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# New concepts and approach for developing energy efficient buildings: Ideal specific heat for building internal thermal mass

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ABSTRACT

The shortcomings or limitations of the traditional approach to developing energy efficient buildings are that they can not determine: (1) the ideal thermophysical properties of building envelope material, where "ideal" means that such material can use ambient air temperature variation and/or solar radiation efficiently to keep the indoor air temperature in the thermal comfort range with no additional space heating or cooling; (2) the best natural ventilation strategy; (3) the minimal additional energy consumption for space heating in winter or air-conditioning in summer. To overcome these problems, some new concepts for developing energy efficient buildings are put forward in this paper. They are the ideal thermophysical properties of the building envelope material, the ideal natural ventilation rate, and a minimal additional space heating or cooling energy consumption. A new approach for determining these properties is also developed. In contrast to the traditional approach (the thermophysical properties of building envelope material are known and constant so that the relating equations describing the indoor air temperature tend to be linear differential equations), the new approach solves the inverse problem (thermophysical properties, etc. of a buildings are unknown), whose solution can be a function instead of a value. As a first step, the ideal specific heat of the building envelope material for internal thermal mass is analyzed for buildings located in various cities in different climatic regions of China, such as Beijing, Shanghai, Harbin, Urumchi, Lhasa, Kunming and Guangzhou. We found that the ideal specific heat is composed of a basic value and an excessive one which is of $\delta$ function for the cases studied. Some limitations that would need further study are introduced in the end of the paper.



## 1. Introduction

In many countries, energy consumption of buildings is 25–40% of the total energy consumption, in which most of the energy is used for space heating or cooling (i.e., air-conditioning) of buildings. In China, the space heating or cooling energy consumption is about 15% of the total energy consumption and the amount is increasing [1]. Because of China's rapid urbanization and the tremendous demand for new buildings, over 1 billion square meters of new buildings have been built per year since 1996. The huge energy consumption for cooling buildings not only demands valuable fossil fuel resources, but also emits a huge amount of $CO_2$ and other pollutants into the atmosphere. Therefore, studies related to energy efficient buildings are of great importance in China.

Traditionally, architects design a building according to the client specification for space and function. In the process, they seldom fully consider energy efficiency since they pay much more attention to the aesthetics and are lacking in the knowledge and skills necessary to develop energy efficient buildings. The task of making the building energy efficient is then placed on the shoulders of the researchers and engineers in the fields of building physics and/or heating, ventilating and air-conditioning (HVAC). The traditional process of space heating or cooling system design for a given building is as follows: (1) knowing the thermophysical properties of the building's envelope material which tend to be constant; (2) calculating the heating or cooling load for the building; (3) determining the suitable space heating or air-conditioning type and equipment; (4) optimizing the operative modes for HVAC system to minimize their energy consumption [2–4]. In brief, the known conditions in the traditional approach mentioned above are: climate condition, building geometry, building envelope material with known thermophysical properties such as thermal conductivity, specific heat, etc. and the parameters to be found are: indoor air temperature, additional space heating load in winter or space cooling load in summer, etc. (Fig. 1(a)).

The shortcomings of the traditional process are that they can not determine: (1) the ideal thermophysical properties of the building's envelope material, where "ideal", defined by our previous paper [5], means that such material can use ambient air temperature varia-

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**Nomenclature**

| | |
|---|---|
| $A_{p,j}$ | area of two plates in the two-plate model ($j$ = 1, 2) ($m^2$) |
| $A_{win}$ | area of window ($m^2$) |
| $ACH$ | air change per hour ($h^{-1}$) |
| $c_p$ | specific heat ($J\,kg^{-1}\,{}^{\circ}C^{-1}$) |
| $c_{p,ex}$ | excessive specific heat ($J\,kg^{-1}\,{}^{\circ}C^{-1}$) |
| $c_p^*$ | optimized specific heat ($J\,kg^{-1}\,{}^{\circ}C^{-1}$) |
| $c_{p,0}$ | basic specific heat ($J\,kg^{-1}\,{}^{\circ}C^{-1}$) |
| $h$ | convective heat transfer coefficient ($W\,m^{-2}\,{}^{\circ}C^{-1}$) |
| $h_{1,2}$ | overall heat transfer coefficient between the two glass layers ($W\,m^{-2}\,{}^{\circ}C^{-1}$) |
| $h_{out}^*$ | overall heat transfer coefficient at outer surface ($W\,m^{-2}\,{}^{\circ}C^{-1}$) |
| $h_{in}^*$ | overall heat transfer coefficient at inner surface ($W\,m^{-2}\,{}^{\circ}C^{-1}$) |
| $H_{ex}$ | excessive specific enthalpy ($J\,kg^{-1}$) |
| $I$ | integrated degree of discomfort (${}^{\circ}C\,h$) |
| $k$ | thermal conductivity ($W\,m^{-1}\,{}^{\circ}C^{-1}$) |
| $U_{win}$ | overall heat transfer coefficient of window ($W\,m^{-2}\,{}^{\circ}C^{-1}$) |
| $L$ | plate thickness (m) |
| $q$ | heat flux ($W\,m^{-2}$) |
| $q_{r,1}$ | thermal radiation heat fluxes subject to outer glass layer ($W\,m^{-2}$) |
| $q_{r,2}$ | thermal radiation heat fluxes subject to inner glass layer ($W\,m^{-2}$) |
| $q_{r,in}$ | indoor thermal radiation heat fluxes (from indoor thermal source, the other internal surfaces of building envelopes and the solar heat through the window) ($W\,m^{-2}$) |
| $q_{r,out}$ | outdoor thermal radiation heat fluxes ($W\,m^{-2}$) |
| $q_{r,p}$ | thermal radiation heat flux between external plate and internal plate ($W\,m^{-2}$) |
| $q_{r,tran}$ | solar radiation heat fluxes through the south-facing window ($W\,m^{-2}$) |
| $q_{r,win}$ | thermal radiation heat flux between window and internal plate ($W\,m^{-2}$) |
| $Q$ | additional space heating/cooling rate (W) |
| $Q_D$ | indoor thermal disturbance (W) |
| $Q_{D,C}$ | convection heat transfer rate from the indoor heat sources (W) |
| $Q_L$ | heat transfer rate by air leakage or natural ventilation (W) |
| $Q_{p,j}$ | convection heat transfer rate between indoor air and two plates ($j$ = 1, 2) (W) |
| $Q_{r,p}$ | thermal radiation heat transfer rate between external plate and internal plate (W) |
| $Q_{r,win}$ | thermal radiation heat transfer rate between window and internal plate (W) |
| $Q_{win}$ | convection heat transfer rate between indoor air and window (W) |
| $t$ | temperature (°C) |
| $t_H$ | higher limit temperature in the thermal comfort range (°C) |
| $t_L$ | lower limit temperature in the thermal comfort range (°C) |
| $\Delta t$ | temperature segment (°C) |
| $\Delta T$ | effective temperature variation range of the thermal mass (°C) |
| $V_R$ | volume of the room ($m^3$) |
| $\Delta x$ | glass layer thickness (m) |

*Subscripts*

| | |
|---|---|
| 1 | outer glass layer |
| 2 | inner glass layer |
| *a* | air |
| *c* | cooling or constant |
| *h* | heating |
| *in* | indoor or inner surface |
| *max* | maximum |
| *min* | minimum |
| *out* | outdoor or outer surface |
| *p* | plate |
| *r* | thermal radiation |
| *L*1 | inner surface of external plate |
| *L*2 | inner surface of internal plate |
| *sum* | summer |
| *win* | window or winter |
| *T* | two-plate model |
| *O* | original model |

*Greek letters*

| | |
|---|---|
| $\varepsilon$ | emissivity |
| $\rho$ | density ($kg\,m^{-3}$) |
| $\sigma$ | Stefan-Boltzmann constant ($W\,m^{-2}\,K^{-4}$) |
| $\tau$ | time (s) |

tion and/or solar radiation efficiently to maintain the indoor air temperature in the thermal comfort range with no additional space heating or cooling; (2) the best natural ventilation strategy; (3) the minimal additional energy consumption for space heating in winter or air-conditioning in summer.

