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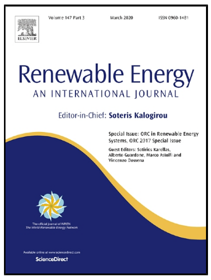

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PII: S0960-1481(19)32031-2
DOI: https://doi.org/10.1016/j.renene.2019.12.152
Reference: RENE 12861

To appear in: *Renewable Energy*

Received Date: 10 April 2019
Accepted Date: 31 December 2019

10. New technologies/systems development and application in built environment
议题：人工环境新技术、新系统发展和应用

Natural Heat Transfer Air-Conditioning Terminal Device and Its System Configuration for Ultra-Low Energy Buildings

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ABSTRACT
In consideration of the lowered heating and cooling load of ultra-low energy buildings, a natural heat transfer air-conditioning terminal device (NHTACTD) is presented by the authors. The terminal device is able to undertake heating, cooling and moisture load of a room according to the inlet water temperature. Its comparative advantages are pointed out by comparing it with radiators, fan coil units, chilled beams and radiant heating and cooling terminals. After the actual thermal properties of the NHTACTD are provided, three air-conditioning system configuration schemes based on the NHTACTDs are presented: (1) In the NHTACTDs plus fresh air system, the NHTACTDs and the fresh air handling unit undertake all the air-conditioning load of the system together, and it is used where high indoor air quality is demanded; (2) In the scheme of the air-conditioning system including the NHTACTDs only, all the air-conditioning load of the system has to be undertaken by the terminals alone and it is used where high indoor air quality is not rigidly demanded. (3) In the scheme of the air-conditioning system including both the NHTACTDs and radiant panels, the NHTACTDs undertake all the moisture load of the system, and the remaining sensible cooling load is undertaken by the radiant panels. The scheme can be used where there is large sensible cooling load while high indoor air quality is not rigidly demanded. Then an ultra-low energy residential building is taken as a case project to elaborate the design method of an air-conditioning configuration scheme based on the terminal device with the help of psychrometric chart.

Keywords: Air-conditioning; Ultra-low energy buildings; Terminal device; Natural heat transfer

Nomenclature

Q_c	cooling capacity per heat transfer pipe of the NHTACTD, kW
Q_h	heating capacity per heat transfer pipe of the NHTACTD, kW
t_1	inlet fluid temperature, °C
t_2	outlet fluid temperature, °C
t_{air}	indoor air temperature, °C
t_r	reference indoor air temperature, °C

t_{sur}	surface temperature of the NHTACTD, °C
Δt	excess temperature, °C
φ	moisture condensation rate on the surface of the NHTACTD, kg/s
ϕ	relative humidity of indoor air

1. Introduction

Ultra-low energy buildings represent the development trend of future buildings because of their characteristics of low energy consumption and high indoor thermal comfort. Therefore more and more researches are being done in the field worldwide[1-8]. The proportion of the building energy consumption to the total energy consumption of the society is continually increasing and it is more than one third in some countries now[9]. Of the whole building energy consumption, the heating, ventilation and air-conditioning (HVAC) system consumes a major part (at least a half). As it is known that the air-conditioning terminal devices of an HVAC system not only dissipate almost all the heating/cooling energy, but also play an important role in the occupants' thermal comfort, so here, the focus is laid on the air-conditioning terminal devices for ultra-low energy buildings. Apart from the air-conditioning terminals that are currently used in buildings like fan coil units, radiant heating/cooling panels, chilled beams, etc., the authors brought out a radiant–convective air-conditioning terminal device [10] that is especially suitable for the buildings featuring low heating/cooling load and high indoor thermal comfort. The terminal air-conditioning device presented by the authors is shown in Figure 1 and it mainly consists of multiple heat transfer pipes in vertical parallel arrangement with a condensate discharge pan underneath. The material of the heat transfer pipes is aluminium alloy which has excellent thermal conductivity. As the heat transfer between the terminal device and the ambient environment is carried out by natural convection and radiation which are the common heat transfer methods in the natural world that our humans have already become accustomed to, the air-conditioning terminal device characterized by these two heat transfer modes are called natural heat transfer air-conditioning terminal devices (NHTACTD) here. This kind of air-conditioning terminal device operates without noise and little disturbance to the indoor air, which is very helpful to enhance the occupants' thermal comfort. The terminal device supplies heating or cooling according to the inlet water temperature, and moisture condensation on its surface is also allowed, as there is a condensate discharge pan under the device, so the terminal device is able to undertake the heating, cooling and moisture load of a room all year round.

