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Optimize the Location of Embedded Pipes by Using Low-Grade Natural Thermal Energy/Waste Heat for Thermal Activated Façade

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Abstract: The final effect of a thermal activated façade (TAF) wouldn't only be influenced by the thermal character of the wall, dimensions and distance of the embedded pipes besides flow rate and temperature of the medium, but also by the location of the pipes. The location of the embedded pipes not only affects the final effect of thermoactivated, there is at the same time an obvious correspondence with the temperature of the medium. The simulations and lab investigations have concluded that the optimization the depth of embedded pipes, is necessary in order to use low-grade natural thermal energy/waste heat (LGTE) to achieve TAF. Using the bench mark of the "equivalent thermal resistance" (ER-Value), it is clear that the closer the pipes are located to the outside, the less energy grade is required. However, not all of the LGTE with higher temperature than outdoor air can be used for TAF. The limit of LGTE is the "Invalid Medium Temperature" (IVMT); although there is still a temperature difference between the medium and the outdoor environment, its thermal driving potential is not enough to form an effective heat transfer between the pipe and its surrounding per unit length, so that the medium circulation in the wall hardly contributes to changing the temperature gradient in the wall.

Keywords: Thermoactivated Façade/Envelop/Wall (TAF), Low-Grade Thermal Energy (LGTE), Equivalent Thermal Resistance (ER-Value), Invalid Medium/Water Temperature (IVMT), Ideal Medium/Water Temperature (IDMT)

1. Introduction

There is no doubt that building energy efficiency is of great importance to the national development goal of 3 0/60, and the energy consumption of buildings is mainly used to create a thermal and humid comfortable environment. The factors affecting the thermal and humid comfortable environment can be roughly divided into internal disturbance and external disturbance. Among them, internal disturbance is mainly caused by the occupation of buildings, such as personnel activities, indoor lighting, and heat generated by various types of energy-consuming equipment's, etc., while external disturbance is caused by the difference in temperature and humidity of indoor and outdoor environments. For summer conditions, especially in the south areas, internal disturbances will account for a large proportion of the air conditioning load, while in northern winter conditions, it is almost entirely necessary to consider external disturbances.

Nowadays the thermal and humidity comfort holding systems do not distinguish between loads caused by internal and external disturbances. For heating in the winter, because the formation of internal disturbances reduces the heat load, which is beneficial to maintaining the indoor thermal environment, therefore it is mostly directly ignored as a system margin; and for the summer cooling and air conditioning, the internal and external disturbances are combined to calculate the load formed, and the air conditioning equipment is uniformly used to eliminate them.

However, the reason is that the way internal and external disturbances occur and the energy grade they carry are not the same. For example, the heat dissipated in winter due to the envelope is due to the heat transfer driven by the temperature difference between the indoor temperature of ±20°C and the outdoor temperature with a large fluctuation range (-25°C-+15°C). In summer, the temperature difference is only about 10°C, that is, the temperature difference between ±26°C room temperature and outdoor temperature (25°C-36°C), as well as the load due to the sun exposure of the translucent part.

As can be seen from the temperature difference range that causes the heat transfer load of the envelope, the grade of this part of the energy quality is not high: the energy grade of the heat transfer load of the envelope in winter is only ±20°C indoor temperature, while the energy grade of the load in summer is only ±36°C for outdoor temperature. Compared to the heat source energy grade applied in conventional heating systems (water supply temperature by heat grid is of 130°C, water supply temperature by radiator is of 50 – 70°C, by low temperature heating 35°C), and the cold source energy grade applied in air conditioning systems (chiller evaporation temperature ≈2°C). Chilled water supply temperature 7°C, water supply temperature by chilled radiation 16 – 19°C), the energy match level between both side is not high. Judging by China's new contribution to thermal theory in recent years, the "(Entransy) theory" [1-3], the loss caused by the mismatch of energy grades is senseless "entransy dissipation".

By embedding an active system to the envelope structure to reduce and eliminate external disturbances, and then partially or even replace the traditional cooling and heating system, it is a new research direction in the academic circles domestic and international in recent years. Among them, Li Xianting et al. [4-7] modeled and analyzed the thermoactivated performance of piper embedded walls and piper embedded windows, Xu Xinhua et al. [8-12] did theoretical modeling and experimental verification of the embedded piper building envelop, and Zhang Zhigang et al. modeled and [13-15] nalyzed the heat pipe-mounted wall Zhuang Zhi et al. [16] modeled and measured the air circulation of the double-layer wall.

2. The Impact of Envelope Performance on Building Energy Consumption

2.1. From the Traditional Concept of Cooling and Heating to the Concept of Creating a Thermal and Humid Comfort Environment

From the perspective of energy conservation, it should be considered that the building does not "consume" energy or "convert" the input energy into other forms of energy, that is, the so-called "work" process, but only in the "inflow" and "outflow" of energy to achieve a balanced state. Jiang Yi et al. [1] proposed that the energy demand of the hot environment part of the indoor thermal and humid environment creation process is actually a process of balancing the thermal dissipation from the room to the surrounding through heat input/output. Different from the traditional concept of cooling and heating, the process control theory of indoor thermal and humidity creation believes that the task of the heating and air conditioning system is to maintain the suitable air parameters in the room through the heat transfer between the room and one or more heat sources or heat sinks. The change of indoor thermal environment is caused by the loss of balance between indoor heat gain (heat dissipation of personnel and equipment, insolation, etc.) and heat loss (heat transfer of envelop, air infiltration, etc.). As a thermodynamic closure system, the boundary is the envelope of the building. It can be considered that the indoor thermal environment balance is fundamentally the balance of thermal energy input/output with the envelope as the boundary.

2.2. Comparison of the Traditional Cooling and Heating Mode with the TAF Based on Entransy Dissipation Theory

Adopting of Guo Z-Y [1] and Liu Xiaohua et al. [2, 3] introduction entransy theory to analyze the heat transfer process, the analysis of the technology of thermal activated building envelop can be more clearly shown the advantages of this technology in the aspect of entransy dissipation.

