations	
ECC	electrical consumption cost
EER	energy efficiency ratio
FS	full scale
IC	initial cost
LCC	life cycle cost
LI	loan interest
MC	maintenance cost
ME	material expense
MRT	mean radiant temperature
OC	operating cost
ODP	ozone depression potential
PMV	predicted mean vote
PPD	predicted percentage dissatisfied
RCRT	refrigerant-cooling radiant terminal
RCS_s	radiant cooling systems
RH	relative humidity

1. Introduction

In China, building energy consumption accounts for about 37% of the total energy use [1]. Nearly 23% of the building energy consumption is used to fulfill occupants' living or working comfort requirements by heating, ventilation and air conditioning (HVAC) systems in residential and commercial buildings [2]. In residential buildings in China, split air-conditioners are adopted as a popular way of fighting hot weather in summer [3]. This system consists of an indoor unit, an outdoor unit and connecting pipes, with refrigerant running inside to exchange heat based on an inverse Carnot cycle [4], as shown in Fig. 1. The compressor is used to compress the refrigerant into the states of high pressure and high temperature. Then the refrigerant is sent to the condenser to release heat to outdoor environment. After exchanging heat with outdoor environment, the refrigerant will be cooled and sent to the expansion valve, resulting in much lower temperature than indoor environment. At this state, the refrigerant is able to absorb heat from indoor environment through the evaporator, and then is sent back to the compressor for a new cycle [5].

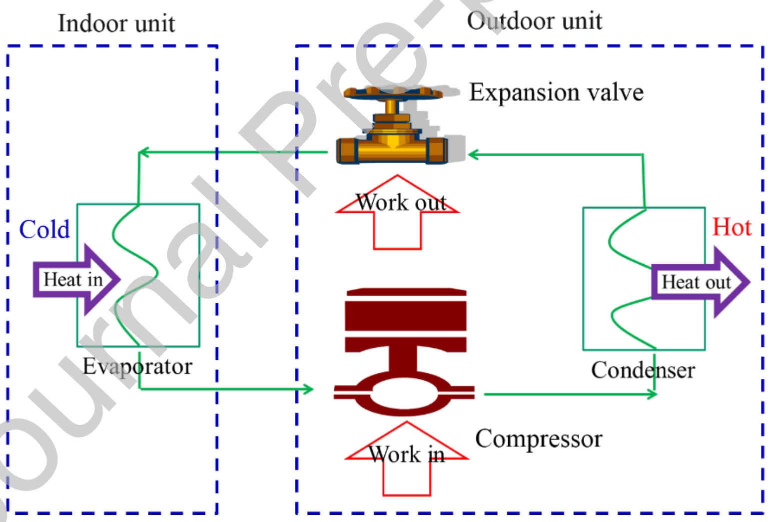


Fig. 1. The refrigerant cycle of split air-conditioners.

This solution, however, has limitations in terms of energy efficiency and local comfortability, due to the high dependence of heat exchange by convection and the use of fans to drive air movement [6]. Opoku et al. [7] conducted a field study on the energy efficiency of split air-conditioners in some offices in Ghana. Air-conditioners were divided into five categories by adopting an energy efficiency category method.

According to their energy efficiency ratio (*EER*), over 85% air-conditioners were in the lowest category and the remaining ones were in the second or the third category. He et al. [8] evaluated the local thermal comfort of rooms applying split air-conditioners, and confirmed that more than 20% of occupants vote on the uncomfortable side, because they had low comfort level with their body parts, like foot. Koo [9] pointed out noise issues of split-unit air conditioners, due to the existence of axial fans to drive air movement.

In recent years, radiant cooling systems (RCSs) have been widely used in commercial buildings, such as office buildings, airport terminals and railway stations [10], which take advantages of lower energy consumption and better indoor thermal comfort, comparing to split air-conditioners. The RCSs have three main components, the chiller, the chilled water distribution network and the radiant cooling terminal. The radiant cooling terminal can generally be divided into roof type, floor type and wall type, in which the former two types are more commonly used [11]. The RCSs are usually utilized together with independent dehumidification fresh air systems to both meet the requirements of fresh air and regulate indoor humidity level (reducing condensation on radiant surfaces) [12]. In RCSs, chilled water is commonly used as heat exchange medium to provide space cooling for occupants through radiation and natural convection [13]. Fig. 2 has depicted the basic operating principle of RCSs.

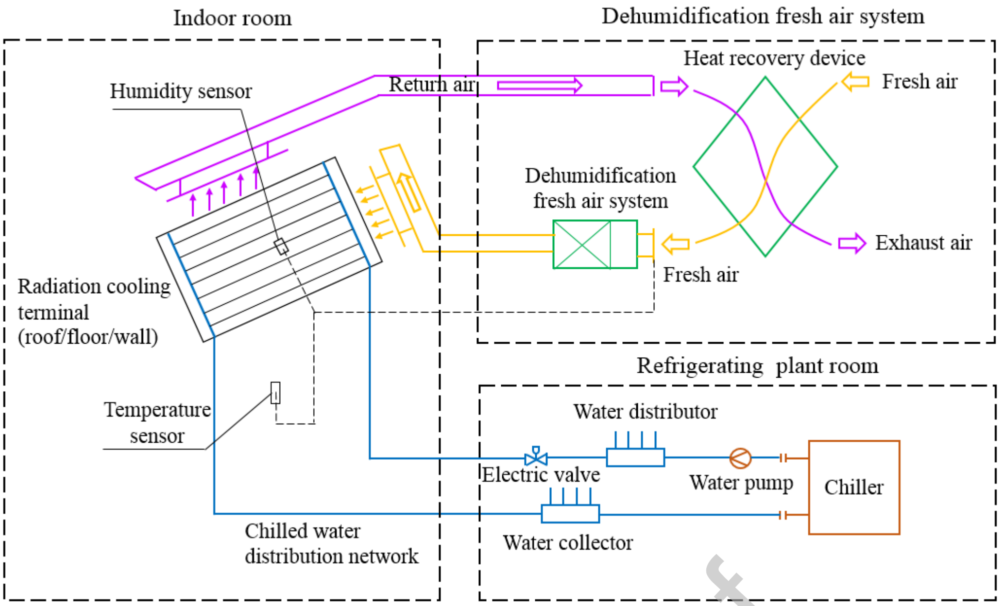


Fig. 2. The operating principle of RCSs.

In terms of energy saving, Zhao et al. [14] studied the performance of radiant floor cooling systems, and suggested that this system could save 20% ~ 30% energy than conventional all-air systems. Stetiu [15] used a numerical model and justified that the RCSs could save 30% of energy consumption in commercial buildings in California, comparing to all-air systems. Niu et al. [16] adopted an energy simulation method to investigate the energy performance of a system combined with chilled-ceiling and desiccant cooling, and suggested that this system could save up to 44% of primary energy consumption in comparison with conventional all-air systems. In terms of better indoor thermal comfort, Tian et al. [17] confirmed that radiant slab cooling systems could provide good vertical temperature gradient to reduce the chance of local discomfort by occupants, with similar conclusion given by Catalina et al. [18] for radiant cooling ceiling systems. Imanari et al. [19] suggested that radiant panel systems could provide better thermal comfort for occupants than conventional air-conditioners, not only from reduced vertical temperature gradient, but also less draught sensation. The latter phenomenon was supported by Corgnati et al. [20] as well, who used a CFD method and evaluated the main supply jet properties and draught risk of radiant ceiling panels. Shu et al. [21] suggested that the RCSs had better acoustic comfort for occupants, for which these systems had no indoor fans and were based on natural convective heat transfer.