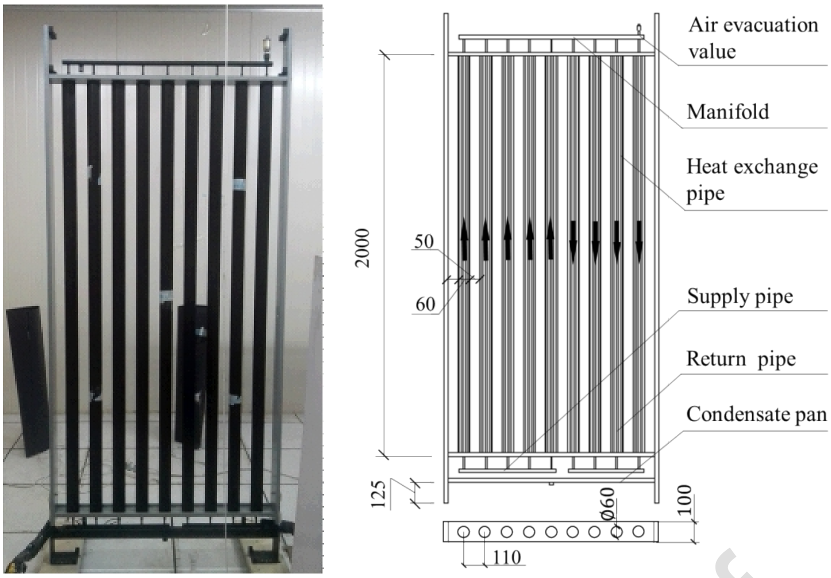


Figure 1. The natural heat transfer air-conditioning terminal device

Except literature [10], few research was found on such kind of air-conditioning terminal device that can supply both space heating and cooling (sensitive and latent) only by radiation and natural convection heat transfer. So comparison will be made between the NHTACTD and other air-conditioning terminals currently used in the HVAC systems and the comparative advantages of the NHTACTD will be specially emphasized. The air-conditioning terminals currently used in HVAC systems include radiators, fan coil units, chilled beams and radiant heating/cooling terminals (e.g., radiant floor or ceiling).

(i) Radiators are commonly used in the building space heating systems with various types[11]. However they only provide space heating but are unable to undertake any cooling load of a building. In other words, other air-conditioning terminal devices should be designed in parallel with the radiators so as to undertake the cooling load of the building. However, in theory, the NHTACTDs can undertake both the space heating and cooling load of the building all year round without other supplementary terminal devices.

(ii) Fan coil units are the most commonly used air-conditioning terminal devices in HVAC systems at present. They can supply both heating and cooling (both sensible and latent cooling) for a room according to the inlet water temperature. Compared with a fan coil unit, the NHTACTD has apparently much lower heating and cooling capacity under the same size because the forced air convection dominates the heat transfer of a fan coil unit that has a quite large heat transfer coefficient. Nevertheless, the NHTACTD supplies heating and cooling with little air disturbance and is noise free without forced air convection, and these help meet the occupants' demand for comfortable indoor thermal environment.

(iii) Chilled beams are used to provide only sensible cooling according to the available research. There are two types of chilled beams which are passive chilled beams and active chilled beams. Passive chilled beams use typical fin-tube heat exchangers and a horizontal perforated panel at the bottom to increase both radiative and convective heat transfer[12]. Since there are no mechanical fans in the passive chilled beams, they are able to supply cooling with little air disturbance and are noise free too. Active chilled beams supply conditioned air via an array of air nozzles and induce room air movement through the beams[13]. The conditioned air in the active chilled beam systems is driven by fans, so forced air convection and noise cannot be avoided. But for both types of chilled beams, moisture condensation is not allowed on their surfaces, so usually a parallel ventilation system (e.g., dedicated outdoor air system) should be used to handle the latent cooling load (i.e., moisture load) which makes the whole system more complicated and expensive[12][14]. However, the NHTACTD not only is capable of handling both the sensible and latent cooling load of a room, but also makes the HVAC system adopting the NHTACTD more concise and less expensive.

(iv) Radiant heating and cooling terminals (e.g., radiant floor or ceiling) also operate with little air disturbance and are noise free, but they can only supply heating and sensible cooling for a room without latent cooling. That is to say, another moisture removal system must be equipped to make it a complete air cooling system that can undertake both sensible and latent cooling of a room[14~18]. So the comparative advantage of the NHTACTD studied in the paper is that it is able to undertake both the sensible and latent cooling load of a room without the need of other dehumidification systems.

From above comparison and analysis, the air-conditioning terminal device studied in the paper has its unique advantages leading to optimistic application prospects in ultra-low energy buildings or Nearly Zero Energy Buildings (NZEBs). In the paper, the thermal performances of the NHTACTD and its calculation models of heating, cooling and dehumidifying capacities will be introduced. Then three air-conditioning system configuration schemes based on the NHTACTD are put forward and their application situations are pointed out. After that, the design method of an air-conditioning system based on the NHTACTD is elaborated through a case study of an ultra-low energy residential building. In the end, a brief summarization is made for the research.

2. Thermal characteristics of the NHTACTD

As mentioned in the introduction, the terminal device supplies heating or cooling for a room according to the inlet water temperature, and it can undertake both the sensible and latent cooling load. The radiation and natural convection heat transfer of the terminal device make it operate in a room with little air disturbance and noise free, but this also leads to the decrease in its heating and cooling capacity. So experiments have been done to test its thermal performances[19] in the environment chamber of Dalian University of Technology.