Using the theory of entransy to analyse the thermal and humidity holding process, and focusing on the comparison between the thermal activated building envelop and the conventional heating and air conditioning system, it is necessary to simplify the heat and humidity holding process accordingly:

1) Ignore the air permeation part and the ventilation part, and only analyse the control room air temperature;
2) Ignore the heat transfer process of the indoor heat source, and only analyse the heat exchange between the indoor hot and cold end and the indoor air;
3) Ignore the unevenness of indoor air temperature, and assume that the indoor air temperature is uniform everywhere;
4) Ignore the heat transfer temperature difference between the indoor air and the inner surface of the enclosure, and assume that the temperature of the inner surface of the envelop is consistent with the room temperature;
5) Ignoring the energy storage change of the envelope material, assuming that only one-dimensional direction heat flow exists;
6) Ignoring the difference in material-to-thermal properties inside the envelope, assume that the wall temperature gradient is straight.

After making the above estimation, the entransy dissipation of heat transfer from the heating/cooling components to the envelope structure and the heat transfer from the inner surface of the envelope to the outdoor surrounding can be formed as the following typical combinations, for the winter and summer working conditions, and for the use of different mechanical systems and thermoactivated systems as Table 1 shown:

Table 1. *Types of winter and summer conditions with different mechanical approaches.*

Temperature control mode Working conditions	Mechanical system, convection heat transfer	Mechanical systems, radiant heat transfer	Thermal activated layer temperature room ≠temperature	Thermal activated layer = room temperature
Winter W	W1: Radiator heating	W2: Floor heating	W3: The thermal activated layer is lower than room temperature	W4: The thermal activated layer is equal to room temperature
Summer S	S1: Fan coil for cooling	S2: Cold radiation ceiling for cooling	S3: The thermal activated layer is lower than room temperature	S4: The thermal activated layer is equal to room temperature

In the following heat transfer model, the entransy dissipation analysis method in Liu Xiaohua et al. [12] literature is used, the inner surface temperature of the envelope structure is approximately regarded as room temperature, ignoring the heat influence of other parts (internal disturbance, air infiltration, etc.), and the heat balance between the two sides bounded by the inner surface of the building envelop is simplified to the thermal balance between the indoor heating/cooling components — the inner surface of the envelope — the outdoor environment. i.e.

$$Q_{En} = cm_c \Delta t_C = UA\Delta t_F \tag{1}$$

there into

Q_{En} Heat transfers of building envelop;

c Specific heat capacity of medium;

m_c Mass flow of heat medium;

Δt_C Temperature difference between the supply and return water at the indoor heat exchange components;

UA Heat transfer capacity of the building envelop.

If the indoor components and the inner surface of the envelope are treated as an isolated system that achieves thermal balance, the entransy dissipation is.

$$\Delta E_{n,C} = \frac{1}{2} Q_{En} \Delta t_C \tag{2}$$

$$\Delta E_{n,R} = \frac{1}{2} Q_{En} \left| \frac{t_s + t_r}{2} - t_{in} \right| \tag{3}$$

$$\Delta E_{n,en} = \frac{1}{2} Q_{En} |t_{in} - t_{out}| \tag{4}$$

there into

$\Delta E_{n,C}$ Entransy dissipation caused by the temperature difference between the supply and return water at the room component;

$\Delta E_{n,R}$ Entransy dissipation caused by the difference between the average temperature at the room component and the room temperature;

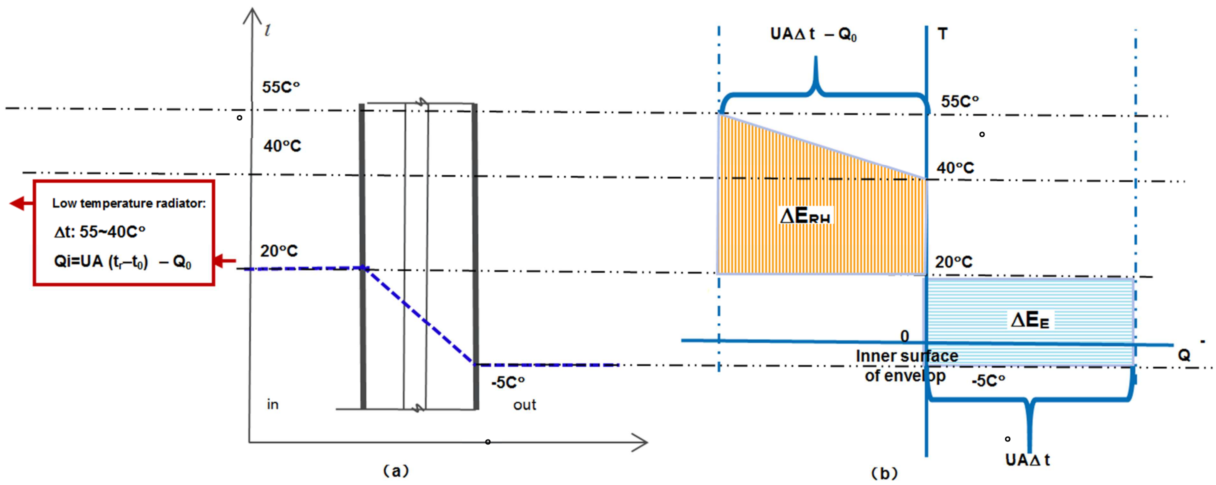
$\Delta E_{n.en}$ Entransy dissipation caused by indoor and outdoor temperature differences;

$\frac{t_s + t_r}{2}$ Water average temperature at the room component;

$t_{in} - t_{out}$ Indoor and outdoor temperature differences.