Although RCSs have many advantages compared to split air-conditioners, they cannot be widely used in most residential buildings in China, due to the following issues:

1) RCSs use chilled water to exchange heat in a secondary loop, which works together with a primary loop to exchange heat with main cooling plants, such as chillers. Therefore, this system is more suitable for centralized HVAC solutions but in most residential buildings in China, the HVAC solutions are mainly based on individual apartments.

2) RCSs need a specific plant room to accommodate its major components, such as chillers, pumps, water pipes and water collectors. However, most existing residential buildings lack of space available for them in China.

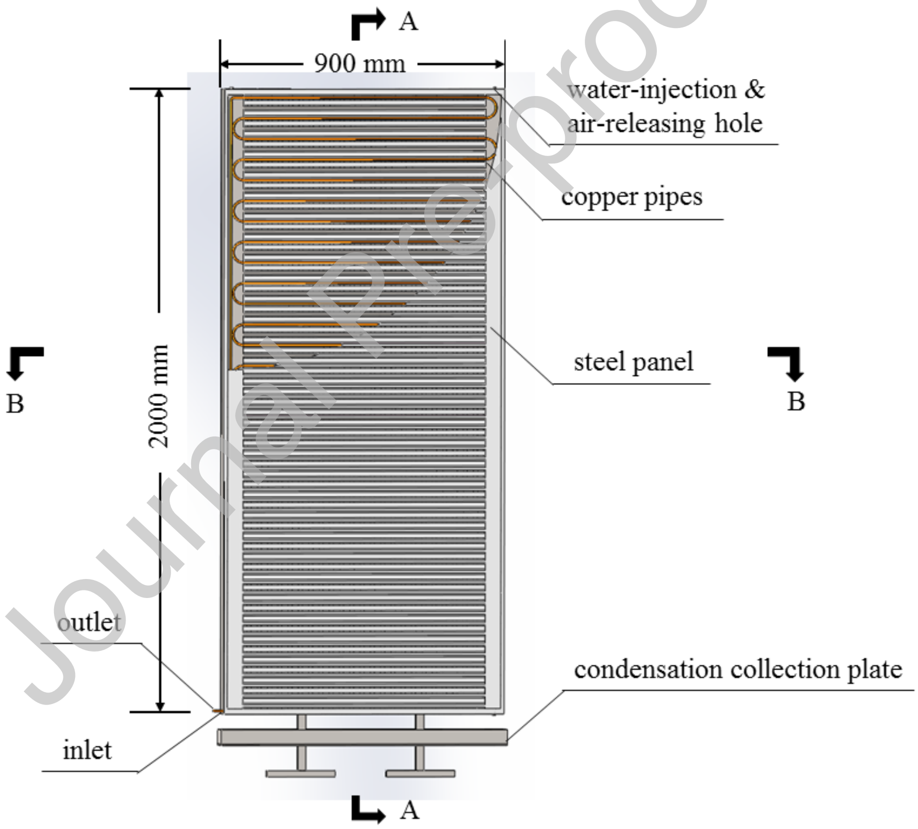
3) Changing split air-conditioners to RCSs needs to take all existing systems away and this is not financially effective.

To overcome the above issues, this study has developed a novel radiant-convective cooling system with the refrigerant-cooling radiant terminal (RCRT). This system was designed for residential buildings and there was no need of a central plant room. Additionally, it simply changed the existing indoor unit of split air-conditioners to a refrigerant-cooling radiant terminal, which could reuse many materials in existing systems, hence saving cost. Using experimental method, this paper has mentioned some main investigation results in terms of thermal comfort performance, energy efficiency ratio and economic performance of the proposed system, comparing to split air-conditioners. The RCRT cooling system could provide more comfortable indoor environment and less energy consumption for residential buildings.

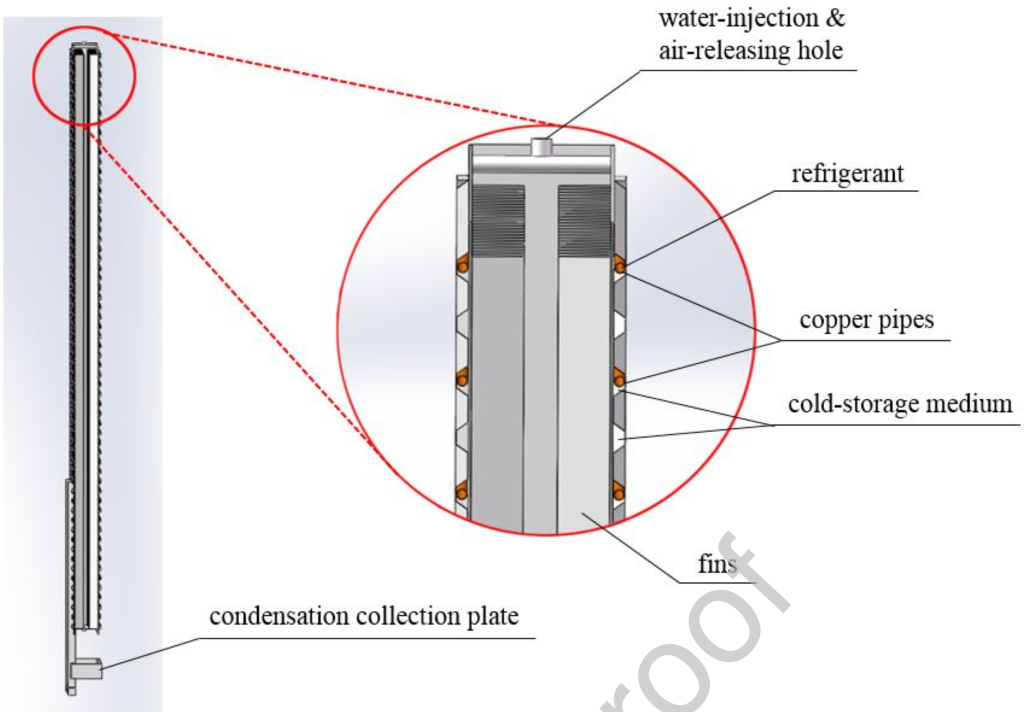
2. Methodology

2.1 Test rig

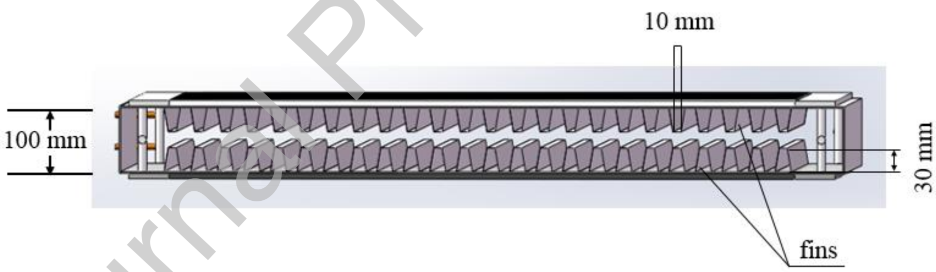
Fig. 3 (a) depicts the schematic of the RCRT, with the dimension of 900 mm (Length) × 100 mm (Width) × 2000 mm (Height). It consists of two steel panels, four branches copper pipes, cold-storage medium (water), many fins and one condensation collection plate. To promote the radiative heat transfer between the RCRT and indoor environment, the front of RCRT adopts a combined shape of plane and arc. The total length of copper pipes is 60 m, with the diameter of 6.35 mm, which is divided into four parallel paths to reduce pressure drop between the inlet and outlet of the RCRT. The copper pipes are designed as S-shape to promote heat transfer between the refrigerant and radiant panels. The diameters of the inlet and outlet pipes of the RCRT are 6.35 mm and 9.52 mm, respectively, allowing the state of refrigerant change from liquid to gas. As shown in Fig. 3 (b), the gap between steel panels and copper pipes is filled with cold-storage medium (water) to enhance the heat capacity. The weight of water is 10.85 kg. Fig. 3 (c) describes another profile of the RCRT, which has a total of 24 fins on the back of each steel panel, and the fins are 1800 mm long, 30 mm high and 1 mm thick. The water-injection hole and the air-releasing hole are located at the upper part of the RCRT to release excess air when filling water into the RCRT. The condensation collection plate is designed to collect condensation water, and then the condensation water is discharged to the specified location for preventing the fungi or mold generation. The RCRT adopts welding technology, with good sealing performance and no risk of refrigerant leakage.