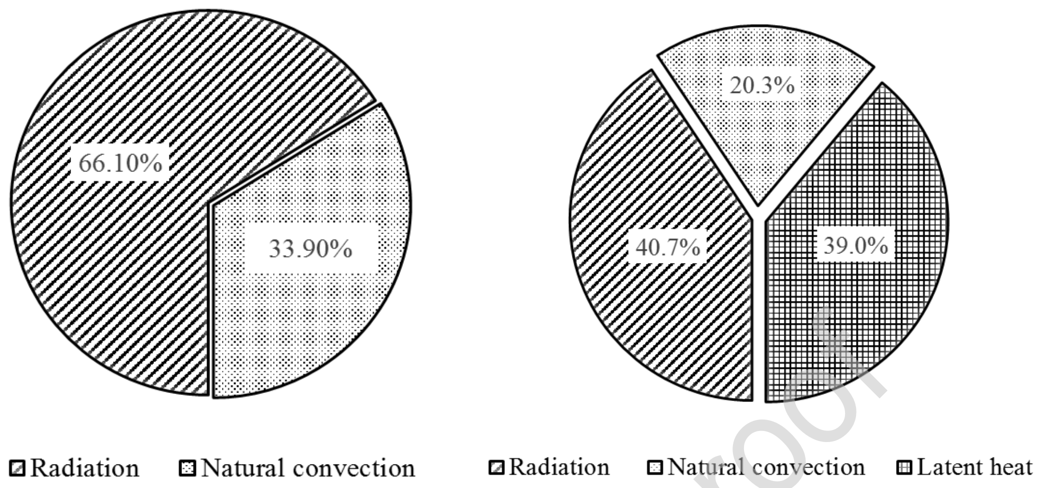
2.1 Heating capacity of the NHTACTD

In order to ascertain the heating capacity of the NHTACTD, various space heating experiments were conducted under different supply temperatures and flow rates to keep the indoor air temperature at 18°C or 20°C. The experimental data under the heating mode are shown in Table 1.

Table 1 Experimental data under the heating mode

No.	Inlet temperature (°C)	Outlet temperature (°C)	Flow rate (L/h)	Indoor air temperature (°C)	Average surface temperature (°C)	Heat capacity (W)
1	49.7	42.5	119.3	19.5	44.4	770.2
2	50.1	45.2	180.1	19.3	45.8	809.0
3	49.3	45.4	244.9	19.8	45.5	862.1
4	44.9	41.4	240.6	18.0	41.7	766.6
5	40.5	37.9	243.2	18.6	38.0	568.8
6	35.5	32.8	247.2	18.8	33.2	590.8
7	31.4	29.0	246.5	18.3	29.9	533.2
8	51.5	43.5	128.0	18.1	45.4	931.3
9	45.5	38.7	127.2	18.7	40.6	779.2
10	40.2	34.5	124.2	20.2	37.8	641.3
11	35.9	30.7	125.3	18.1	32.5	585.5

According to the calculation models of the radiation and natural convection heat transfer as well as the experimental data[19], the percentages of radiation and natural convection in the total heat transfer of the terminal device are shown in Figure 2(a). It is obvious that the radiation dominates the heat transfer of the terminal device in the heating mode (i.e,, 66.1%).



(a) Heating mode (b) Cooling mode

Figure 2. The percentage of different type of heat transfer in the total heating/cooling capacity of the NHTACTD

Through data regression, the calculation model of the heating capacity per heat transfer pipe of the terminal device under the flow rate of 240L/h is expressed as:

$$Q_h = \frac{87.51\Delta t^{0.7}}{9} \tag{1}$$

2.2 Cooling capacity of the NHTACTD

Similar to the heating mode experiments, various cooling performance experiments were carried out under different inlet temperatures and flow rates to keep the indoor air temperature at 26°C and relative humidity at 60% or 80%. The experimental data of the NHTACTD under the cooling mode are shown in Table 2.

Table 2 Experimental data under the cooling mode

No.	Inlet temperature (°C)	Outlet temperature (°C)	Flow rate (L/h)	Indoor air temperature (°C)	Relative humidity of indoor air (%)	Cooling capacity (W)	Moisture removal (ml/h)
1	12.8	15.9	124.1	26.5	54.4	349.1	18
2	8.3	13.0	120.2	26.4	60.8	512.6	216
3	10.9	14.3	118.8	26.7	59.6	395.4	135
4	13.3	18.1	119.4	26.0	80.6	520.0	260
5	15.9	19.9	121.1	25.8	80.5	439.5	113
6	18.5	21.3	120.4	26.3	76.1	305.9	40
7	10.7	15.6	124.1	27.7	66.5	556.8	249
8	13.9	17.9	126.1	28.1	63.4	453.1	138
9	7.7	10.5	235.4	26.2	61.2	598.1	273

10	13.8	15.6	181.8	26.4	58.7	304.6	72
11	13.9	15.3	238.6	26.6	61.4	368.1	94
12	9.8	12.4	253.2	26.1	62.8	597.3	222
13	12.1	14.0	245.8	26.3	58.0	423.8	108
14	15.6	16.6	243.3	26.3	58.6	242.8	41

Similar to the heating mode, the percentage of different types of heat transfer in the total cooling capacity of the NHTACTD is calculated on the basis of radiation and natural convection heat transfer calculation models and the results are shown in Figure 2(b). It can be seen from the figure that the proportion of radiation heat transfer to the total heat transfer (or the cooling capacity) is almost the same as latent heat transfer (i.e., 40.7% and 39.0% respectively), and the natural convection accounts for a relatively lower proportion of the total heat transfer (i.e., 20.3%). So the terminal device can be regarded as a radiation-dominated air-conditioning terminal device.

Through data regression, the calculation model of the cooling capacity per heat transfer pipe of the terminal device under the flow rate of 240L/h is expressed as:

$$Q_c = \frac{7.82\Delta t^{1.55}}{9} \tag{2}$$

In Equations (1) and (2), Q_h is the heating capacity per heat transfer pipe of the NHTACTD, kW; Q_c is the cooling capacity per heat transfer pipe of the NHTACTD, kW; Δt is the excess temperature, °C, which is the absolute temperature difference between the mean surface temperature of the NHTACTD and the indoor air temperature, and it is expressed as:

$$\Delta t = \left| \frac{t_1 + t_2}{2} - t_r \right| \tag{3}$$

in which, t_1 is the inlet water temperature of the NHTACTD, °C; t_2 is the outlet water temperature of the NHTACTD, °C; t_r is the indoor air temperature, °C.