In order to overcome the shortcomings or limitations, some new concepts and a different approach should be adapted. That is an inverse problem to the traditional one should be solved, which is shown in Fig. 1(b).

The objectives of this paper are: (1) to put forward some new concepts for developing energy efficient buildings; (2) to find an approach for developing energy efficient buildings; (3) to demonstrate the application of the new concept and approach by using an illustrative example; and (4) to present some limitations that will need further research.

## 2. Physical problem

For a given building located in a given region, if the indoor thermal disturbances from people, lights and equipment are known, the natural room temperature of the building (i.e., the indoor air temperature of the room without any active cooling or heating) is the function of the wall's thermophysical properties [5],

$$t_{in} = f_1(k(t), \rho c_p(t), ACH(\tau)) \tag{1}$$

Fig. 2 shows the schematic diagram of a natural room temperature and the integrated degree of discomfort in winter and in summer. The region below the natural room temperature and above the upper limit of the comfortable region $t_H$ is defined as the integrated degree of discomfort in summer, $I_{sum}$, while the region below the lower limit of the comfortable region $t_L$ and above the natural room temperature is defined as the integrated degree of discomfort in winter, $I_{win}$. Their defining equations are as follows [5],

$$I_{win} = \int_{year} (t_L - t_{in})\,d\tau \quad \text{when} \quad t_{in} < t_L \tag{2}$$

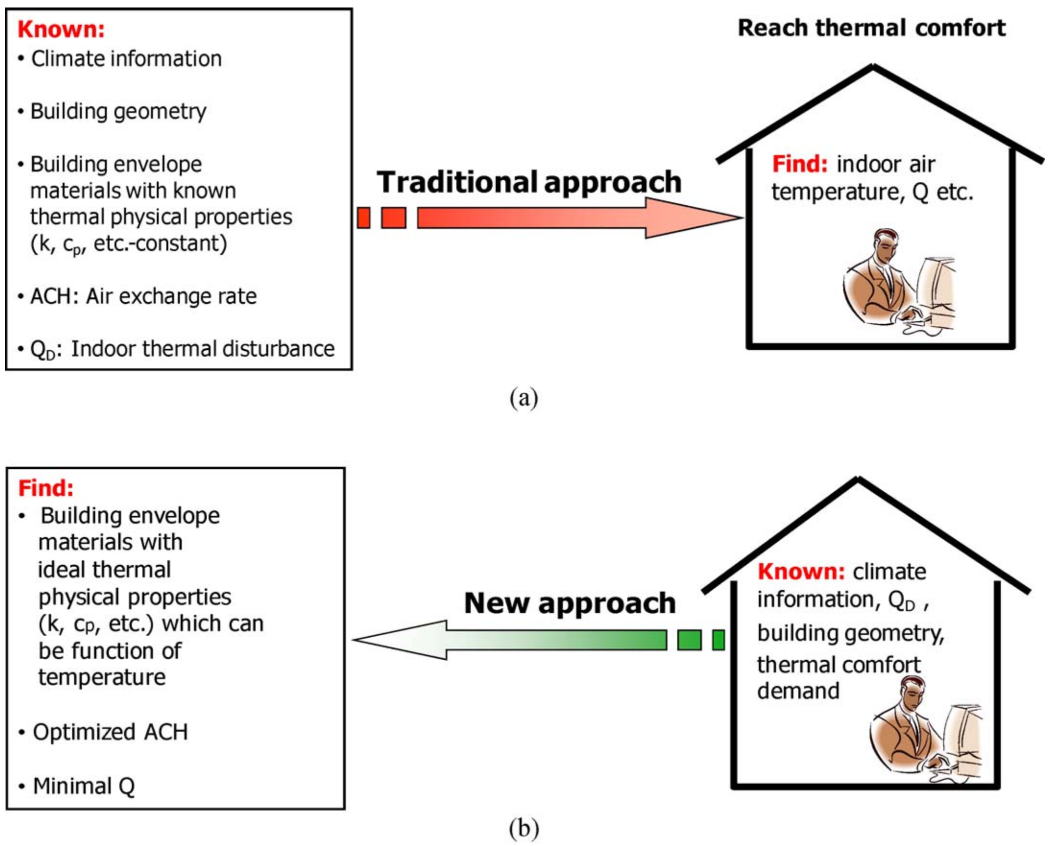



**Fig. 1.** Comparison of the traditional approach and the new approach for space heating and/or cooling: (a) traditional approach and (b) new approach.

$$I_{sum} = \int_{year} (t_{in} - t_H)\, d\tau \quad \text{when} \quad t_{in} > t_H \tag{3}$$

It is easily understood that $I_{win}$ is proportional to the space heating load while $I_{sum}$ is proportional to the space cooling load.

If we put in the additional heating rate or cooling rate, $Q(\tau)$, into the space, the room temperature can be expressed by the following equation,

$$t_{in} = f_2(k(t), \rho c_p(t), ACH(\tau), Q(\tau)) \tag{4}$$

Combining Eqs. (2)–(4), we have,

$$I_{win} = f_3(k, \rho c_p, ACH, Q) \tag{5}$$

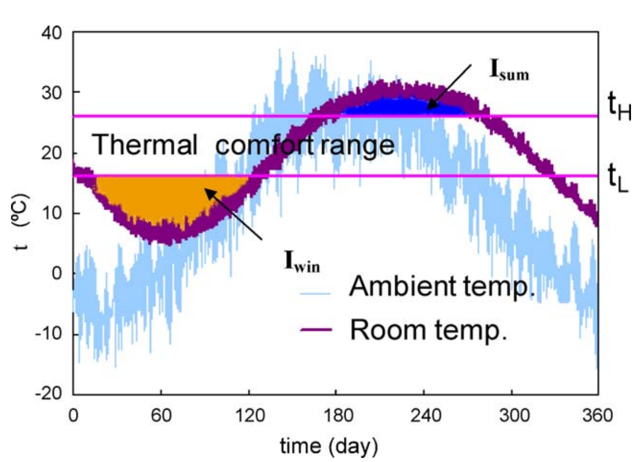
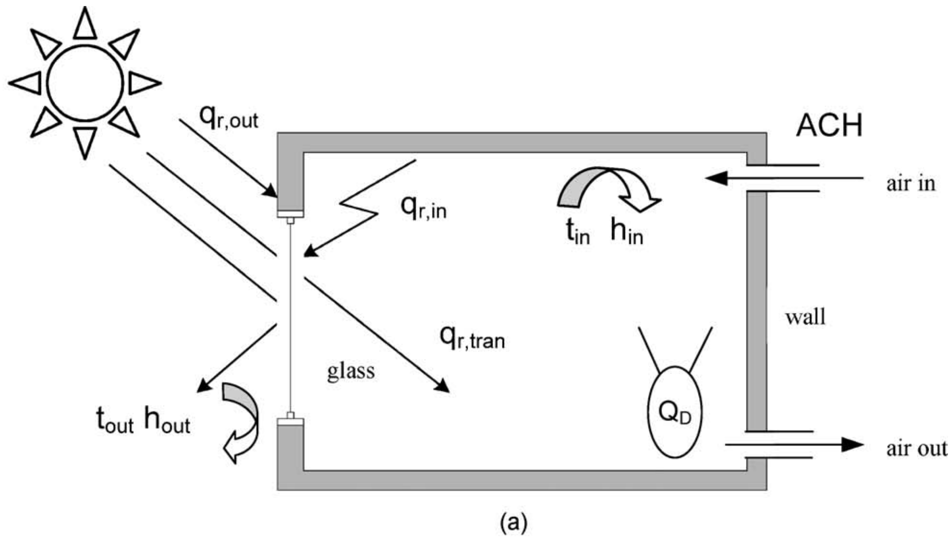