Using the TQ diagram to represent the thermal balance on both sides, the following schematic diagram can be created:

2.2.1. Entransy Dissipation by Using Radiator Heating



(a): Temperature gradient within the envelope (b): Entransy dissipation delimited by the inner surface of the envelope

Figure 1. *Temperature – entransy dissipation relationship for W1 operating conditions (radiator heating).*

Figure 1 above on the left (a) represents the case of temperature change by low temperature radiator heating in winter, while heat is transferred from radiator to the room and the indoor heat is transferred to the outdoor through the building envelop; right (b) then in T – Q diagrams is expressed on entransy dissipation each other. In this case the radiator transports heat with 15°C temperature difference between the supply and return water to the room, to reach the 20°Caverage temperature of the inner surface of the envelope, thus forms the process of the indoor component (heat source) actively providing heat to the envelope, and forms entransy dissipation ($\Delta E_{n,C}$). At the same time, due to the temperature difference between inner surface of the building envelop (semi-infinite large flat wall, 20°C) with the outdoor environment (heat sink, -5°C), that is (t_r–t_0=20°C–(-5°C)=25°C), heat transfer will also occur, i.e. the part of the building envelop entransy dissipation ($\Delta E_{n,en}$). According to the set conditions, the radiator is only responsible for UA part of the building envelop heat dissipation by Δt=25°C, the indoor heat production part will be deducted. In this case the radiator formed entransy dissipation ($\Delta E_{n,C}$) is much larger than the entransy dissipation ($\Delta E_{n,en}$), which is formed from inner surface temperature of the envelope at 20°C, and the outdoor ambient temperature at -5°C as the thermal load (excessive heat dissipation).

According to the above entransy dissipation expression, the entransy dissipation formed by using radiator is as follows:

Building envelop entransy dissipation $\Delta E_{n,en}$:

$$\Delta E_{n,en}=\frac{1}{2}Q_{En}|t_{in}-t_{\ out}|1/2\ (20-(-5))\ Q_{En}=12.5\ Q_{En} \quad (5)$$

Radiator entransy dissipation $\Delta E_{n,C}$:

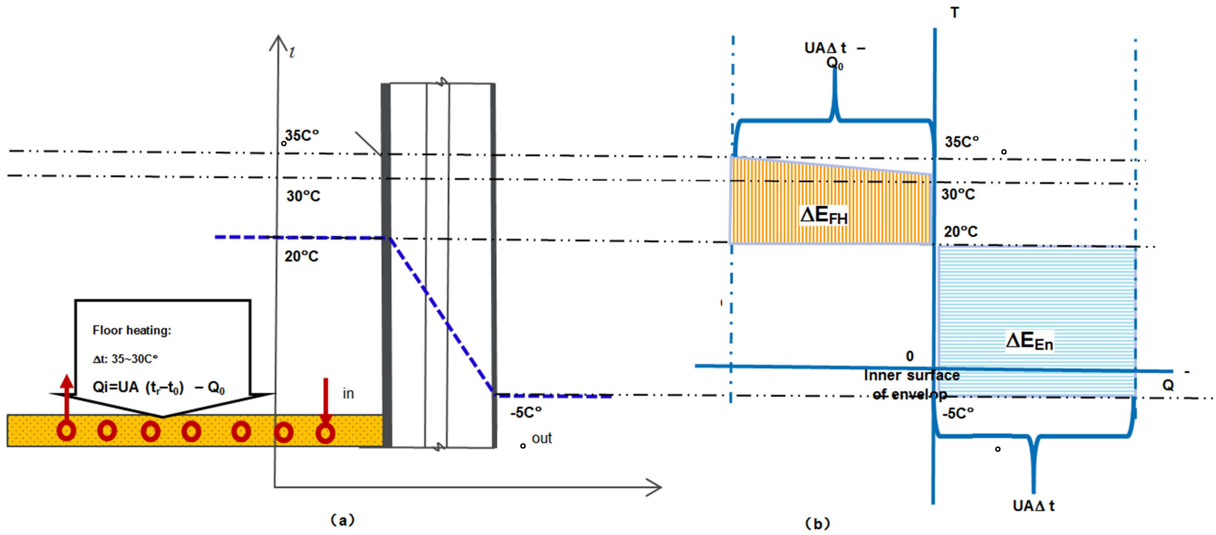
$$\Delta E_{n,C}==\frac{1}{2}Q_{En}\left|\frac{t_s+t_r}{2}-t_{in}\right|1/2\ ((55+40)/2-20)\ Q_{En}=12.75\ Q_{En} \quad (6)$$

entransy dissipation difference $\Delta E_{n,C}-\Delta E_{n,F}$:

$$\Delta E_{n,C}-\Delta E_{n,F}=12.75Q_E - 12.5\ Q_E = 0.25Q_E \quad (7)$$

It can be seen that the entransy dissipation (to provide heat to the room) by using of low temperature radiator is almost the same level of the building envelop entransy dissipation due to indoor and outdoor temperature difference caused by. This also means that the low temperature radiator for heating should be within a reasonable range, and the heating method above this temperature (90/70 radiator supply/return temperature, 130/60 heat grid supply/return temperature) will bring unnecessary entransy dissipation.

2.2.2. Entransy Dissipation by Using Floor Heating



(a): Temperature gradient within the envelope (b): Entransy dissipation delimited by the inner surface of the envelope

Figure 2. *Temperature – entransy dissipation relationship for W2 conditions (floor heating).*

Figure 2 shows that in the case of the same envelope load (inner surface temperature 20°C, outdoor temperature -5°C) i.e., the same entransy dissipation ΔE_E of the envelope part, the entransy dissipation by using flat radiant heating (35 – 30°C) on the indoor heat source side. Ignoring the different wall temperature gradient difference between floor heating to the wall heating, assuming that the wall surface of the envelope still maintains an average temperature of 20°C, as thus the entransy dissipation ΔE_E formed by the heat exchange between the surface and the outdoor environment is no difference from the previous case. However, due to the use of flat radiant heating, the water temperature is much lower than that of the radiator, so the entransy dissipation ($\Delta E_{F)}$ formed is much smaller than the entransy dissipation ΔE_R using the radiator).