2.3 Moisture removal capacity of the NHTACTD

In respect of moisture removal capacity of the terminal device, equation (4) can be used to calculate the moisture condensation rate on the surface of the NHTACTD after experimental data collection and regression[19]:

$$\varphi = 5.21 \times 10^{-6} \times \left(t_{air} - t_{surf}\right)^{0.39} \times \left(\frac{\phi}{100} \exp\left(\frac{17.269 \cdot t_{air}}{t_{air} + 237.3} \right) - \exp\left(\frac{17.269 \cdot t_{surf}}{t_{surf} + 237.3} \right) \right) \tag{4}$$

in which, φ is the moisture condensation rate on the surface of the NHTACTD, kg/s; ϕ is the relative humidity of the indoor air; t_{air} is the indoor air temperature, °C; t_{sur} is the average surface temperature of the NHTACTD, °C.

2.4 Comparison of heating and cooling capacity between the NHTACTD and radiant floor heating and cooling terminal

Since the NHTACTD can be regarded as a radiation-dominated air-conditioning terminal device, comparison can be made with the radiant heating and cooling terminals in respect of heating and cooling capacity. Taking the radiant floor heating and cooling as an example, the heat transfer intensities are calculated and compared with the NHTACTD. Normally speaking, the two kinds of air-conditioning terminals should be compared under the same operation conditions which are the same inlet water temperature and flow rate. But for the radiant floor heating and cooling system, the surface temperature of the floor should not exceed 29°C[20] in the heating mode in order to avoid thermal discomfort of the occupants and should not be lower than 18°C in the cooling mode so as to prevent moisture condensation on the surface. However, the NHTACTD does not have these limitations and the standard heating operation conditions of the NHTACTD are selected as 50°C for the inlet water temperature and 240L/h for the water flow rate while the standard cooling operation conditions are selected as 13°C for the inlet water temperature and 240L/h for the water flow rate. So comparison is made under these conditions, and the comparison results of the two air-conditioning terminals are shown in Table 3.

Table 3 Comparison of heat transfer intensities between the two air-conditioning terminals

Operation mode	Heat transfer intensity per unit projection area（W/m^2）		Enhanced percentage of NHTACTD（%）
	Floor heating and cooling	NHTACTD	
Heating	109.1	180.9	65.8%
Cooling	55.5	78.5	41.5%

It is shown in Figure 3 that the heat transfer intensity of the NHTACTD is enhanced by 65.8% and 41.5% in heating and cooling mode respectively compared with the radiant floor heating and cooling terminal.

3. Configuration schemes of the air-conditioning system based on the NHTACTD

In consideration of the thermal characteristics of the terminal device and different project conditions as well as the users' requirements, different configuration schemes of the air-conditioning system based on the NHTACTD can be established and the following three schemes are summarized here:

- Scheme 1: The NHTACTDs plus fresh air system
- Scheme 2: The air-conditioning system with the NHTACTDs only
- Scheme 3: The air-conditioning system with the NHTACTDs and radiant panels

Next, the above three schemes will be elaborated under the cooling mode.

3.1 Scheme 1: The NHTACTDs plus fresh air system

The system consists of the NHTACTDs and the fresh air handling system whose schematic diagram is shown in Figure 3. The chilled water from the supply water manifold is divided into two routes, one is for the NHTACTDs and the other is for the fresh air handling units. The supply water temperatures for the two air handling units (i.e., the NHTACTDs and fresh air handing units) can be the same or different according to the detail conditions of the project.