(a) front



(b) A-A profile



(c) B-B profile

Fig. 3. The schematic of the RCRT.

Fig. 4 depicts the overall heat transfer process between the refrigerant and indoor environment. During the process, the cooling energy taken from the cold source is firstly transferred to the surface of copper pipes via heat convection. Then cooling energy is delivered from the surface of copper pipes to the water via heat conduction, and then to the RCRT surface via heat conduction as well. Lastly, the RCRT surface exchanges heat with indoor environment through radiation and natural convection.

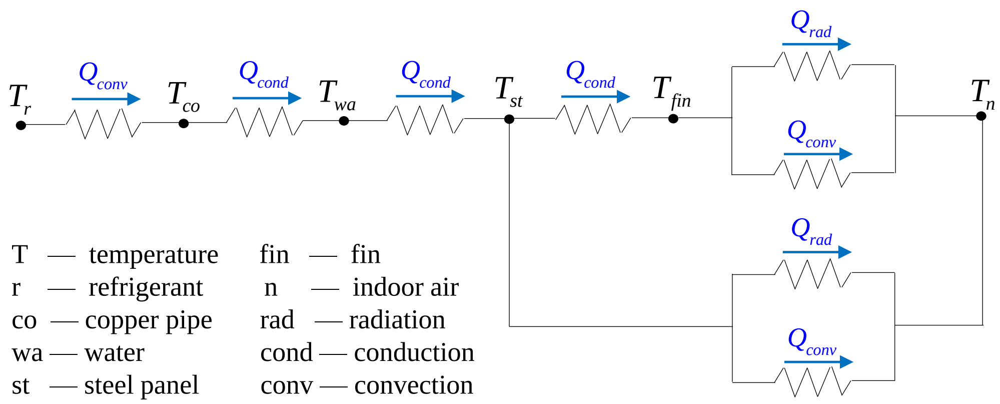
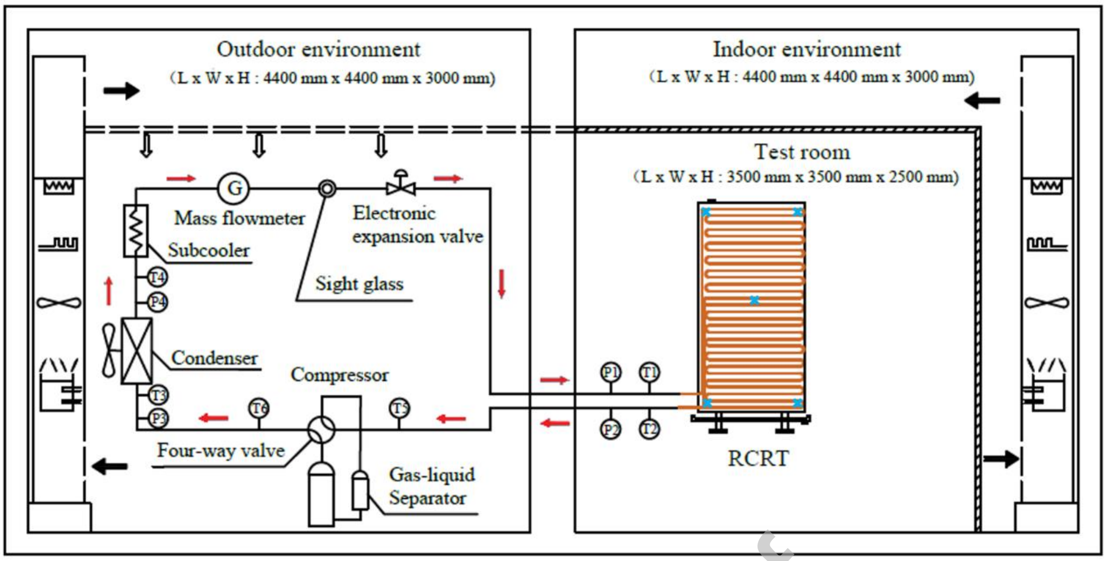
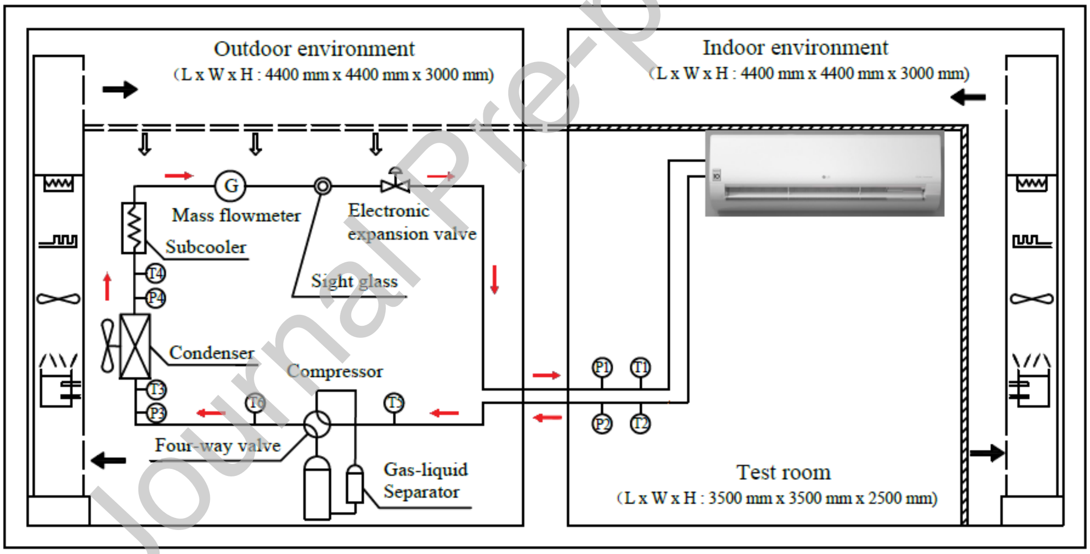


Fig. 4. The heat transfer process between the RCRT and indoor environment.

The performances of the RCRT cooling system and split air-conditioners are separately evaluated in a lab-based experiment, as shown in Fig. 5. The laboratory consists of two chambers, which are used to simulate outdoor environment (outdoor-environmental chamber) and indoor environment (indoor-environmental chamber). Their sizes are all 4400 mm (Length) × 4400 mm (Width) × 3000 mm (Height). The indoor environment of both chambers can be maintained at certain conditions using their separate air-conditioning systems. The outdoor conditions can be set between -20 ^{o}C and 60 ^{o}C for temperature, and between 20% and 90% for relative humidity, and the indoor conditions can be set between 10 ^{o}C and 50 ^{o}C for temperature, and between 20% and 90% for relative humidity. To reduce the impact of air movement in the chamber, a test room is built of steel plates in the indoor-environmental chamber, with the dimension of 3500 mm (Length) × 3500 mm (Width) × 2500 mm (Height).