**Fig. 2.** Schematic of room temperature and integrated degree of discomfort in winter and in summer.

$$I_{sum} = f_4(k, \rho c_p, ACH, Q) \tag{6}$$

The traditional approaches to attain thermal comfort are to adjust the four parameters to make $I_{win}$ and $I_{sum}$ be as low as possible (or even to approach 0). For the new approach presented in this paper, the process is just opposite: the ideal value or function of one of the four parameters (the other three parameters should be known) can be obtained by minimizing the integrated degree of discomfort $I$ value. There are two kinds of buildings. In the case of buildings without any additional space heating or cooling systems, $Q = 0$. It is frequently not possible to make the value of $I$ be 0. In this case, the ideal parameters of $k$, $\rho c_p$, $ACH$ are the values or functions that can minimize $I$. The second kind of building is that where additional space heating or cooling is available, and in this case the ideal parameters for $k$, $\rho c_p$, $ACH$, $Q$ are those values or functions which can make $I_{win}$ and/or $I_{sum}$ be zero. In other words, the ideal value or function of one of the four parameters can be obtained by solving the inverse problem.

It is noted that: (1) in contrast to the traditional approach for analyzing the heating or cooling load of a given building where the thermophysical properties are known and are constant so that the related equations describing the room temperature tend to be linear differential equations, in our approach the ideal $k(t)$ or $\rho c_p(t)$ or $Q(\tau)$ or $ACH(\tau)$ is an unknown function, and the related equations for determining them are non-linear differential equations; (2) a series of new concepts are put forward, i.e., ideal $k(t)$, $\rho c_p(t)$, $ACH(\tau)$ for passive buildings (without any space heating or cooling) and ideal $k(t)$, $\rho c_p(t)$, $ACH(\tau)$ and $Q(\tau)$ for active buildings (with space heating or cooling). The buildings corresponding to the ideal parameters are called ideal passive energy efficient buildings and ideal active energy efficient buildings, respectively. After getting such ideal $ACH(\tau)$, $k(t)$, $\rho c_p(t)$ for passive buildings and ideal $ACH(\tau)$, $k(t)$, $\rho c_p(t)$ and $Q(\tau)$ for active buildings, some guidance can



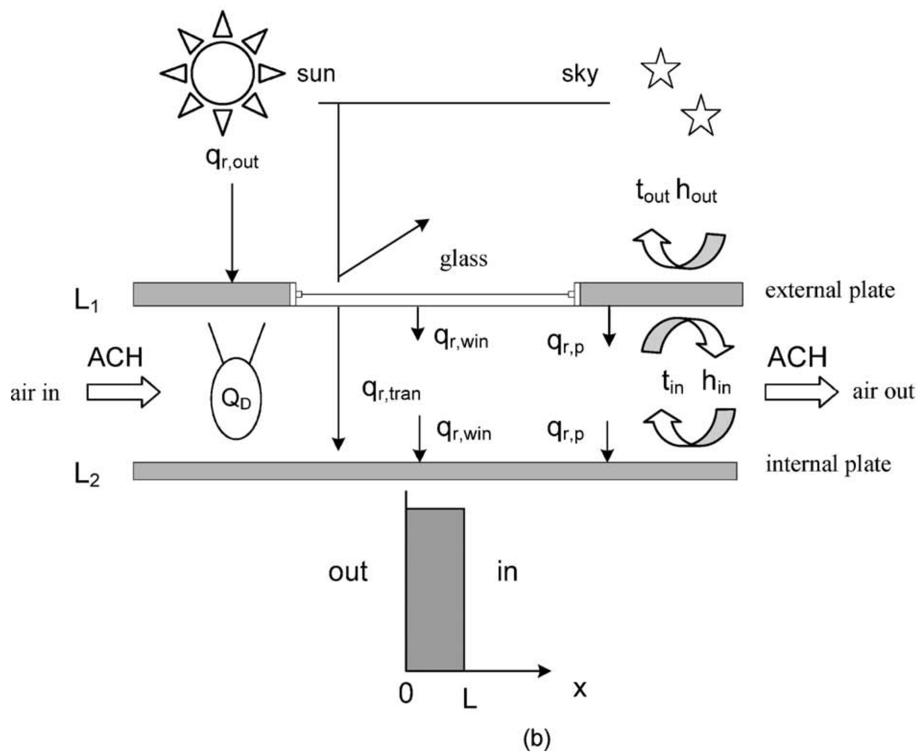




**Fig. 3.** Schematic of (a) a room, and (b) two-plate room model.

be gotten from them: (1) researchers in building material science can know the best thermophysical properties for building envelope materials; (2) clients can know how to control natural ventilation; (3) the HVAC engineers can know how to most efficiently operate space heating or cooling systems. As an initial step, the approach for determining an ideal $\rho c_p(t)$ function for a passive room is studied in this paper.

## 3. Mathematical problem

In order to find the non-linear ideal $\rho c_p(t)$ of a typical passive room in a multi-story building, a room with a south external wall, a south-facing double-glazed window and three inner walls (Fig. 3(a)) is considered. To simplify the analysis, the following assumptions are made: (1) due to symmetry, the internal thermal mass components such as internal wall, ceiling and floor are taken as adiabatic at the corresponding center-line; (2) the long wave radiations between the internal thermal mass components are ignored; (3) the convective heat transfer coefficients between the indoor air and all the internal thermal mass are assumed to be the same. According to the assumptions, the internal thermal mass components can be combined into one plate, which is designated as an internal plate. Meanwhile, the building's external thermal mass such as the building's external wall is treated as an external plate, in which the south-facing window is fixed (Fig. 3(b)). Therefore, a simplified two-plate model can be used to analyze the thermal performance of a room in order to solve the inverse problem quickly and easily. In addition, the following assumptions are also made: (1) all heat disturbances in a room can be combined in one heat source term, $Q_D$; (2) the air in the room is fully mixed.

**Table 1**
Comparison of the integrated degrees of discomfort between the two-plate model and the original model of the comparative case.

| | $I_{win}$ (°C h) | $I_{sum}$ (°C h) |
|---|---|---|
| Two-plate model | 1072.3 | 5550.9 |
| Original model | 1194.2 | 5205.8 |
| Relativity deviation (%) | −10.2 | 6.6 |

The transient heat transfer equation of the plate is:

$$\rho_p c_{p,p} \frac{\partial t_p}{\partial \tau} = k_p \frac{\partial^2 t_p}{\partial x^2} \tag{7}$$

The boundary conditions for Eq. (7) are:

$$h_{out}(t_{out} - t_{p,out}) + q_{r,out} = -k_p \left.\frac{\partial t_p}{\partial x}\right|_{x=0} \tag{8}$$

$$h_{in}(t_{in} - t_{p,in}) + q_{r,in} = k_p \left.\frac{\partial t_p}{\partial x}\right|_{x=L} \tag{9}$$

The initial condition for Eq. (7) is:

$$t_p(x, \tau)|_{\tau=0} = t_{init} \tag{10}$$

where $h_{in}$ and $h_{out}$ are the convective heat transfer coefficients of the inner and outer plate surfaces, respectively, which are calculated by the correlations according to Ref. [6].