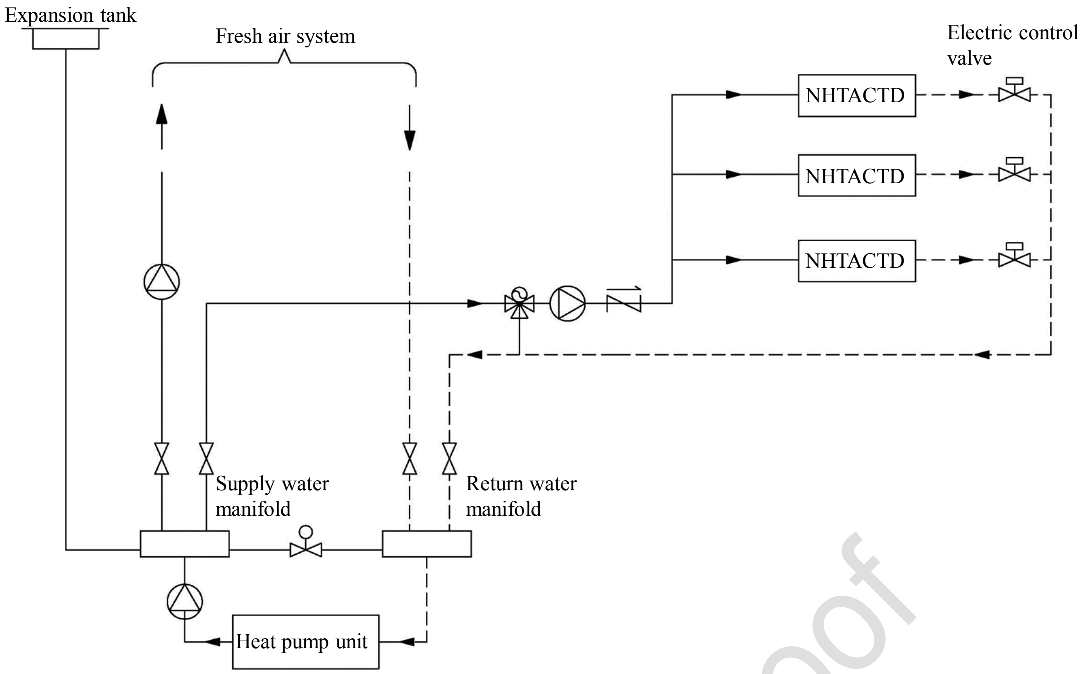


Figure 3. Schematic diagram of the NHTACTDs plus fresh air system

According to the fresh air treatment method, there are various system design schemes. For instance, as shown in Figure 4, the fresh air is cooled to the apparatus dew-point temperature (i.e., the relative humidity of the air reaches 95%) with the same moisture content of the indoor air, and Figure 4 displays the air treatment process in the psychrometric chart where **h** and **d** stand for the enthalpy (kJ/kg) and moisture content (g/kg) of the air respectively. In such circumstance, the fresh air handling unit undertakes the whole moisture and cooling load of the fresh air and part of the indoor cooling load as well. And the remaining air-conditioning load (i.e., the whole indoor air moisture load and the residual part of the indoor cooling load) is undertaken by the NHTACTD.

Since there is an independent fresh air treatment sub-system in the whole air-conditioning system, it is usually applied in the occasions where high indoor air quality is required.

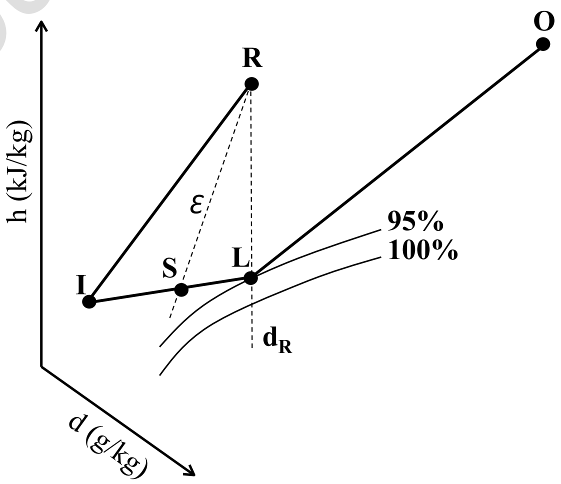


Figure 4. Air treatment process in the psychrometric chart of Scheme 1

3.2 Scheme 2: The air-conditioning system with the NHTACTDs only

The schematic diagram of the scheme is shown in Figure 5. All the chilled water prepared by the chiller is supplied to the NHTACTDs. As there is no fresh air handling sub-system in the scheme, it can be applied in occasions where high indoor air quality is not rigidly demanded and the air conditioning load is relatively small.

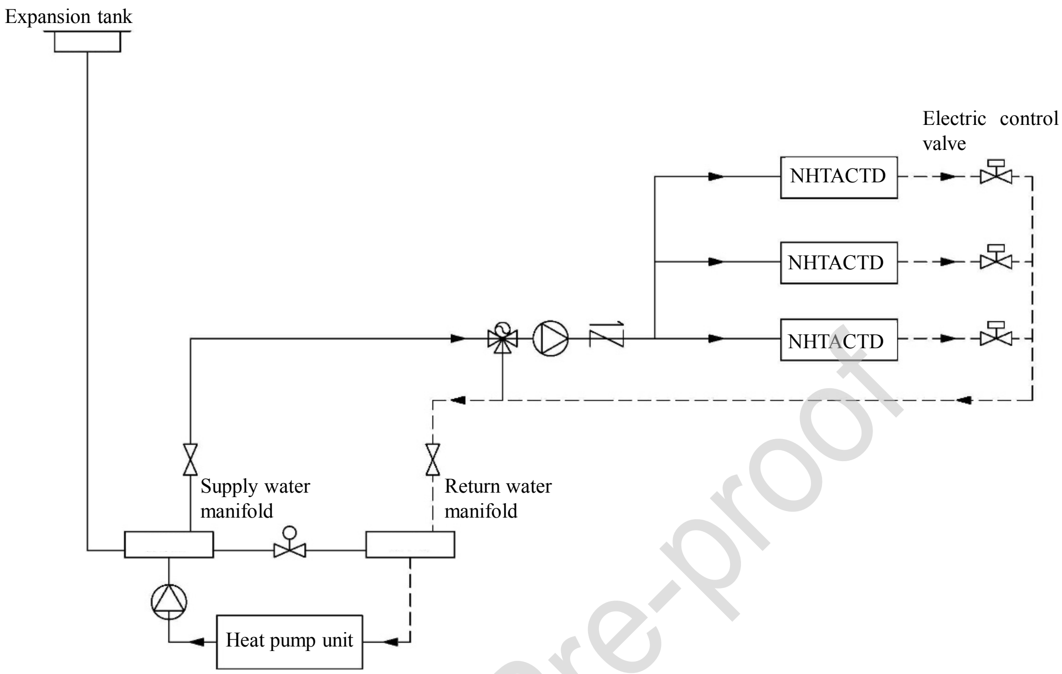


Figure 5. Schematic diagram of the air-conditioning system with the NHTACTDs only

As the NHTACTD is the only air handling unit in the scheme, so all the air-conditioning load should be undertaken by the terminal device. If the number of NHTACTDs is calculated according to the moisture load of the system, the cooling capacity of the terminal devices should be verified to ensure that the terminal devices are able to undertake both the moisture and cooling load at the same time.