(a) The RCRT cooling system



(b) Split air-conditioner

Fig. 5. The schematic of the two experimental systems.

Fig. 5 (a) depicts the basic structure of the RCRT cooling system in this study. This system is based on a refrigeration cycle, consisting of an inverter-driven rotor compressor, a condenser, a sub-cooler, an electronic expansion valve, a RCRT (evaporator) and connecting tubes. The RCRT is installed at the center of the vertical wall of the test room, and 100 mm above the floor for indoor air circulation. Correspondingly, the indoor unit of the split air-conditioner is installed above the vertical wall of the test room, as shown in Fig. 5 (b). Other components are installed in the outdoor-environmental chamber. A mass flowmeter is installed at the outlet of the sub-cooler to ensure the refrigerant in the liquid state, and a sight glass is installed behind the mass flowmeter to observe the state of refrigerant. The refrigerant of R410A is chosen for the two cooling systems since its ODP is zero. The refrigerant charge quantity of the RCRT cooling system is 900 g, which is a little higher than split air-conditioners (850 g) due to its longer copper pipes. The nominal cooling capacity and input electrical power of the compressor are 3200 W and 976 W, respectively.

In the experiments, the temperatures of test room are set at 22 ~ 28 ^{o}C, and relative humidity is remained 50% constantly. Taking the indoor air temperature of 26 ^{o}C as an example, the temperature set-point of the split air-conditioner is 26 ^{o}C. When the temperature set-point is reached, the split air-conditioner will stop. In terms of the RCRT cooling system, the indoor air temperature depends on the mean surface temperature of the RCRT. And the mean surface temperature of the RCRT is determined by the evaporating temperature. Then the evaporating temperature is relevant to the refrigerant mass flow rate, compressor frequency, opening of expansion valve and fan setting value. If the indoor air temperature keeps 26 ^{o}C, the setting values of the evaporating temperature, refrigerant mass flow rate, compressor frequency, opening of expansion valve and fan setting are 10.0 ^{o}C, 57.3 kg/h, 51 Hz, 60 b and 100 rpm, respectively. When the indoor air temperature set-point is reached, the RCRT cooling system will maintain its current condition. The transient response time of the proposed system is about 21 min due to the filled water between the steel panels and copper pipes. If the user intends to shut-down this system, the RCRT will continuously provide cooling for a period of time owing to the impact of heat capacity.

2.2 Data collection

To evaluate the performance of the RCRT cooling system, some parameters are collected in this experiment, as shown in Fig. 5. *T1* and *T2* are the inlet and outlet temperatures of the RCRT, and *P1* and *P2* are the inlet and outlet pressures. *T3*, *T4*, *P3* and *P4* are the corresponding values for the condenser. In addition, four temperature points are located at the boundary of the RCRT and one temperature point is located at the center. The arithmetic mean value of these temperature points is regarded as the mean surface temperature of the RCRT. The mass flow rate of the refrigerant and the input power of this system are measured as well, as indicated by G and W_s. All temperatures and pressures are measured by calibrated temperature sensors and pressure sensors. The mass flow rate of the refrigerant is measured by a mass flowmeter, and the input power of this system is measured by a power meter.

In addition to the above system parameters, some temperatures in the test room are measured to reflect its thermal environment. These temperatures include the surface temperatures of ceiling, floor and four walls, as indicated in Fig. 6. The indoor temperatures are measured at different locations, as indicated by A1, A2, A3, A4 and A5, and at different distances from the RCRT, namely, 0.25 m, 1 m, 1.75 m, 2.5 m and 3.4 m, as well. According to ASHRAE Standard 55 [22], the measured temperature at the height of 1.1 m in the vertical centerline is used to represent the temperature of the test room. The measured humidity place is also located at this point since there is no standard for measuring humidity at present. The air velocity is measured by an anemometer at the height of 1.5 m in the centerline of the test room, which is available in ASHRAE Standard 138 [23]. The acoustic level is measured by a noise detector, as suggested in the standard [24], and the location is shown in Fig. 6. Correspondingly, the same measuring points are obtained for the split air-conditioner as well. All parameters are recorded by the central data logger every 2 s. Table 1 lists the detailed specifications of instruments.

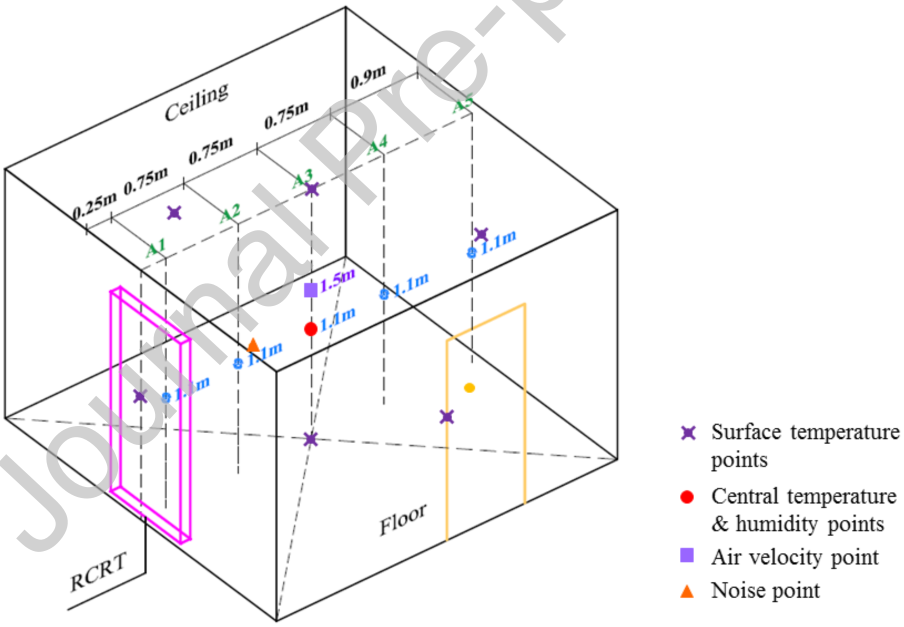


Fig. 6. Parameter measurement in the test room.

Table. 1 Detailed specifications of instruments.

Measured parameter	Instruments	Full scale	Accuracy

temperature	copper-constantan thermocouple	-50 – 150 ^{o}C	± 0.5 ^{o}C
relative humidity	testo humidity sensor	2 – 98% RH	± 2% RH
pressure	pressure transmitters	0 – 45 bar	0.1% FS
mass flow rate	mass flowmeter	0 – 100 kg·h^{-1}	± 0.02 kg·h^{-1}
power meter	/	0.01 – 999 kW	± 0.01 kW
noise level	AR844	30 –130 dB(A)	± 0.1 dB
air velocity	TSI-Q-Trak	0 – 40 m·s^{-1}	± 0.02 m·s^{-1}

2.3 Performance indicators

Fanger's predicted mean vote (*PMV*) and predicted percentage of dissatisfied (*PPD*) models are used to evaluate the indoor thermal environment, based on temperature (measured T_n at the height of 1.1 m), relative humidity (measured *RH* at the height of 1.1 m), air velocity (measured V_a at the height of 1.5 m), mean radiant temperature (*MRT*), clothing insulation (0.5 clo) and metabolic rate (1 met). Their detailed calculation methods are available in [25]. The classification of thermal environment is listed in Table 2 [26]. The acoustic level is evaluated by Eq.(1):

$$\overline{LP}=10\lg\frac{1}{3}[\sum_{i=1}^{3}10^{0.1L_{pi}}] \quad (1)$$

where $\overline{LP}$ is the mean sound pressure level of the measured surface, L_{pi} is the sound pressure level at point *i*.