In both cases, the active system (radiator, floor heating) is used to supplement the heat lost in the passive heat dissipation process of the building envelop, and the driving temperature difference between the passive heat dissipation process in this case is the winter room temperature t_r and the outdoor ambient temperature t_0, and set the surface temperature of the envelope to be the same as the room temperature. Since the driving temperature difference is linear with the outdoor ambient temperature, in the case of UA as a fixed value, once the outdoor temperature is lower than a certain value, the heat dissipation of the envelope structure is greater than the passive heat dissipation limit, and the active system needs to be activated to compensate for the excessive heat dissipation part.

Also according to the above entransy dissipation expression, the entransy dissipation formed by using floor heating is as follows:

Envelope loss entransy dissipation $\Delta E_{n,en}$ constant, = $\Delta E_{n,en}12.5\ Q_E$

Floor heating entransy dissipation $\Delta E_{n,C}$:

$$\Delta E_{n,C}==\frac{1}{2}Q_{En}\left|\frac{t_s+t_r}{2}-t_{in}\right|1/2\ ((35+30)/2-20)\ Q_{En}=6.\ 25\ Q_{En} \quad (8)$$

entransy dissipation difference $\Delta E_{n,C}-\Delta E_{n,F}$:

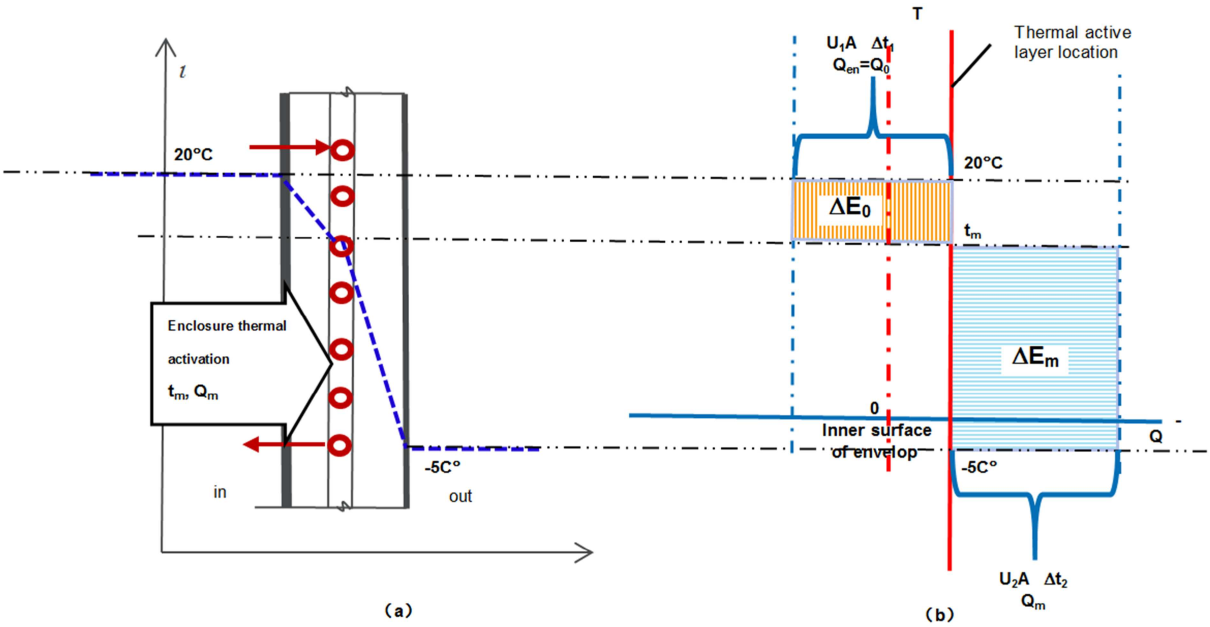
$$\Delta E_{n,C}-\Delta E_{n,F}=6.25Q_{En}-12.5\ Q_E=-6.25\ Q_{En} \quad (9)$$

It can be seen that the entransy dissipation by using of floor heating (to provide heat to the room) is even lower than the building envelop entransy dissipation, so the use of floor heating should be the "entransy optimal" choice in the mature heating variations.

2.2.3. Entransy Dissipation by Using Thermal Activated Building Envelop (t_m<t_r)

Figure 3 shows the dissipation of the entransy after the thermoactivated of the building envelope structure, which is different from the working conditions indicated in the diagram above, which is further set as follows:

1) The TAF is located somewhere in the middle of the envelope, and the temperature is uniform at all locations in this layer;
2) The temperature of this layer (t_m) is a hypothetical temperature, formed by the water circulation in the water pipe set up in this layer, and the temperature difference (t_{in}-t_{m}) between this temperature and the room temperature (20°C) is just enough to make the room heat Q_0 can be passed out ($Q_{en}=Q_0$).
3) Under this setting, the indoor heat gain Q_o will be transmitted to the TAF to maintain the indoor side in a state of thermal balance, neither too much transmission caused by room temperature drop, nor insufficient room temperature rise. The dissipation generated by this part of the heat transfer entransy is passive and no energy consumption occurs. The heat transfer from the TAF to the outdoor is the heat driven by the temperature difference between the TAF and the outdoor environment (t_m–t_{out}). Since $Q_{en}=Q_0$ is set and there is no more other supplementary heat source in the room necessary, the heat that maintains the temperature of the TAF will be directly input through the circulating water pipe implanted in the layer.