The heat transfer equations of the south-facing double-glazed window are:

$$\rho_1 c_{p,1} \Delta x_1 \frac{\partial t_1}{\partial \tau} = h_{out}(t_{out} - t_1) + h_{1,2}(t_2 - t_1) + q_1 \tag{11}$$

$$\rho_2 c_{p,2} \Delta x_2 \frac{\partial t_2}{\partial \tau} = h_{in}(t_{in} - t_2) + h_{1,2}(t_1 - t_2) + q_2 \tag{12}$$

where $h_{1,2}$ is the overall heat transfer coefficient between the two glass layers. $h_{1,2}$ can be calculated by the following equation:

$$h_{1,2} = \frac{1}{(1/U_{win}) - (1/h^*_{out}) - (1/h^*_{in})} \tag{13}$$

The radiation heat transfer rate between two plates and the window are:

$$Q_{r,p} = \varepsilon_{L1} A_{L1} \sigma[(t_{L1} + 273)^4 - (t_{L2} + 273)^4] \tag{14}$$

$$Q_{r,win} = \varepsilon_{win} A_{win} \sigma[(t_2 + 273)^4 - (t_{L2} + 273)^4] \tag{15}$$

The energy balance equation of indoor air is:

$$V_R \cdot \rho_a \cdot c_{p,a} \frac{\partial t_a}{\partial \tau} = \sum_{j=1}^{2} Q_{p,j} + Q_{win} + Q_{D,C} + Q_L \tag{16}$$

where $Q_{p,j}$, $Q_{win}$ and $Q_L$ can be calculated by the following equations:

$$Q_{p,j} = h_{in} \times (t_{p,in,j} - t_{in}) \times A_{p,j} \tag{17}$$

$$Q_{win} = h_{in} \times (t_2 - t_{in}) \times A_{win} \tag{18}$$

$$Q_L = V_R \cdot \rho_a \cdot c_{p,a} \times ACH \times \frac{(t_{out} - t_{in})}{3600} \tag{19}$$

To validate the two-plate model, the calculated results and those directly calculated from the heat transfer model without the simplification (original model) are compared. The original model is the same as the model in Ref. [5] and has been validated with experimental results. The integrated degree of discomfort in winter $I_{win}$ and in summer $I_{sum}$ calculated by the two models is compared in Table 1. It is assumed that ACH is 0.5 $h^{-1}$ when the window is closed and the indoor heat disturbance is 10.8 W $m^{-2}$ for the comparative case. The other conditions are the same as the studied cases. The relative error is less than 11% in this case. It is seen that the relative error caused by the two-plate model is acceptable.

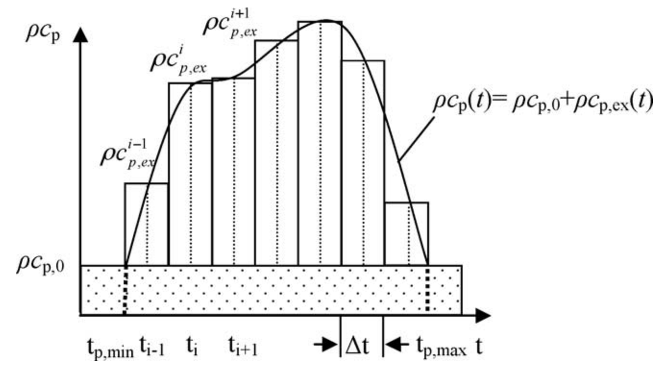



**Fig. 4.** Schematic of the N-segment method of $\rho c_p(t)$.

## 4. Solution

### 4.1. Optimization target

As mentioned earlier, the objective of the inverse analysis is to get the ideal volumetric specific heat $\rho c^*_p(t)$, by minimizing the $I_{win}$ and $I_{sum}$ values in a passive room. Because the specific heat could not be zero in practice, it is assumed that the $\rho c_p$ of all building material has a basic value $\rho c_{p,0}$. The ideal volumetric specific heat $\rho c^*_p(t)$ includes the basic volumetric specific heat $\rho c_{p,0}$ and the excessive volumetric specific heat $\rho c_{p,ex}(t)$ (see Fig. 4). In practice, building material density $\rho$ dose not change a lot in the normal temperature range, so $\rho$ is assumed to be constant in this paper. Only the specific heat $c_p$ is considered as a function of temperature.

### 4.2. N-segment method

We put forward the N-segment method to obtain the ideal excessive volumetric specific heat $\rho c_{p,ex}(t)$. First, $\rho c_{p,ex}(t)$ can be approximated by $\rho c^i_{p,ex}$, $i = 1, \ldots, N$, where the $\rho c^i_{p,ex}$ is the excessive volumetric specific heat in each temperature segment, so that $\rho c_p(t) = \rho c_{p,0} + \rho c_{p,ex}(t)$ can be approximated by $\rho c_{p,0} + \rho c^i_{p,ex}$, $i = 1, \ldots, N$ (see Fig. 4). The procedure then involves the following steps: (1) solve the direct problem using the simplified two-plate room model and obtain the effective temperature variation range for a year of the thermal mass with the basic volumetric specific heat $\rho c_{p,0}$ (the effective temperature variation range was set to be the initial optimized temperature range); (2) evenly divide the optimized temperature range into N temperature segments and distribute the excessive volumetric specific heat, $\rho c^i_{p,ex}$ $i = 1, \ldots, N$, in the N temperature segments; (3) adjust the $\rho c^i_{p,ex}$ $i = 1, \ldots, N$ values in different temperature segments to minimize the thermal degree of discomfort I using a non-linear constrained optimization method such as the Sequential Quadratic Programming method (SQP) [7–9]; (4) finally, adjust the optimized temperature range and repeat steps (2) and (3) until the I value reaches a minimum.

There are some limiting conditions for $\rho c_{p,ex}(t)$,

$$\int_{T_1}^{T_2} \rho c_{p,ex}\, dt = \rho H_{ex} = constant \tag{20}$$

$$\rho c_{p,ex}(t) \geq 0 \tag{21}$$

where $\rho H_{ex}$ is the excessive volumetric enthalpy over the whole temperature range. In practice $\rho H_{ex}$ could be adjusted by changing the proportion of the high specific heat material in the thermal mass. $\rho c_{p,ex}(t)$ can be composed by $\rho c^i_{p,ex}$, $i = 1, \ldots, N$, so the limiting

**Table 2**
Thermophysical properties of the room envelopes.

| Material | $\rho c_{p,c}$ (MJ m$^{-3}$ °C$^{-1}$) | $k$ (W m$^{-1}$ °C$^{-1}$) | $\rho H_{ex}$ (MJ m$^{-3}$) |
|---|---|---|---|
| Reinforced concrete | 2.3 | 0.81 | – |
| Polystyrene board | 0.048 | 0.047 | – |
| Glass | 2.1 | – | – |
| Concrete hollow block | 0.84 | 0.41 | – |
| Internal thermal mass | 1.5 | 1 | 40–300 |

conditions for $\rho c_{p,ex}(t)$ can be rewritten as

$$\sum_{i=1}^{N} \rho c_{p,ex}^{i} \Delta t = \rho H_{ex} = \text{constant} \quad (22)$$

$$\rho c_{p,ex}^{i} \geq 0, \quad i = 1, \ldots, N \quad (23)$$

The governing differential equations for the problem are Eqs. (2), (3), (7)–(23). The optimization of $\rho c_{p,ex}^{i}$ ($i$ = 1, 2, ..., $N$) can be rewritten using a matrix as follows:

$$\text{Min}\, I_{win}(\rho c_{p,ex}^{1}, \ldots, \rho c_{p,ex}^{N}) \ \text{or} \ I_{sum}(\rho c_{p,ex}^{1}, \ldots, \rho c_{p,ex}^{N})$$

$$\text{subject to} \begin{cases} \begin{pmatrix} -1 & & \\ & \ddots & \\ 0 & & -1 \end{pmatrix}_{N\times N} \cdot \begin{pmatrix} \rho c_{p,ex}^{1} \\ \vdots \\ \rho c_{p,ex}^{N} \end{pmatrix}_{N\times 1} \leq \begin{pmatrix} 0 \\ \vdots \\ 0 \end{pmatrix}_{N\times 1} \\ \begin{pmatrix} \Delta t & \ldots & \Delta t \end{pmatrix}_{1\times N} \cdot \begin{pmatrix} \rho c_{p,ex}^{1} \\ \vdots \\ \rho c_{p,ex}^{N} \end{pmatrix}_{N\times 1} = \rho H_{ex} \end{cases} \quad (24)$$