Table. 2. The classification of thermal environment.

Category	Thermal sensation of human body	
	PMV	*PPD*/%
A	-0.2<PMV<+0.2	<6
B	-0.5<PMV<+0.5	<10
C	-0.7<PMV<+0.7	<15

For the RCRT cooling system, the total cooling capacity is estimated by the refrigerant mass flow rate and the enthalpy difference between the inlet and outlet of the RCRT, as shown in Eq.(2):

$$Q = G \times (H_{\mathrm{out}} - H_{\mathrm{in}}) \tag{2}$$

where Q is the total cooling capacity, G is the refrigerant mass flow rate, H_{in} and H_{out} are the enthalpy of the inlet and outlet of the RCRT, which are obtained by the measured temperature and pressure.

The *EER* of this system can be calculated according to Eq.(3):

$$EER = \frac{Q}{W_s} \tag{3}$$

where W_s is the input power of the system.

2.4 Uncertainty analysis

The uncertainties of various directly and indirectly measured parameters are given as follows [27]:

$$u_{xi} = \frac{e_i}{k} \tag{4}$$

$$u_y = \sqrt{\sum_{i=1}^{n}\left(\frac{\partial f}{\partial x_i} u_{x_i}\right)^2} \tag{5}$$

where u_{xi} is the uncertainty of x_i, e_i is the maximum error of sensors, k is the coverage factor and u_y is the uncertainty of u_{xi}.

The uncertainties of G, H_{in}, H_{out}, L_{pi} and W_s are 1.2%, 5.8%, 5.8%, 1.4% and 4.6%, respectively, due to instrument calibration errors. The uncertainties of Q, EER and $\overline{LP}$ are 2.8%, 4.0% and 1.5%, respectively.

3. Results and discussions

3.1 Indoor thermal comfort

3.1.1 Overall thermal comfort based on PMV and PPD

In this study, the overall thermal comfort is decided by the Fanger's *PMV* and *PPD* models, as adopted in many building design standards [25]. During the experiments, the temperature and relative humidity of the test room, clothing insulation and metabolic rate are kept at the setting value. The fan settings of the split air-conditioner are set at low speed, medium speed and high speed, respectively. It can be noted that air velocity and *MRT* are two crucial factors affecting the overall thermal comfort in residential buildings. However, the effect of air velocity on the overall thermal comfort of the two systems is fairly small. It can be attributed that the measured values of air velocity difference between the split air-conditioner and the RCRT cooling system are merely 0.01 ~ 0.05 m·s^{-1}. This phenomenon is also observed in Ref. [28]. Hence, when the fan setting of the split air-conditioner keeps medium speed, the effects of *MRT* on the overall thermal comfort of the two systems are investigated. Fig. 7 depicts the measured *MRTs* of the split air-conditioner and the RCRT cooling system under different indoor air temperature. From the comparison, it can be found that with the increase of indoor air temperature, both *MRTs* are increased linearly, with almost the same growth rate. In addition, the *MRT* of the split air-conditioner is very close to the indoor air temperature, while the *MRT* of the RCRT cooling system is lower than the indoor air temperature consistently, with an average difference of 1.4 ^{o}C. This is because the RCRT cooling system exchanges heat with indoor environment through radiation and natural convection. The variation trend of mean surface temperature of the RCRT with indoor air temperature is also exhibited in Fig. 7. With the increase of indoor air temperature, the mean surface temperature of the RCRT ascends linearly, within the range of 11.6 ^{o}C to 12.0 ^{o}C. However, the *MRT* of the split air-conditioner is generally not evaluated by its mean surface temperature, which is ignored in this study. It can be concluded that the RCRT cooling system can provide cooler indoor thermal environment than split air-conditioners under the same indoor air temperature.

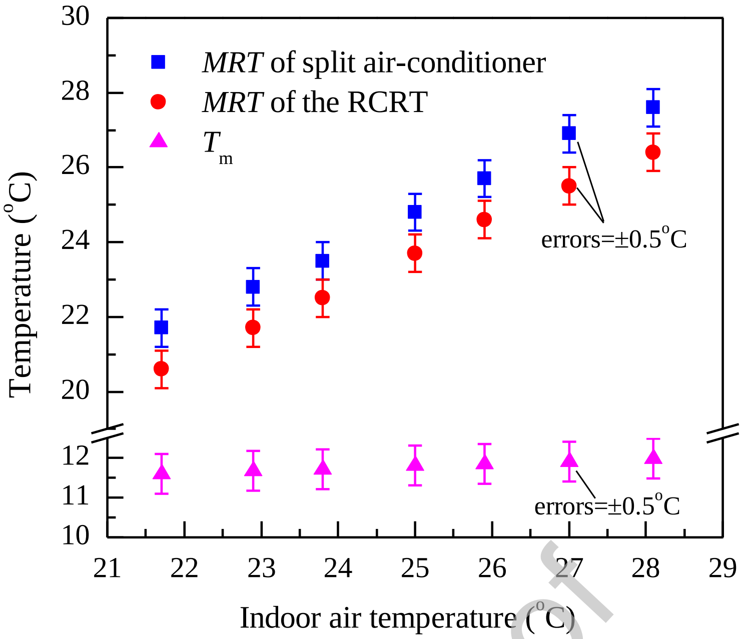


Fig. 7. The *MRTs* and the RCRT surface temperature under different indoor air temperature.

The calculated *PMVs* of the split air-conditioner and the RCRT cooling system under different indoor air temperature are compared in Fig. 8. With the increase of indoor air temperature, both *PMVs* are increased. Besides, the mean values of *PMVs* of the RCRT cooling system are lower than the split air-conditioner constantly. When the indoor air temperature keeps 26 ^{o}C, the *PMVs* of the split air-conditioner and the RCRT cooling system are -0.2 and -1, respectively. It is indicated that the occupants will feel more comfortable in a room with split air-conditioners since its thermal comfort level attains Class A, as shown in Table 2. Correspondingly, when the indoor air temperature keeps 28 ^{o}C, the *PMVs* of the two systems are 1.5 and 0.2, respectively. It should be noted that the RCRT cooling system can provide an ideal thermal environment as its thermal comfort level meets Class A. The similar conclusion is also displayed in Fig. 9, the *PPDs* of the split air-conditioner and the RCRT cooling system are 5 (Class A) and 14, respectively, with the indoor air temperature of 26 ^{o}C. Correspondingly, the values of the two systems are 20 and 6

(Class A), respectively, with the indoor air temperature of 28 °C. Therefore, in the premise of ensuring the same indoor thermal comfort, it is capable of improving the indoor set temperature from 26 °C to 28 °C when using the RCRT cooling system for supplying cooling in summer.

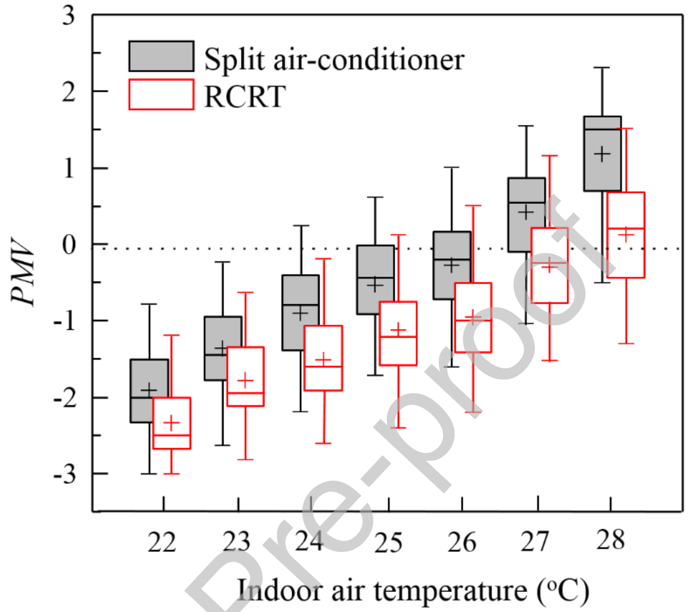


Fig. 8. The *PMVs* under different indoor air temperature.