(a): Temperature distribution within the envelope (b): Entransy dissipation of bounded by the thermal active layer

Figure 3. *Temperature – entransy dissipation relationship at W3 operating conditions (thermal activation, $t_m<t_{in}$).*

The heat transfer coefficient of the building envelop from the inner surface to the TAF is U_1, and the heat transfer coefficient from the TAF to the outer surface of the building envelop is U_2, like Figure 3 shows, then

$$Q_0=U_1A\ (t_{in}-t_m), t_m=t_{in}-\frac{Q_0}{U_1A} \quad (10)$$

$$Q_m=U_2A\ (t_m-t_{out}) \quad (11)$$

And because

$$U=\frac{1}{\sum R}=\frac{1}{\sum R_1+\sum R_2}, \text{ and } U_1=\frac{1}{\sum R_1}, U_2=\frac{1}{\sum R_2} \quad (12)$$

Therefore $U_1, U_2>U$, the thermal $\sum R_1$, $\sum R_2$ are the thermal resistance from the inner surface of the building envelop to the m – point of the TAF, and the thermal resistance from the m – point of the TAF to the outer surface of the building envelop, respectively.

$$Q_0=\text{U,}A\ (t_{in}-t_{out})=U_1A\ (t_{in}-t_m), \text{ and } t_m=t_{in}-\frac{U}{U_1}(t_{in}-t_{out})<t_{in} \quad (13)$$

t_m Will be slightly lower than t_r, but higher than the temperature along the original temperature gradient without the thermally activated layer. Thus, the total amount of heat input by the TAF may be greater than Q_i (Q_{ac}), but its entransy dissipation will be much smaller than by Q_i. This also means that the thermal energy that replaces of Q_i is even lower in energy quality than the original energy quality due to room temperature.

According to the above entransy dissipation expression, the entransy dissipation by the thermally activated building envelop (tm<tr) is as follows:

The entransy dissipation of the envelope $\Delta E_{n,F}$ is unchanged and is $\Delta E_{n,F}$ =12.5. Q_E . However, due to the addition of the TAF, the dissipation of the envelope entransy will be split into two parts, one of which is to eliminate the dissipation of the heat production room entransy, that is,

Indoor heat generation entransy dissipation $\Delta E_{n,0}$:

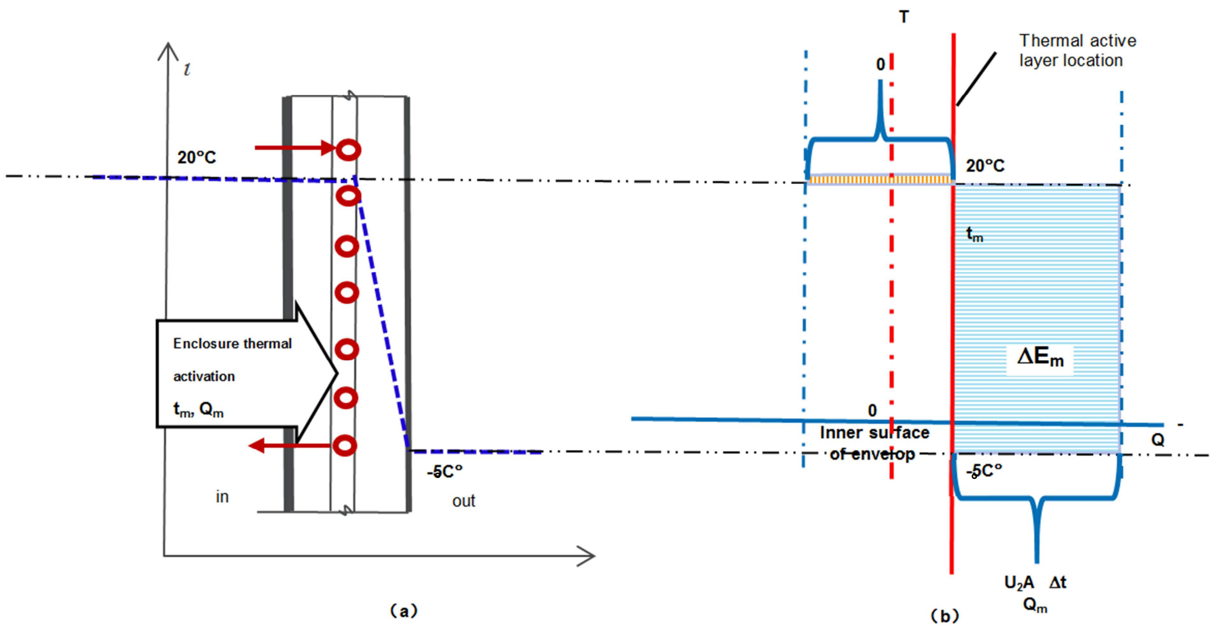
$$\Delta E_{n,0}=\frac{1}{2}Q_0(t_{in}-t_m) = 1/2\ (20 - t_m)\ Q_0 \qquad (14)$$

TAF dissipation $\Delta E_{n,m}$:

$$\Delta E_{n,F}=\frac{1}{2}Q_m(t_m-t_{out}) = 1/2\ (t_m-(-5)\ Q_m = 1/2\ (t_m+5)\ Q_m \qquad (15)$$

Due to the embedding of the TAF, the heat dissipation (Q_{en}) of the envelope is only equal to the indoor heat production (Q_0), which belongs to the part of the energy that needs to be considered even for winter conditions to be discharged outdoors, or used to compensate for other heat losses (such as due to air leakage), to ensure that the indoor temperature not rises out of control. The entransy dissipation of heat required for thermoactivated to form the controlled heat outflow, because its average temperature t_m, is lower than room temperature t_0, is therefore also less than the heat dissipation of the original envelope structure.