3.3 Scheme 3: The air-conditioning system with the NHTACTDs and the radiant panels

On occasions where high indoor air quality is not rigidly demanded but the sensible cooling load is relatively large, the scheme of the air-conditioning system with the NHTACTDs and the radiant panels (i.e., scheme 3) can be adopted. The schematic diagram of the scheme is shown in Figure 6 where the chilled water from the supply water manifold is divided into two routes, one is for the NHTACTDs and the other is for the radiant panels. Usually the supply water temperatures of the two air-conditioning terminals are not the same because the supply water temperature of the NHTACTDs is lower than the dew point temperature of indoor air in order to undertake the moisture load while the supply water temperature of the radiant panels should be higher than the dew point temperature of indoor air so as to prevent moisture condensation on the panels' surfaces.

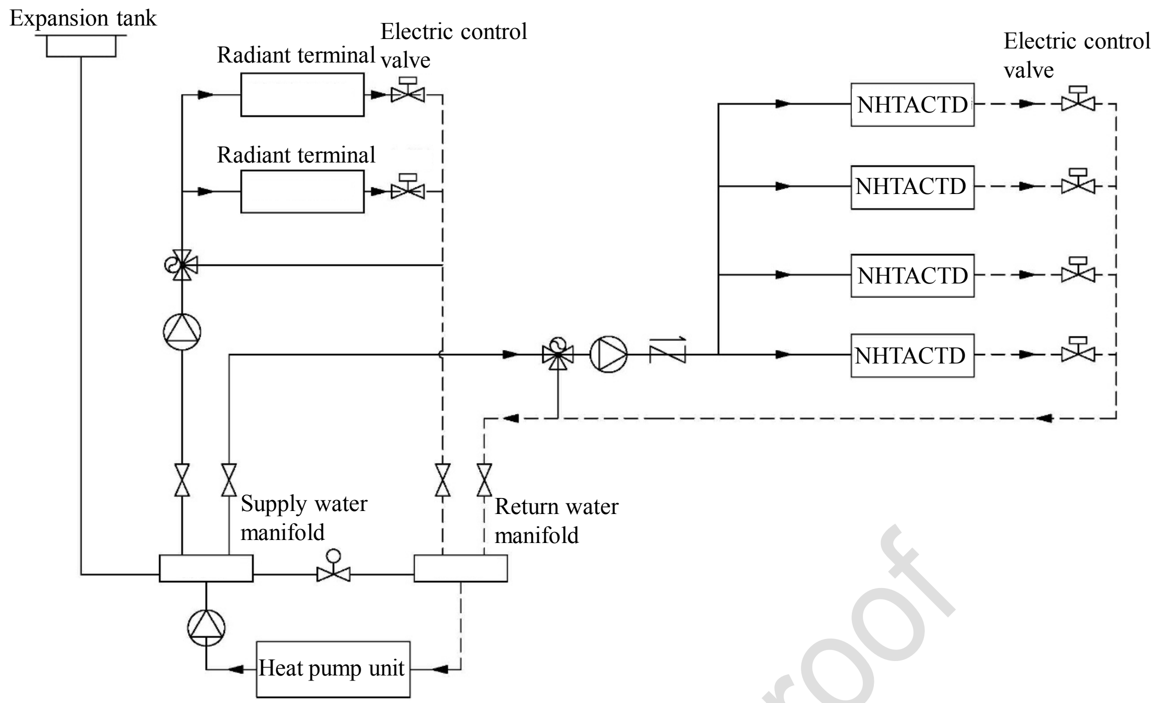


Figure 6. Schematic diagram of the air-conditioning system with the NHTACTDs and the radiant panels

In the design of the scheme, the NHTACTDs should undertake all the moisture load and at the same time part of the sensible cooling load is undertaken by the terminal devices too. For the remaining sensible cooling load, this should be undertaken by the radiant panels.

4. Case study

Here an ultra-low energy residential building is taken as a case study and a room of the building is selected as an example to show how the number of NHTACTDs is determined after the configuration scheme of the air-conditioning system based on the NHTACTDs is adopted.

4.1 Brief introduction of the project

The residential building is located in Dalian city (in the cold climate zone) of China with a gross floor area of 3,415m^2. There are six storeys in the building with each storey's height of 3.0m. The standard floor plan of the building displays in Figure 7.