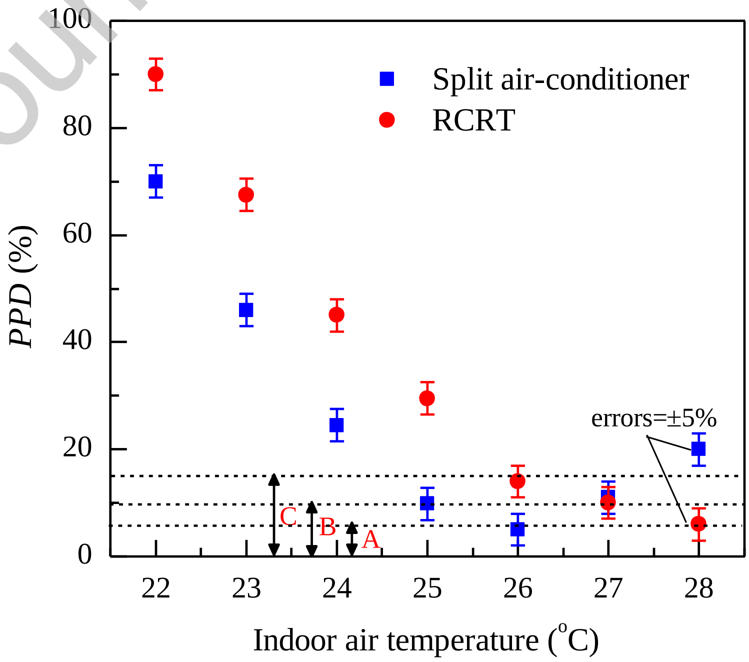


Fig. 9. The *PPDs* under different indoor air temperature.

3.1.2 Distribution of horizontal temperature

The distribution of horizontal temperature at the height of 1.1 m is compared in Fig. 10, between the split air-conditioner and the RCRT cooling system, with indoor air temperature of 27 ^{o}C. The results reflect that with the increase of distance from the RCRT, the horizontal temperatures of the two systems are improved. The horizontal temperature of the split air-conditioner is heterogeneous, with a fluctuation 1.2 ^{o}C. However, the horizontal temperature of the RCRT cooling system is homogenous, with a fluctuation merely 0.5 ^{o}C. It can be demonstrated that the RCRT cooling system is able to provide more comfortable indoor thermal environment than split air-conditioners in terms of horizontal temperature distribution.

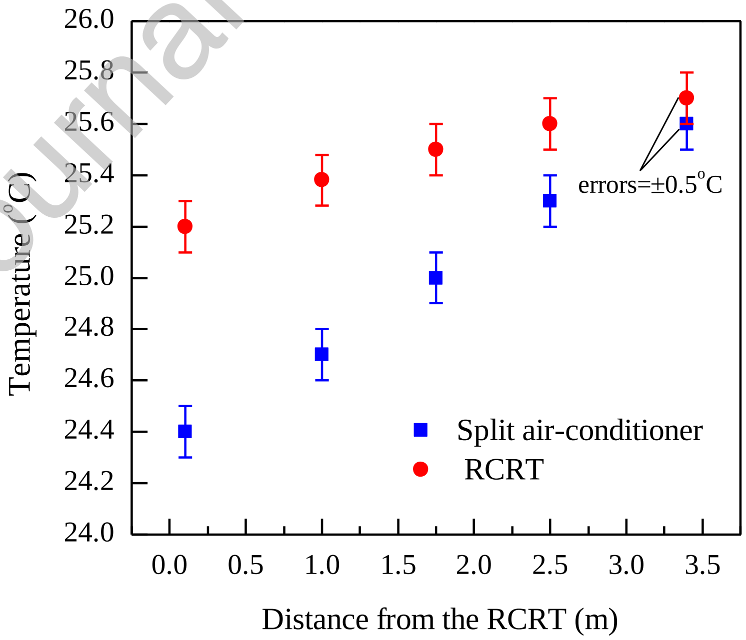


Fig. 10. The temperature distribution at the horizontal direction.

3.1.3 Acoustic performance

For designing the air-conditioning system in buildings, acoustic comfort is also an important index [29]. When evaluating indoor acoustic comfort, the weighted equivalent sound pressure level can be utilized, as defined in EN 15251 [30]. In this study, the sound levels of different fan speeds, namely, low, medium and high, are measured under the indoor air temperature maintains 26 ^{o}C. The same data are obtained of the RCRT cooling system as well. Results in Fig. 11 show that the mean values of sound levels are 32 dB(A), 33 dB(A), 34 dB(A) under three different fan speeds of the split air-conditioner. However, the mean value of sound level is only 25 dB(A) in the RCRT cooling system. In addition, the sound levels of the split air-conditioner are higher than the RCRT cooling system constantly, with the maximum difference of 9 dB(A). Therefore, compared with the split air-conditioner, the RCRT cooling system can operate in lower acoustics, which is very helpful to enhance the occupants' acoustic comfort.

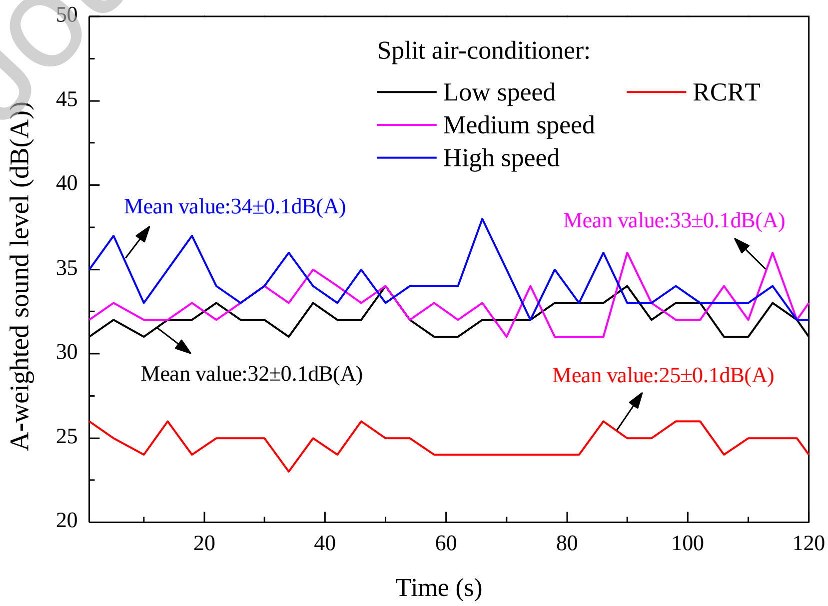


Fig. 11. Sound levels of the two cooling systems.

3.2 Energy performance

To compare the energy performance of the split air-conditioner and the RCRT cooling system, the *EERs* at different outdoor air temperature are calculated under the indoor air temperature remains 26 ^{o}C, as shown in Fig. 12. Obviously, the *EERs* of both systems are negatively correlated with outdoor air temperature, which can be ascribed that with the increase of outdoor air temperature, the heat exchange intensity of condenser descends, then the condensing temperature and pressure raise, resulting in the enhancement of compressor compression ratio and depression of compressor volumetric efficiency. At the same time, the evaporating temperature and pressure ascend, leading to the reduction of total cooling capacity and increment of power consumption. The results reveal that the RCRT cooling system has higher energy efficiency ratio than the split air-conditioner, within the range of 0.20 to 0.35. Fig. 12 compares the *EERs* of the proposed system with a split air-conditioner from an existing study [31], the results show that the energy efficiency ratio of the RCRT cooling system is superior to the split air-conditioner. It is attributed to the RCRT cooling system with higher evaporating temperature and no fan energy consumption.