This non-linear optimization problem can be solved by using the non-linear optimization method SQP, from where the optimized $\rho c_{p,ex}^{1}, \ldots, \rho c_{p,ex}^{N}$ in the given temperature segments are obtained. Because the temperature segments division may influences the optimized results, it should be adjusted until the $I$ value is minimal.

## 5. Case study

In order to find out the typical form of the ideal volumetric specific heat of the internal thermal mass, an ordinary passive room in Beijing is analyzed in this section. Furthermore, from the mathematical model it can be seen that the building's thermal performance is closely related to climatic conditions. To study the ideal volumetric specific heat of the internal thermal mass in different climatic regions is very useful for practical applications. Seven representative cities in different climatic regions of the Chinese mainland are chosen for the target room to be investigated in this paper.

### 5.1. The calculated conditions

An ordinary passive room in a multi-storey building is analyzed (see Fig. 5). The dimensions of the simulated room are 5.7 m (depth) × 3.6 m (width) × 3.2 m (height). It has an external south-facing wall, which includes 180-mm-thick reinforced concrete and 70-mm-thick polystyrene board. A 1.7 m × 2.0 m double-glazed window is fixed in the exterior south wall. The overall heat transfer coefficient of the double-glazed window is 3.1 W (m$^2$ °C)$^{-1}$. The shading coefficient (SC) value of the window is 0.74 in winter and it is 0.44 with a curtain in summer. The thicknesses of three concrete hollow block internal walls, reinforced concrete floor and ceiling are all 200 mm. The thermal properties of the building envelope components are listed in Table 2. The thermophysical properties of the external thermal mass remain unchanged while the internal thermal mass is optimized.

The ACH is assumed to be 0.75 h$^{-1}$ when the window is closed. The ACH becomes 5.0 when the window is opened to make full use

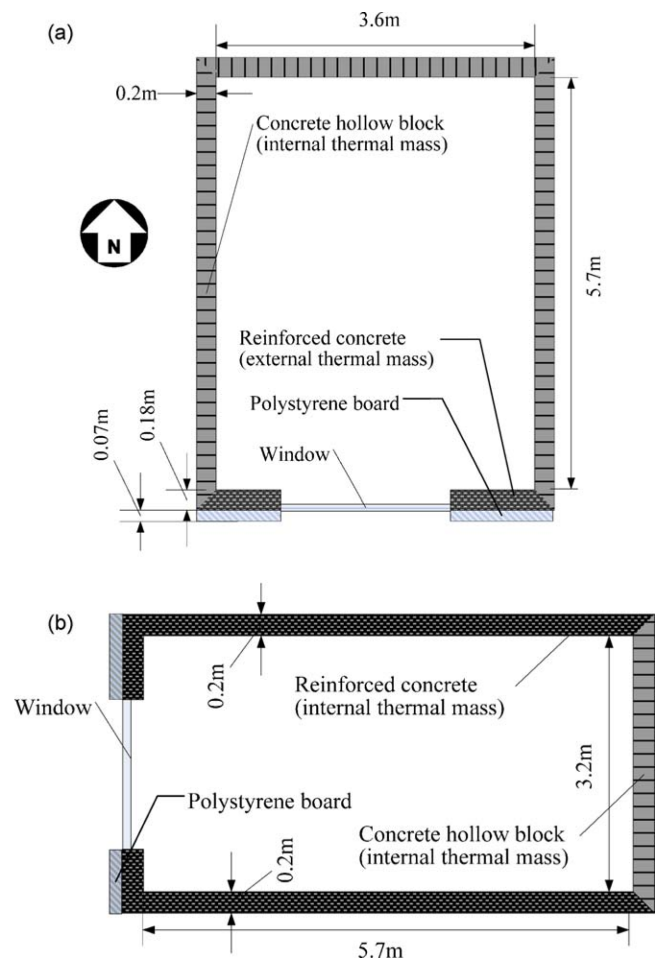



**Fig. 5.** Schematic of the simulated room: (a) plan view and (b) cross-section view.

of the summer night ventilation when the outdoor temperature is less than 26 °C and greater than 20 °C. The indoor heat disturbance from people, lights and equipment is 3.8 W m$^{-2}$. In this paper, $t_L$ and $t_H$ are set to be 16 and 28 °C, respectively [10,11].

The climate data for simulation are generated by Chinese Architecture-specific Meteorological Data Sets for Thermal Environment Analysis. The indoor air temperature of the room is calculated using the aforementioned two-plate room model. The ideal volumetric specific heat function can be obtained using the $N$-segment method and the non-linear optimization methods.

### 5.2. Results and discussion

#### 5.2.1. Ideal specific heat of the internal thermal mass in Beijing

The ideal specific heat of the internal thermal mass in summer is calculated by minimizing the thermal degree of discomfort in winter $I_{sum}$. In order to illustrate the effect of the non-linear optimization, the constant $\rho c_{p,c}$ is introduced, which satisfies: $\rho c_{p,c} = \rho c_{p,0} + \rho H_{ex}/\Delta T$, where, $\Delta T = t_{p,max} - t_{p,min}$ is the effective temperature variation range over a year for the thermal mass. This means that the excessive volumetric enthalpy is evenly distributed in the effective temperature variation range of the thermal mass by applying $\rho c_{p,c}$. $\rho c_{p,c}$ and $\rho c_p^*(t)$ both consist of the same basic volumetric specific heat $\rho c_{p,0}$ and excessive volumetric enthalpy $\rho H_{ex}$. Since the only difference between $\rho c_{p,c}$ and $\rho c_p^*(t)$ is their excessive volumetric specific heat form, it is reasonable to illustrate the effect of the non-linear optimization by comparing the $I$ values of the rooms having the two different internal thermal masses (with $\rho c_p^*(t)$ and with $\rho c_{p,c}$).