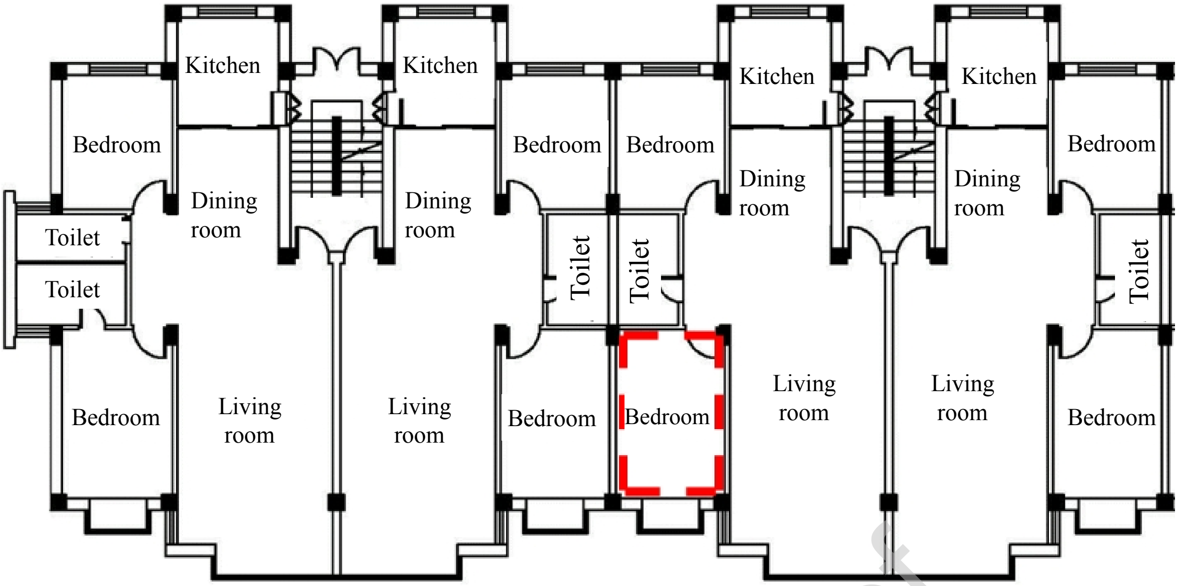


Figure 7. Standard floor plan of the residential building

Comparisons are made between the design values of the thermal properties of the building envelope and the corresponding specified values in 'Technical Guidelines for Ultra-Low Energy Green Buildings (Residential Buildings)'[21], and the results are listed in Table 4. It shows that all the design values of the thermal properties of the building envelope satisfy the requirements of the guidelines.

Table 4. Comparisons of the thermal properties of the building envelope between the specified values in the Guidelines and the corresponding design values

	Design values	Specified values in Guidelines
U value of external wall ($W/(m^2 \cdot K)$)	0.177	0.1~0.25
U value of roof ($W/(m^2 \cdot K)$)	0.218	0.1~0.25
U value of ground ($W/(m^2 \cdot K)$)	0.185	0.15~0.35
U value of window ($W/(m^2 \cdot K)$)	1.0	0.80~1.50
SHGC of window	0.45 (Winter)	≥0.45 (Winter)
	0.30 (Summer)	≤0.30 (Summer)

Note: SHGC stands for solar heat gain coefficient.

The hourly meteorological data of Dalian city in DeST (Designer's Simulation Toolkit, a software developed by Qinghua University of China) software is used to calculate the air-conditioning load of the building. The design values of the indoor air are listed in Table 5.

Table 5. Design values of indoor air

	Winter	Summer
Indoor air temperature（°C）	≥20	≤26
Tolerant temperature（°C）	20	28
Relative humidity（%）	≥30	≤60
Fresh air ($m^3/(h \cdot p)$)	≥30	

For the city of Dalian, the heating season is from November 5th to March 5th next year and the cooling season is from June 1st to August 31st.

4.2 Calculation results of the heating, cooling and moisture load of the building

The simulation results of the air-conditioning load of the building are obtained by utilizing DeST software. The annual cumulative heating load intensity is 11.71 kWh/(m^2·a) and the annual cumulative cooling load intensity is 5.80 kWh/m^2(m^2·a) for the case building, and both satisfy the requirements of the Guidelines (i.e., ≤15kWh/(m^2·a) and ≤12 kWh/(m^2·a) respectively).

As for the air-conditioning system, the scheme of the NHTACTDs plus fresh air system is adopted for the building, and a bedroom (circled with red dotted line in Figure 7) with the floor area of 13.1m^2 is selected as an example to elaborate the design process of the air-conditioning scheme. The whole air-conditioning load of the bedroom is divided into two parts: one is the indoor air-conditioning load including the load from the building envelope, the load caused by the occupants, lighting and other devices or equipment that dissipate heat into the indoor air; the other is the air-conditioning load caused by the fresh air delivered into the bedroom. The calculated air-conditioning load of the bedroom is listed in Table 6.

Table 6. Design air-conditioning load of the bedroom

	Cooling load (W)	Moisture load (g/h)	Heating load (W)
Indoor air-conditioning load	535.9	240.0	768.6
Air-conditioning load of fresh air	355.6	432.0	714.0

4.3 Design method for the configuration scheme of the NHTACTDs plus fresh air system

(1) Allocation of the air-conditioning load

The fresh air volume for the bedroom is 60 m^3/h and the fresh air treatment scheme is shown in Figure 4. That is, the fresh air (i.e., the outdoor air) is treated to the apparatus due point temperature with the same moisture content as the indoor air. The thermophysical properties of outdoor and indoor air for the air-conditioning system design in cooling mode are listed in Table 7. Calculation results show that if the outdoor fresh air is processed from point **O** to **L** (refer to Figure 4), the fresh air handling unit will undertake all the moisture load of fresh air (432g/h) and all the cooling load of fresh air (355.6W) and part of the indoor cooling load (155.2W) as well. Then the NHTACTD should undertake the indoor air moisture load (240.0g/h) and the remaining indoor cooling load (380.7W).