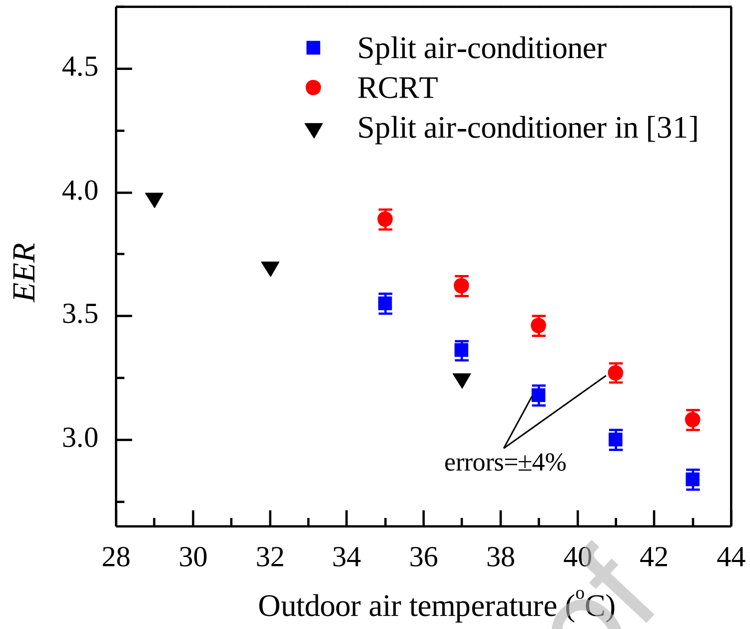


Fig. 12. The *EERs* of the two cooling systems.

To further justify the energy saving potential of the proposed system, a calculation is conducted to evaluate the annual electrical consumptions of the two cooling systems. According to the Air-conditioning System Design Manual in China [32], the indoor cooling load of multi-storey buildings is within the range of 88 ~ 150 W·m^{-2}. For this study, the medium value of 130 W·m^{-2} is adopted. The average living area of Beijing (China) is 94 m^2 and the total cooling time of this city is 782.7 h [33]. The electrical consumptions of both systems are estimated separately based on the hourly outdoor temperature of Beijing in summer [33], as shown in Table 3. The results show that the energy saving potential of the RCRT cooling system is 8.4% in comparison with split air-conditioners.

Table. 3. Details of the electrical consumption.

Type	Cooling load	Cooling time	Electrical consumption

	(kW)	(h)	(kWh·year^{-1})
Split air-conditioner	12.22	782.7	2359.04
RCRT	12.22	782.7	2161.47

3.3 Life cycle and payback period

Life cycle cost (*LCC*) is an important indicator to evaluate the economic performances of the RCRT cooling system and split air-conditioners. Life cycle cost is composed of initial cost (*IC*) and operating cost (*OC*) [34], as shown in Eq.(6). As the outdoor units of the two systems are same in this study, the initial costs of their indoor units are compared. The initial cost of the RCRT consists of the material expense (*ME*), such as steel panels, coppers pipes, fins and loan interest, using Eq.(7). The loan interest (*LI*) of investment is 4.9% [35]. The initial cost of split air-conditioners is acquired from air-conditioner manufacturers, and the result is shown in Table 4. The operating cost includes electrical consumption cost (*ECC*) and maintenance cost (*MC*), can be calculated by Eq.(8) .Using Beijing as a case study again, the electricity price is 0.07 \$·kWh^{-1} [35], and the electrical consumption costs of the two systems can be obtained. The maintenance costs of both systems are equal to 3% of their initial costs [36], and the life cycle of them is considered as 10 years. The life cycle costs of the two systems are compared in Table 4.

$$LCC = IC + \sum_{n=1}^{t} OC_n (1+d)^{-n} \tag{6}$$

where *LCC* is the life cycle cost, *IC* is the initial cost, OC_n is the operating cost of the *n*th year, *t* is the life time, *d* is the discount rate, 5% [37].

$$IC = ME_{\mathrm{st}} + ME_{\mathrm{co}} + ME_{\mathrm{fin}} + LI \tag{7}$$

where ME_{st} is the material expense of steel panels, ME_{co} is the material expense of coppers pipes, ME_{fin} is the material expense of fins, LI is the loan interest of investment.

$$OC = ECC + MC \tag{8}$$

where ECC is the electrical consumption cost, MC is the maintenance cost.

Table. 4. The life cycle costs of the two cooling systems.

Type	ME_{co} (\$)	ME_{st} (\$)	ME_{fin} (\$)	IC (\$)	OC (\$·year^{-1})	LCC (\$)
Split air-conditioner	/	/	/	73.5	169.4	1525.3
RCRT	59.1	31.0	12.4	102.5	155.2	1449.4

The RCRT cooling system has higher initial cost than the split air-conditioner, due to its long copper pipes and large steel panel areas. However, the life cycle cost of the proposed system is lower than the split air-conditioner, which can save 75.9 $. As shown in Fig. 13, the payback period of using the RCRT cooling system to replace split air-conditioners is 2.4 years, which is much shorter than the life time of both cooling systems. From the long-term perspective, adopting the RCRT cooling system in residential buildings has excellent economic performance.

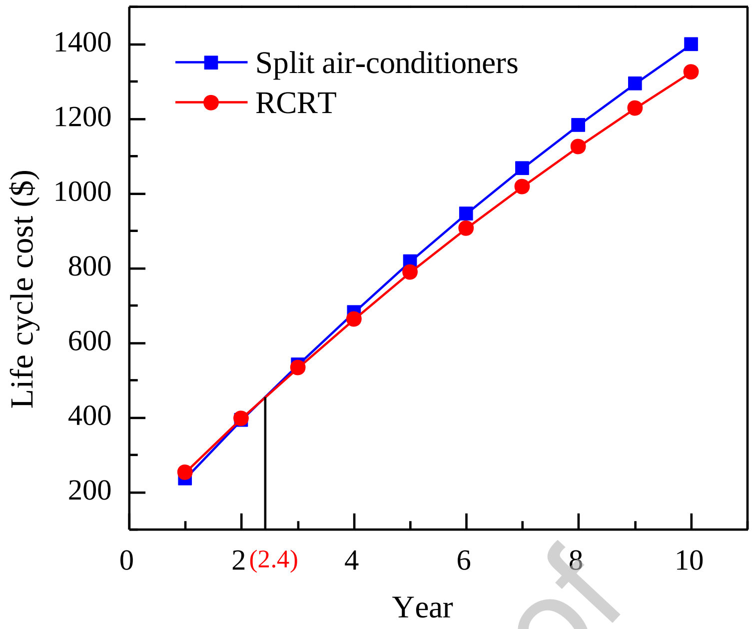


Fig. 13. The payback period of using the RCRT cooling system.

4. Conclusions

To overcome the drawbacks of split air-conditioners and RCSs, a novel radiant-convective cooling system with RCRT was proposed in this paper. Experiments were conducted to investigate the thermal comfort performance, energy efficiency ratio and economic performance of the proposed system, comparing to split air-conditioners. The following conclusions were drawn:

1) Compared with split air-conditioners, the indoor set temperature could be improved from 26 ^{o}C to 28 ^{o}C to ensure the same indoor thermal comfort when using the RCRT cooling system for provide cooling.