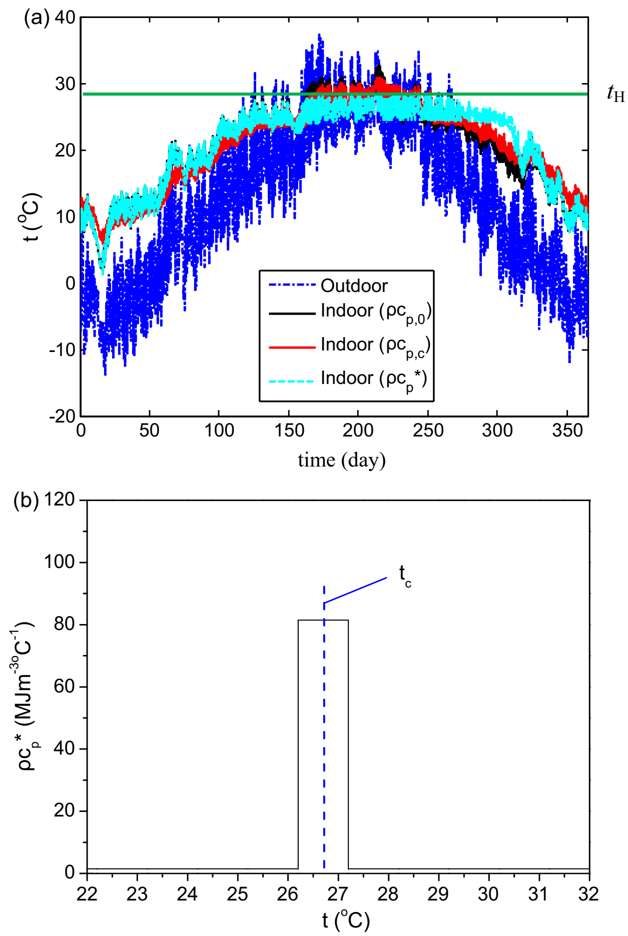



**Fig. 6.** Optimization results for the internal thermal mass in summer in Beijing. ($\rho H_{ex}$ = 80 MJ m$^{-3}$) (a) $t_{in}$ and (b) $\rho c_p^*(t)$.

Fig. 6 shows the optimization results for the internal thermal mass in summer with excessive volumetric enthalpy $\rho H_{ex}$ = 80 MJ m$^{-3}$. It is seen from Fig. 6(a) that the highest indoor air temperature of the internal thermal mass with the $\rho c_{p,0}$ is about 33 °C in summer. This can be reduced by 2 °C by applying $\rho c_{p,c}$ to the internal thermal mass. After optimization the highest indoor air temperature with $\rho c_p^*(t)$ is reduced to 28 °C, which is 3 °C lower than that of the internal thermal mass with $\rho c_{p,c}$. That is to say, this kind of internal thermal mass can meet the thermal demand in summer without additional space cooling. It is also seen from Fig. 6(a) that the time lag of the indoor air temperature increases after adding the excessive volumetric enthalpy. From Fig. 6(b) it is found that the form of the ideal volumetric specific heat $\rho c_p^*(t)$ of the internal thermal mass approaches the $\delta$ function. The central point of the temperature segment in which the excessive volumetric enthalpy is concentrated is defined as the "characteristic temperature" $t_c$. From an energy saving perspective, the implication is that the ideal building envelope material has the thermal mass characteristics of phase change material. The characteristic temperature corresponds to the melting point temperature, and the excessive enthalpy corresponds to the heat of fusion of the phase change material.

The physical mechanism can be explained as follows: The internal thermal mass supplies heat to the room when its surface temperature is higher than the indoor air temperature. Conversely, the internal thermal mass absorbs heat from the room. The heat transfer rate between the surface of internal thermal mass and the indoor air depends on the corresponding temperature difference when the heat transfer area $A_{p,j}$ and heat transfer coefficient $h_{in}$ are unchanged. If the excessive volumetric enthalpy is concentrated in $t_c$, the surface temperature of the internal thermal mass could be kept around $t_c$ when the excessive volumetric enthalpy is used. In that case, most of the stored cold of the internal thermal mass would be used when the indoor air temperature is higher than $t_c$. The heat transfer rate depends on the temperature difference between the indoor air temperature and $t_c$. On the other hand, the total cold supplied by the internal thermal mass depends on the thermal storage capacity (i.e., the amount of cold which can be stored in the internal thermal mass). When the excessive volumetric enthalpy is given, the excessive cold storage capacity is constant. A suitable $t_c$ should be chosen so that the most excessive cold storage capacity can be used to reduce the highest indoor temperature rather than be wasted when the indoor temperature is too low. In order to supply cold to the room, the $t_c$ should be lower than the highest indoor air temperature, the temperature difference depends on the cooling load, cold storage capacity and thermal conductivity of the internal thermal mass as well as the thermal resistance between the internal thermal mass and the indoor air. The more excessive volumetric enthalpy is given, the more cold storage capacity can be used to reduce the highest indoor temperatures. In this way, the thermal degree of discomfort in summer $I_{sum}$ decreases.

It is also indicated in Fig. 6(a) that after optimization, the thermal degree of discomfort in summer $I_{sum}$ is zero when the excessive volumetric enthalpy $\rho H_{ex}$ is over 80 MJ m$^{-3}$. That is to say, 80 MJ m$^{-3}$ is the critical value of $\rho H_{ex}$ for ideal thermal mass in a passive-cooling building. In selecting or developing building envelope material for practical purposes, it is not necessary for $I_{sum}$ to be zero. Thus, $\rho H_{ex}$ can be much lower than 80 MJ m$^{-3}$. The desired $\rho H_{ex}$ depends on the expected integrated degree of discomfort. Therefore, the method provides an approach to determine the desired $\rho H_{ex}$ according to the expected integrated degree of discomfort.

The ideal specific heat of thermal mass in winter is also calculated by minimizing thermal degree of discomfort in winter $I_{win}$. It was found that the critical value of $\rho H_{ex}$ for an ideal thermal mass for a passive-heating building in Beijing is 220 MJ m$^{-3}$ for the case studied. Its corresponding characteristic temperature $t_c$ was found to be 19.3 °C. After optimization the lowest indoor air temperature is increased by 5 °C.

### 5.2.2. *Ideal volumetric specific heat of the internal thermal mass in different climatic regions*

In the previous section, the ideal heat specific functions of the internal thermal mass were obtained using the *N*-segment method and the non-linear optimization methods for a target room in Beijing. In this section, target rooms in seven representative cities in different climatic regions of the Chinese mainland are selected to research the ideal volumetric specific heat of the internal thermal mass. Fig. 7 shows the location of the seven studied cities in China and Table 3 lists the climatic characteristics of them.

When the basic volumetric specific heat $\rho c_{p,0}$ is given, the ideal volumetric specific heat $\rho c_p^*(t)$ of the thermal mass can be expressed by $\rho H_{ex}$ and $t_c$ since it approaches the $\delta$ function. Table 4 lists the critical values of $\rho H_{ex}$ and their corresponding $t_c$ of the internal thermal mass for passive-cooling and passive-heating in buildings in the seven cities.

In Harbin, where there are large heating loads in winter and no cooling loads in summer, it is not possible to rely on thermal storage to meet the comfort demand in winter. In Urumchi, with large heating loads in winter and small cooling loads in summer, it is possible to rely on the thermal storage to meet the comfort demand in summer. In Beijing and Shanghai comfort demand

**Table 3**
Climatic characteristics of the studied cities.

| Regions | Latitude (°) | January | | July | | Climatic type |
|---|---|---|---|---|---|---|
| | | Average temp. (°C) | Average solar radiation ($W\,m^{-2}$) | Average temp. (°C) | Average solar radiation ($W\,m^{-2}$) | |
| Harbin | 45.75 | −21.8 | 67.4 | 22.9 | 209.1 | Severe cold |
| Urumchi | 43.78 | −18.2 | 56.8 | 25.0 | 261.6 | Severe cold |
| Beijing | 39.93 | −3.6 | 97.8 | 25.3 | 205.0 | Cold |
| Shanghai | 31.17 | 3.7 | 86.7 | 29.2 | 188.9 | Hot summer and cold winter |
| Lhasa | 29.67 | −0.2 | 189.8 | 15.4 | 270.3 | Cold |
| Kunming | 25.02 | 7.5 | 168.1 | 19.5 | 180.0 | Moderate |
| Guangzhou | 23.13 | 9.4 | 103.1 | 27.8 | 166.5 | Hot summer and warm winter |