2.2.4. Entransy Dissipation by Using Thermal Activated Building Envelop (t_m=t_r)



(a): Temperature distribution within the envelope (b): Entransy dissipation of bounded by the thermal active layer

Figure 4. *Temperature – entransy dissipation relationship for W3 conditions (thermal activation, t_m =t_{in}).*

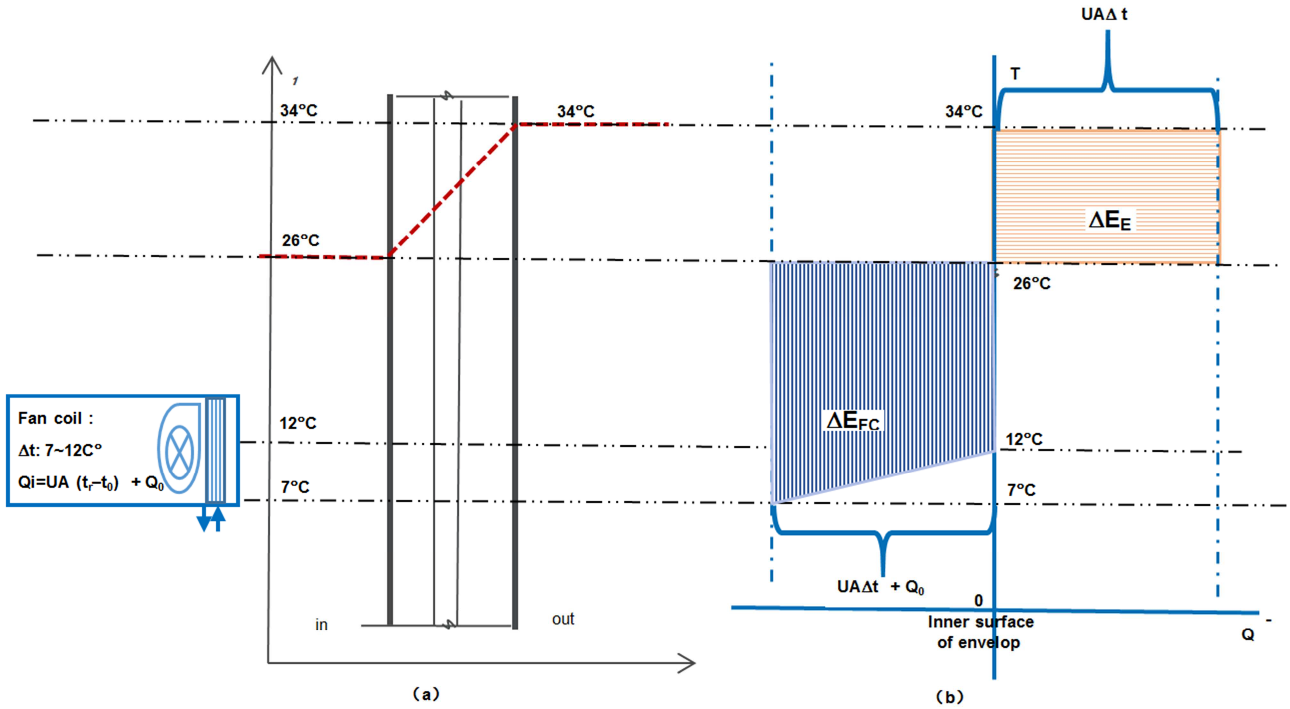
Figure 4 shows if the temperature of the TAF is the same as the room temperature, the heat transfer from the indoor to the TAF is 0, thus forming a "quasi-adiabatic" surface. In this case the heat transfer from the TAF to the outside becomes the "thermal barrier" of the building, and the water temperature that forms and maintains the thermal barrier is close to t_{in}. Under this condition, the external disturbance formed by the envelope is completely offset by the TAF, and the load of the envelope structure is 0 for the indoor side, and the indoor heat production and insolation load become the heat gained part of the room. In the absence of other balancing measures, the heating of this part can even cause the room temperature rise too high.

At this time, the entransy dissipation of the TAF entransy is the same as that formed by the original indoor and outdoor temperature differences, but due to the use of TAF technology instead of the original heating system, so that the system water supply temperature is equal to or lower than the room temperature, it can be said that the cost of its entransy dissipation can even be negligible as sacrifice.

For cooling conditions, the entransy dissipation is similar to that of heat supply.

2.2.5. Entransy Dissipation by Using Fan-Coil-Unit

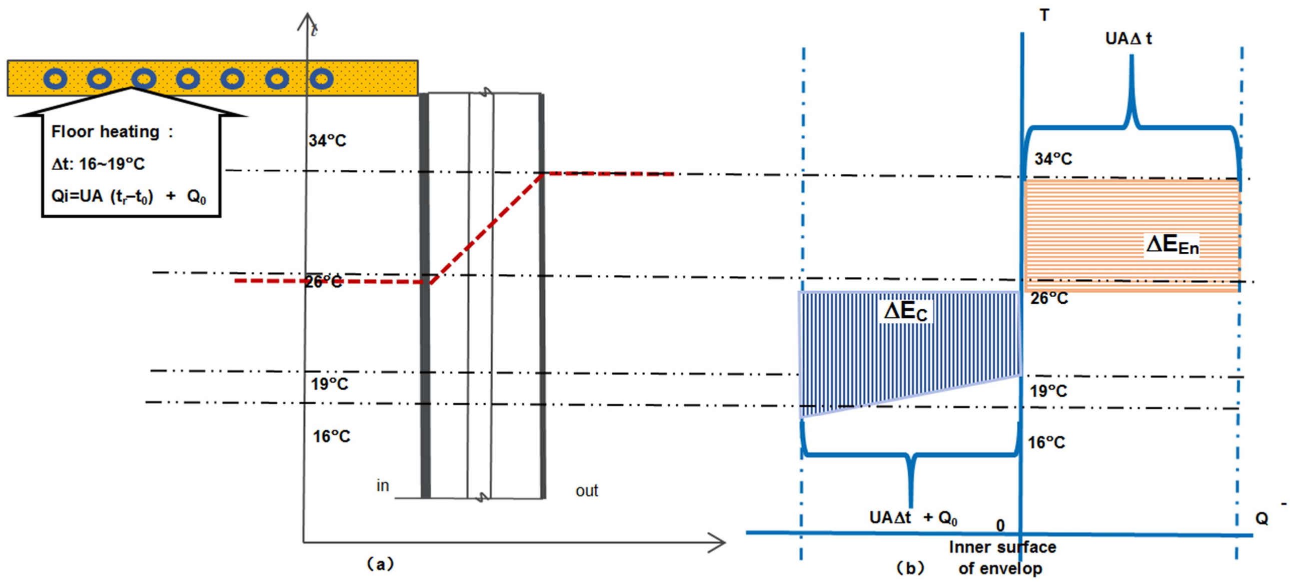
Figure 5 shows the entransy dissipation states by using of fan-coil-unit to handle the cooling load under summer conditions. Unlike winter conditions, the internal heat production Q_0 in the summer conditions, must be superimposed to the indoor load and eliminated by mechanical method.