2) Compared with split air-conditioners, the RCRT cooling system had higher energy efficiency ratio, within the range of 0.2 to 0.35, and the energy saving potential of

this system was 8.4%.

3) The RCRT cooling system had higher initial cost and lower life cycle cost compared with split air-conditioners. The payback period of using the proposed system to replace split air-conditioners was 2.4 years, which had excellent economic performance.

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.

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References

[1] S. Hu, Y. Liu, 2020 Annual report on China building energy efficiency, China Architecture & Building Press, China, 2020.

[2] G. Lv, C. Shen, Z. Han, W. Liao, D. Chen, Experimental investigation on the cooling performance of a novel grooved radiant panel filled with heat transfer liquid, Sustainable Cities and Society, 50 (2019) 101638.

[3] Z. Zhang, Y. Zhang, A. Khan, Thermal comfort of people from two types of

air-conditioned buildings – Evidences from chamber experiments, Building and Environment, 162 (2019) 106287.

[4] M.K. Khan, Refrigeration, Air Conditioning and Cold Storage, John Wiley & Sons, Ltd, 2012.

[5] L. Zhang, J. Dong, Y. Jiang, S. Deng, S. Huang, An experimental study on frosting and defrosting performances of a novel air source heat pump unit with a radiant-convective heating terminal, Energy and Buildings, 163 (2018) 10-21.

[6] J. Le Dréau, P. Heiselberg, Sensitivity analysis of the thermal performance of radiant and convective terminals for cooling buildings, Energy and Buildings, 82 (2014) 482-491.

[7] R. Opoku, I.A. Edwin, K.A. Agyarko, Energy efficiency and cost saving opportunities in public and commercial buildings in developing countries – The case of air-conditioners in Ghana, Journal of Cleaner Production, 230 (2019) 937-944.

[8] H.E. Feustel, C. Stetiu, Hydronic radiant cooling–preliminary assessment, Energy and Buildings, 22 (1995) 193-205.

[9] H.M. Koo, An experimental study of the noise and the performance of cross-flow fans in room air-conditioning systems, Noise Control Engineering Journal, 48 (2000) 41-47.

[10] K.-N. Rhee, B.W. Olesen, K.W. Kim, Ten questions about radiant heating and cooling systems, Building and Environment, 112 (2017) 367-381.

[11] H. Wang, H. Shu, L. Duanmu, A Review on Radiant Cooling System in Buildings of China, in: A. Li, Y. Zhu, Y. Li (Eds.), Proceedings of the 8th International Symposium on Heating, Ventilation and Air Conditioning, Springer Berlin Heidelberg, Berlin, Heidelberg, 2014, 101-108.

[12] A. Zarrella, M. De Carli, C. Peretti, Radiant floor cooling coupled with dehumidification systems in residential buildings: A simulation-based analysis, Energy Conversion and Management, 85 (2014) 254-263.

[13] H. Tang, X.-H. Liu, Y. Jiang, Theoretical and experimental study of condensation rates on radiant cooling surfaces in humid air, Building and Environment, 97 (2016) 1-10.

[14] K. Zhao, X.-H. Liu, Y. Jiang, Application of radiant floor cooling in a large open space building with high-intensity solar radiation, Energy and Buildings, 66 (2013) 246-257.

[15] C. Stetiu, Energy and peak power savings potential of radiant cooling systems in US commercial buildings, Energy and Buildings, 30 (1999) 127-138.

[16] J.L. Niu, L.Z. Zhang, H.G. Zuo, Energy savings potential of chilled-ceiling combined with desiccant cooling in hot and humid climates, Energy and Buildings, 34 (2002) 487-495.

[17] Z. Tian, J.A. Love, A field study of occupant thermal comfort and thermal environments with radiant slab cooling, Building and Environment, 43 (2008) 1658-1670.

[18] T. Catalina, J. Virgone, F. Kuznik, Evaluation of thermal comfort using combined CFD and experimentation study in a test room equipped with a cooling ceiling, Building and Environment, 44 (2009) 1740-1750.

[19] T. Imanari, T. Omori, K. Bogaki, Thermal comfort and energy consumption of the radiant ceiling panel system: Comparison with the conventional all-air system, Energy and Buildings, 30 (1999) 167-175.

[20] S.P. Corgnati, M. Perino, G.V. Fracastoro, P.V. Nielsen, Experimental and

numerical analysis of air and radiant cooling systems in offices, Building and Environment, 44 (2009) 801-806.

[21] H. Shu, X. Bie, H. Zhang, X. Xu, Y. Du, Y. Ma, L. Duanmu, G. Cao, Natural heat transfer air-conditioning terminal device and its system configuration for ultra-low energy buildings, Renewable Energy, 154 (2020) 1113-1121.

[22] ANSI/ASHRAE. Standard 55: Thermal Environmental Conditions for Human Occupancy, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA, 2017.

[23] ASHRAE, ANSI/ASHRAE 138: Method of testing for rating ceiling panels for sensible heating and cooling, American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, GA, 2009.

[24] ISO, ISO 5151: Non-ducted air conditioners for heat pumps-Testing and rating for performance, International Organization for Standard, Geneve, Switzerland, 2017.

[25] P.O. Fanger, Thermal comfort, Danish Technical Press, Copenhagen, 1970.

[26] ISO, ISO 7730: Ergonomics of the Thermal Environment-Analytical Determination and Interpretation of Thermal Comfort Using Calculation of the PMV and PPD Indices and Local Thermal Comfort Criteria, International Organization for Standardization, Geneva, Switzerland, 2005.

[27] H. Zhang, L. Jiang, W. Zheng, S. You, T. Jiang, S. Shao, X. Zhu, Experimental study on a novel thermal storage refrigerant-heated radiator coupled with air source heat pump heating system, Building and Environment, 164 (2019) 106341.

[28] Y. He, N. Li, W. Zhang, J. Peng, Overall and local thermal sensation & comfort in air-conditioned dormitory with hot-humid climate, Building and Environment, 101 (2016) 102-109.

[29] M. Frontczak, P. Wargocki, Literature survey on how different factors influence human comfort in indoor environments, Building and Environment, 46 (2011) 922-937.

[30] BS EN 15251: Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics, European Committee for Standardization, Brussels, Belgium, 2007.

[31] W.-H. Chen, H.-E. Mo, T.-P. Teng, Performance improvement of a split air conditioner by using an energy saving device, Energy and Buildings, 174 (2018) 380-387.

[32] W.T. Grondzik, Air Conditioning System Design Manual (Second Edition), Butterworth-Heinemann, Burlington, 2007, 1-7.

[33] J. Wu, C. Liu, H. Li, D. Ouyang, J. Cheng, Y. Wang, S. You, Residential air-conditioner usage in China and efficiency standardization, Energy, 119 (2017) 1036-1046.

[34] C. Luerssen, O. Gandhi, T. Reindl, C. Sekhar, D. Cheong, Life cycle cost analysis (LCCA) of PV-powered cooling systems with thermal energy and battery storage for off-grid applications, Applied Energy, 273 (2020) 115145.

[35] Beijing Municipal Development and Reform Commission, 2020. Notice of Beijing Municipal Development and Reform Commission on relevant concerning the adjustment of the power transmission and distribution price of the city. http://fgw.beijing.gov.cn/fgwzwgk/zcgk/bwqtwj/202011/t20201130_2153823.htm

[36] M. Sun. Technical and economic analysis of two kinds of air conditioning system of office building in a university. Donghua University, 2017.

[37] M. Sun. Study on the applicability of seawater source heat pumps in different climate zones. Tianjin University, 2020.