Table 7. Thermophysical properties of outdoor and indoor air under the design conditions

Outdoor air	Dry-bulb temperature (°C)	Wet-bulb temperature (°C)	Enthalpy (kJ/kg)	Moisture content (g/kg)
	28.4	25.0	76.55	18.81
Indoor air	Dry-bulb temperature (°C)	Relative humidity (%)	Enthalpy (kJ/kg)	Moisture content (g/kg)
	26	60	58.77	12.81

(2) Design temperatures of water supply and return for heating and cooling

The design temperatures of chilled water supply and return are 7°C /12°C, and the design indoor air temperature and relative humidity are 26°C and 60% respectively in the cooling mode. The design temperatures of hot water supply and return are 45°C /40°C with the design indoor air temperature of 20°C in the heating mode.

(3) Selection of the NHTACTD

Since the moisture load of fresh air is undertaken by the fresh air handing unit, the indoor air moisture load should be undertaken by the NHTACTD. The moisture removal capacity of the terminal device is calculated to be 28.65g/h per heat transfer pipe from Equation (4). So nine heat transfer pipes are needed to remove the moisture load of 240g/h.

(4) Verification of the cooling capacity of the selected terminal device

The cooling capacity of the terminal device is calculated to be 67.0W per heat transfer pipe from

Equation (2). So nine heat transfer pipes can provide a cooling load of 603.0W which is larger than the required cooling load of 355.6W. That means the selected terminal device can not only undertake the moisture load of the indoor air but also the required cooling load of the room.

(5) Verification of the heating capacity of the selected terminal device

As the heating load of fresh air is undertaken by the fresh air handling unit, so the terminal device should take up the indoor heating load. The heating capacity of the terminal device is calculated to be 86.0W per heat transfer pipe from Equation (1). So nine heat transfer pipes can provide heating load of 773.7W which is also larger than the required heating load of 768.6W.

In conclusion, the terminal device selected by the indoor air moisture load can also satisfy the cooling and heating demand in the case study.

5. Discussion

As the heat transfer between the NHTACTD and the indoor environment is carried out by radiation and natural convection without forced air convection, so the cooling and heating capacity of the terminal device is much less than a fan coil unit with the same size. Thus the NHTACTD is suitable for such occasions where there is low cooling/heating load (e.g., ultra-low energy buildings), otherwise the number of the terminal devices would be much bigger and it would be difficult to find enough space to place them.

For the situations where the design air-conditioning load is quite large, but the relatively low partial load dominates the operation period, the NHTACTDs can be designed together with such conventional terminal devices as the fan coil units. And in such circumstances, the NHTACTDs can operate alone during the low air-conditioning load period while the fan coil units should operate together during the heavy air-conditioning load period.

In fact, the NHTACTDs already have some actual applications (e.g., small offices and apartments, etc.). The room temperature can be kept at 25~26°C and the relative humidity under 70%. The most significant advantage of the terminal device is the comfortable experience without any draft risks.

6. Conclusions

For the ultra-low energy buildings characterized by low heating/cooling load but high indoor thermal comfort, a natural heat transfer air-conditioning terminal device is presented and its actual thermal properties are summarized. Then three configuration schemes of the air-conditioning system based on the terminal device are further presented and studied in the paper, which provides a brand new solution to the air-conditioning system for ultra-low energy buildings. The main conclusions of the research are listed below:

(1) The natural heat transfer air-conditioning terminal device presented by the authors can provide heating in winter and cooling in summer, and it also allows moisture condensation on its surface. In other words, it can be used to undertake the heating, cooling and moisture load of a building. The calculation models of its thermal properties are briefly summarized.

(2) Three configuration schemes of the air-conditioning system based on the terminal device are presented for different situations: 1) The NHTACTDs plus fresh air system for the high indoor air quality situations; 2) The air-conditioning system with the NHTACTDs only can be used where high indoor air quality is not rigidly demanded and the air conditioning load is relatively small; 3) The air-conditioning system with the NHTACTDs and radiant panels can be adopted where high indoor air quality is not rigidly demanded but the sensible cooling load is relatively large.

(3) The design method of an air-conditioning configuration scheme based on the terminal device (i.e., the NHTACTDs plus fresh air system) is presented and elaborated through a case study, and verifications are made to demonstrate that the selected terminal device can satisfy the cooling and heating requirements as

well as the moisture removal demand of the room according to the thermal properties of the terminal device.

Acknowledgments

This work is supported by China National 13th Five-Year Plan of Key Research and Development Program "The technical system and key technologies development of nearly zero-energy buildings" (2017YFC0702600). The authors also extend their gratitude to David Johnston from Northumbria University of UK for his effort on the careful proof reading of the paper.

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Authors' contribution

Haiwen Shu designed the study and drafted the manuscript. Hongliang Zhang, Xiaoyue Xu, Yu Du and Yi Ma carried out the experiment and collected the data. Xu Bie, Lin Duanmu and Guangyu Cao coordinated the data-analysis.

This work is supported by China National 13th Five-Year Plan of Key Research and Development Program "The technical system and key technologies development of nearly zero-energy buildings" (2017YFC0702600).

Highlights

1. A novel air-conditioning terminal device is put forward and described.
2. The actual heating and cooling capacities of the terminal device are provided.
3. Three system configuration schemes based on the device are established.
4. A design method of the device in a scheme is elaborated through a case study.