(a): Temperature gradient within the envelope (b): Entransy dissipation delimited by the inner surface of the envelope

Figure 5. *Temperature – entransy dissipation relationship in S1 operating conditions (fan-coil-unit).*

2.2.6. Entransy Dissipation by Using Chilled Ceiling



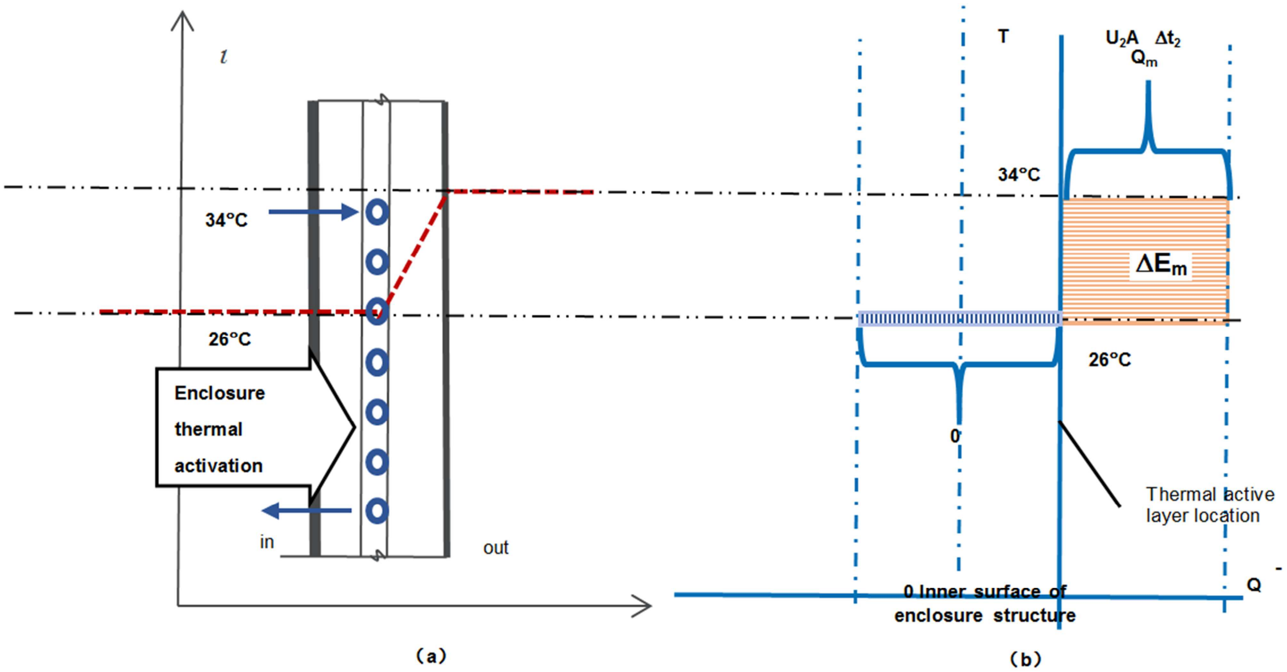
(a): Temperature gradient within the envelope (b): Entransy dissipation delimited by the inner surface of the envelope

Figure 6. *Temperature – entransy dissipation relationship in S2 operating conditions (chilled ceiling).*

Figure 6 shows the entransy dissipation by using a chilled ceiling system, and it can be seen that the dissipation generated by the system will be smaller than that of the fan coil system.

2.2.7. Entransy Dissipation by Thermoactivated (t_m=t_{in})

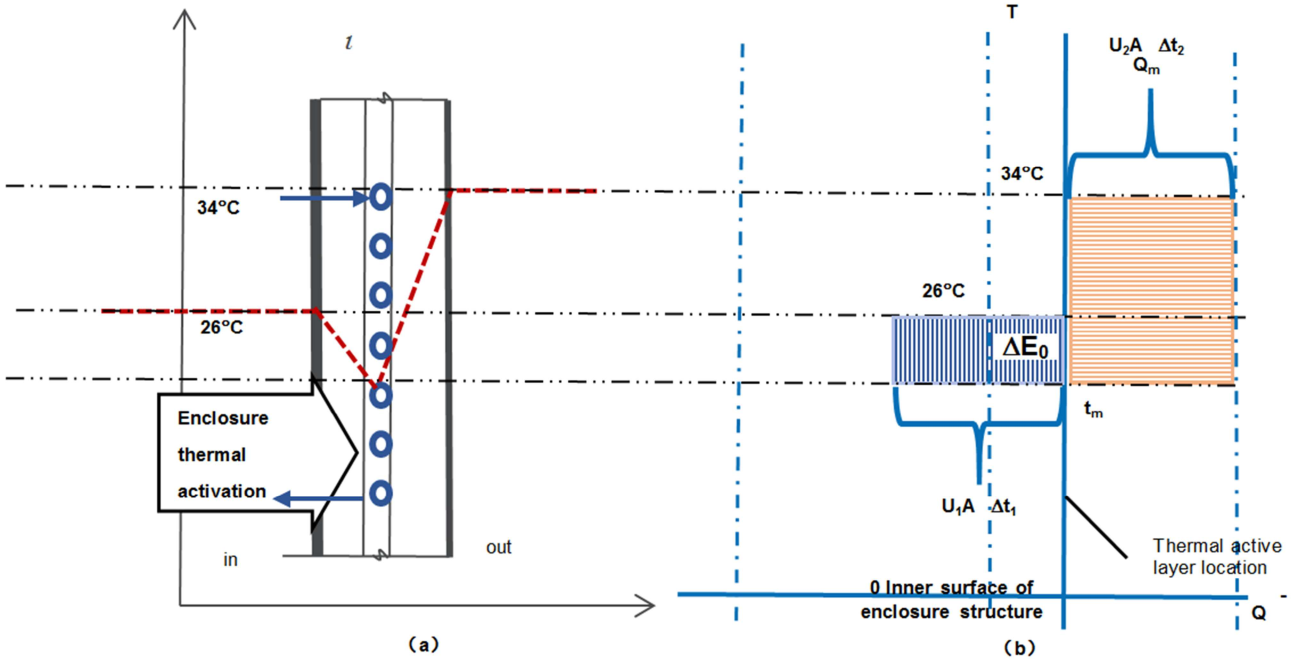
Figure 7 shows that a homogeneous temperature exactly the same as the room temperature is formed in the TAF, and the heat transfer from indoor to the layer no longer occurs due to the drive temperature difference of 0. The heat from the outside to the envelope is completely absorbed by the TAF and carried away by the circulating water in the embedded pipe.