**Table 4**
The critical values of $\rho H_{ex}$ and their corresponding $t_c$ of internal thermal mass for free-cooling and free-heating in building for the studied cities.

| Regions | | Winter | | Summer | |
|---|---|---|---|---|---|
| | | $t_c$ (°C) | $\rho H_{add}$ ($MJ\,m^{-3}$) | $t_c$ (°C) | $\rho H_{add}$ ($MJ\,m^{-3}$) |
| Harbin | ▲ | – | – | – | 0 |
| Urumchi | ▲ | – | – | 26.7 | 60 |
| Beijing | ● | 19.3 | 220 | 26.7 | 80 |
| Shanghai | ● | 18.3 | 80 | 26.6 | 130 |
| Lhasa | ● | 18.9 | 40 | – | 0 |
| Kunming | ● | – | 0 | – | 0 |
| Guangzhou | ● | – | 0 | 26.5 | 300 |

*Note*: '●' thermal comfort can be met in a whole year; '▲' thermal comfort can be met in summer; '–' not exist/any.

can be met for the entire year by using thermal storage, however, the critical value of $\rho H_{ex}$ is very large. In Lhasa, the critical value of $\rho H_{ex}$ is small in winter, and there is no need for cooling in summer. In Kunming, there is no need for adding $\rho H_{ex}$ to meet comfort demand in a whole year. In Guangzhou, the large critical value of $\rho H_{ex}$ is needed in summer; and there is no need for heating in winter. From Table 4, it is seen that the critical values of excessive volumetric enthalpy $\rho H_{ex}$ are different under different climates, but the corresponding characteristic temperatures $t_c$ of the ideal thermal mass are very similar. They fall in the temperature ranges 18.3–19.3 °C in winter and 26.5–26.7 °C in summer. This implies that the characteristic temperature $t_c$ should be set near the thermal comfort limit, so that the most excessive volumetric enthalpy can be used to enhance thermal comfort, rather than be wasted when thermal comfort can be achieved without using excessive enthalpy. The small differences between $t_c$ in the different cities are caused by the different maximum heating/cooling loads which would be carried by the internal thermal mass.

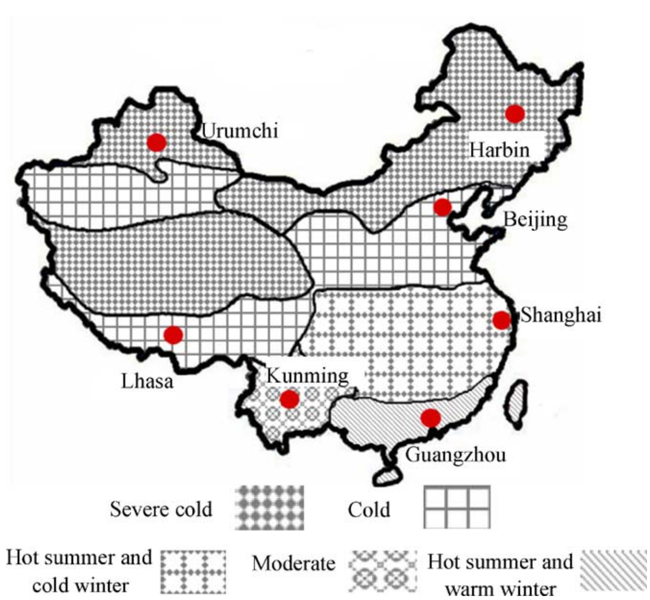



**Fig. 7.** Location of the seven studied cities in different climatic regions in China.

The thermal degree of discomfort in winter $I_{win}$ values and the lowest temperatures of the rooms having three different internal thermal masses (with basic volumetric specific heat $\rho c_{p,0}$, with ideal volumetric specific heat $\rho c_p^*(t)$ and with constant volumetric specific heat $\rho c_{p,c}$) in different climatic regains are compared in Fig. 8. The critical values of $\rho H_{ex}$ in winter are added to the basic specific heat $\rho c_{p,0}$ both for $\rho c_p^*(t)$ and $\rho c_{p,c}$ in these cases. It can be seen that the thermal degree of discomfort in winter $I_{win}$ of a room having an internal thermal mass of $\rho c_{p,c}$ is lower than that with $\rho c_{p,0}$. After optimization, $I_{win}$ can be reduced to zero. By applying $\rho c_{p,c}$, the lowest temperature of the room can be increased. After optimization, the lowest temperature of the room can be increased to 16 °C. The effect of optimization is most remarkable in Beijing.

Fig. 9 shows the thermal degree of discomfort in summer $I_{sum}$ values and the highest temperatures of the rooms having three different internal thermal masses (with basic volumetric specific heat $\rho c_{p,0}$, with ideal volumetric specific heat $\rho c_p^*(t)$ and with constant volumetric specific heat $\rho c_{p,c}$) in different cities. The critical values of $\rho H_{ex}$ in summer are also added to the basic specific heat $\rho c_{p,0}$ both for $\rho c_p^*(t)$ and $\rho c_{p,c}$ in these cases. Here it can be seen that the thermal degree of discomfort in summer $I_{sum}$ of a room with an internal thermal mass of $\rho c_{p,c}$ is lower than that of $\rho c_{p,0}$. After optimization, $I_{sum}$ can be reduced to zero. By applying $\rho c_{p,c}$, the highest temperature of the room can be reduced, and after optimization this can be reduced to 28 °C. The effect of optimization is most remarkable in Shanghai. The $I_{sum}$ values and the highest temperatures of the rooms having an internal thermal mass of $\rho c_p^*(t)$ and $\rho c_{p,c}$ are the same. This is because the high $\rho H_{ex}$ causes the $\rho c_p^*(t)$ and $\rho c_{p,c}$ to be the same over the narrow effective temperature range.

The ideal specific heat of the thermal mass for an entire year is calculated by minimizing the thermal degree of discomfort in a year $I = I_{sum} + I_{win}$. Fig. 10 shows the optimization results for the internal thermal mass for the entire year in Shanghai. It is found that the ideal $\rho c_p^*(t)$ of the thermal mass for an entire year is close to the superposition of the $\rho c_p^*(t)$ of the thermal mass in winter and that in summer. However, the $I$ value does not significantly decrease after

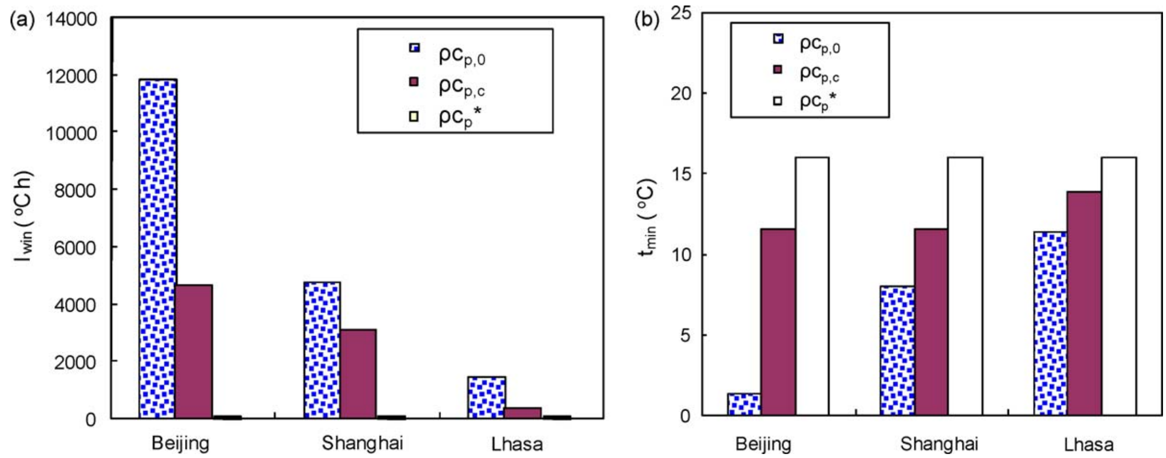



**Fig. 8.** Optimization effects for the internal thermal mass in winter in different cities: (a) $I_{win}$ and (b) $t_{min}$.

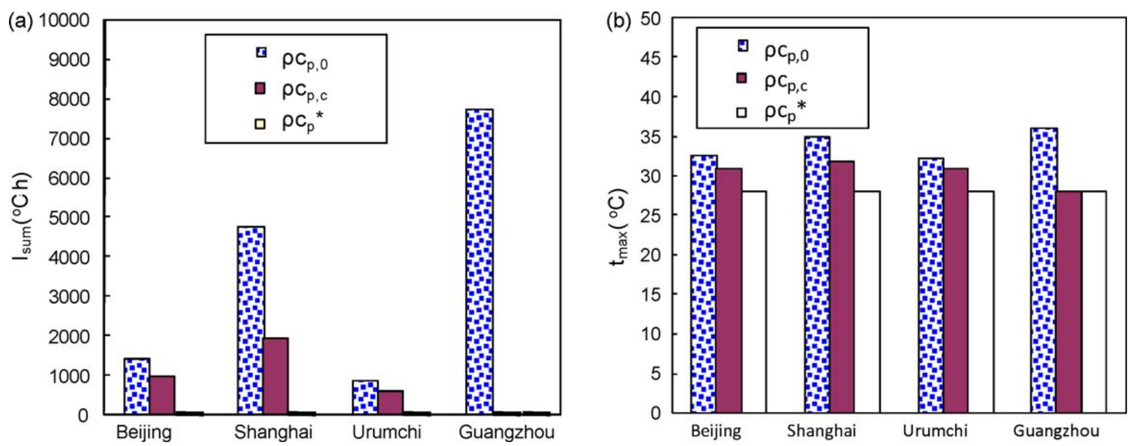



**Fig. 9.** Optimization effects for the internal thermal mass in summer in different cities: (a) $I_{sum}$ and (b) $t_{max}$.

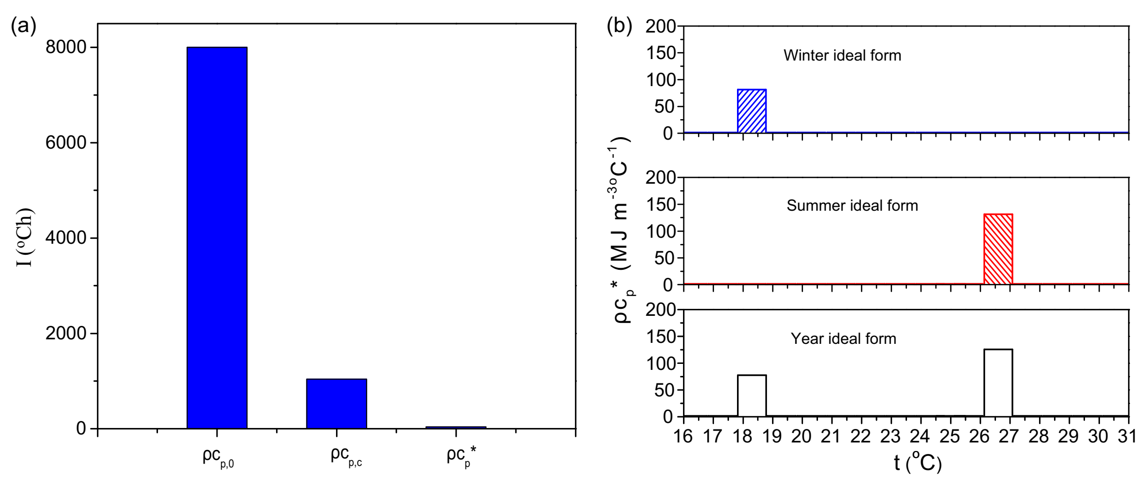



**Fig. 10.** Optimization results for the internal thermal mass for an entire year in Shanghai: (a) $I$ and (b) $\rho c_p^*(t)$.

optimization. It is indicated that a high $\rho H_{ex}$ causes the excessive volumetric specific heat form in the effective temperature range can not significantly influence the *I* value.

## 6. Conclusions

In this paper, some new concepts and an approach for developing energy efficient buildings are put forward. As an initial step, the ideal specific heat of a building wall is determined. The results show that:

(1) For the cases studied, the ideal specific heat of a building wall is composed of a basic value and an ideal excessive value. The ideal form of excessive specific heat approaches the $\delta$ function.
(2) The critical values of excessive volumetric enthalpy $\rho H_{ex}$ are different at different climates, but the corresponding charac-

teristic temperatures $t_c$ of the ideal thermal mass are close to each other in the Chinese mainland. They fall in the temperature ranges of about 18.3–19.3 °C in winter and about 26.5–26.7 °C in summer.

(3) The ideal excessive volumetric specific heat $\rho c_p^*(t)$ of the thermal mass over an entire year is close to the superposition of the $\rho c_p^*(t)$ of the thermal mass in winter and that in summer.

This paper presents a completely novel angle for approaching the development of energy efficient buildings. However, there are many other factors to be considered for optimization such as the minimal additional space heating or cooling rate, optimized air change rate, etc. when applying the new concepts and approach. In the process of studying this further, the approach itself will also be improved.

## Acknowledgements

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## References

[1] 2009 Annual Report on China Building Energy Efficiency, Tsinghua University Buildings Energy Efficiency Research Center, China Architecture & Building Press, Beijing, 2009 (in Chinese).

[2] S.W. Wang, X.H. Xu, Parameter estimation of internal thermal mass of building dynamic models using genetic algorithm, Energy Conversion and Management 47 (13–14) (2006) 1927–1941.

[3] K. Gregory, B. Moghtaderi, H. Sugo, A. Page, Effect of thermal mass on the thermal performance of various Australian residential constructions systems, Energy and Buildings 40 (2008) 459–465.

[4] C.A. Balaras, The role of thermal mass on the cooling load of buildings. An overview of computational methods, Energy and Buildings 24 (1) (1996) 1–10.

[5] Y.P. Zhang, K.P. Lin, Q.L. Zhang, H.F. Di, Ideal thermophysical properties for free-cooling (or heating) buildings with constant thermal physical property material, Energy and Buildings 38 (10) (2006) 1164–1170.

[6] ASHRAE, ASHRAE Handbook – Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, Chapter 3: Heat Transfer 2005.

[7] P.E. Gill, W. Murray, M.H. Wright, Numerical Linear Algebra and Optimization, 1991.

[8] P.E. Gill, W. Murray, M.A. Saunders, Procedures for Optimization Problems with a Mixture of Bounds and General Linear Constraints, ACM Transactions on Mathematical Software 10 (1984) 282–298.

[9] S.P. Han, Globally convergent method for nonlinear-programming, Journal of Optimization Theory and Applications 22 (3) (1977) 297–309.

[10] HVAC Design Criterion, Standard of PR China (GBJ 19-87), Beijing, 2001 (in Chinese).

[11] Energy-Saving Design Standard for Civil Construction (Residential Building Heating Part) (JGJ 26-95), Beijing, 1995 (in Chinese).