(a): Temperature distribution within the envelope (b): Entransy dissipation of bounded by the thermal active layer

Figure 7. *Temperature – entransy dissipation relationship for S4 conditions (thermal activation, t_m=t_{in}).*

2.2.8. Entransy Dissipation by Thermoactivated (t_m<t_{in})



(a): Temperature distribution within the envelope (b): Entransy dissipation of bounded by the thermal active layer

Figure 8. *Temperature – entransy dissipation relationship at W3 operating conditions (thermal activation, t_m < room temperature).*

Figure 8 shows that a temperature t_m below room temperature is formed by circulating water in the TAF, and the setting requirements of this temperature are such that (t_{in}–t_m), which as the driving temperature is just right to discharge the room heat Q_0. At this time, the heat on the indoor side is discharged outward to the TAF to form a thermal balance, so no other auxiliary mechanical method is needed to control room temperature.

2.3. Comparison by the Traditional Cooling and Heating System Methods and the Thermoactivated to the Building Envelop

If we compare the required temperatures of different forms of heating and cooling systems, as Table 2 shown, we can see the difference in the required energy qualities.

Table 2. *Comparison of cold and heat source temperatures required for different mechanical methods.*

Temperature control mode Working conditions	**Mechanical system, convection heat transfer**	**Mechanical systems, radiant heat transfer**	**Thermal activated layer temperature≠ room temperature**	**Thermal activated layer temperature = room temperature**
Winter W	W1: Regular: 80 ~ 60°C Low temperature: 5 ~ 40°C	W2: 45 ~ 40°C	W3: <18 ~ 20°C	W4: = 18 ~ 20°C
Summer S	S1: 7 ~ 12°C	S2: 16 ~ 19°C	S3: < 26°C	S4: = 26°C

It can be seen that among all the methods, the water temperature of the W3 variant in winter is the lowest, that means the closest to the ambient temperature, while the water temperature of the summer S4 variant is the highest and also the closest to the ambient temperature. This also means that the lowest energy quality (exergy) required to prepare the corresponding grade of heat and cold media, as well as the widest range of renewable energy sources available. For example, the temperature of the TAF required in winter is 20°C, added the preparation – transmission and temperature drop due to heat transfer, finally the heat source temperature should not exceed 30°C, which will have a greater scope of application for solar winter heat utilization, waste heat utilization and so on. The same summer required 26°C TAF temperature, coupled with temperature drop, its cold source temperature should not be less than 15 ~ 20°C, close to the wet bulb temperature in many areas, so it is completely possible to use cooling towers to supply. Other areas can be supplied by surface water, groundwater or soil source heat pumps.

Using the LowEx – theory to evaluate the energy consumption of each system, and the room temperature is set as the "setpoint zero", furthermore the energy with the temperature between indoor and outdoor temperature differences is regarded as "anergy", then W1 and S1 are the conventional systems consumed, W2, S2 are LowEx systems, and W3, W4, S3, S4 can almost be defined as "anergy" systems.

3. Simulation Results of the Thermoactived Building Envelope

The existing literature has pointed out that in the winter working conditions, the TAF has two working modes: heating and thermal insulation. However, most of the simulations are committed to the use of the TAF to assume the full functionality of the original heating system, that is, replacing the heating system with the TAF. In the current study, there is still insufficient research on the extent to how far existing resources can improve the thermal performance of the envelope, so that the overall energy consumption in winter can be reduced, especially the use of existing low-quality energy to maximize substitution, so that the demand for high-quality energy consumption can be reduced, and thus save primary energy consumption.

As analysed above, the use of thermal activity technology to form a "dynamic U–value" of the envelope structure may achieve the effect of completely replacing the mechanical cooling and heating system, and then completely by the thermoactivated part of the low-quality energy to undertake the energy consumption of the building thermal environment, but at this time there are still clear quality requirements for the applied energy, that is, t_m related to room temperature and outdoor ambient temperature. However, in the case that t_m cannot be guaranteed, whether the available low-quality cold and heat source can still be used for TAF, so that the energy consumption of the necessary mechanical cooling and heating system can be considerably reduced, and the indoor comfort environment (improving the asymmetry of the indoor radiation surface) is still worth exploring by improving the temperature of the inner surface of the building envelop.

The starting point of the thermoactivated technology of the envelope structure itself is to maximization of natural energy sources, or the cold and heat source (waste cold and waste heat), which no longer have economic value. In principle, the existing cold and heat source should not be further upgraded, except the energy consumption of transportation. The question at this point is: If the hot and cold source cannot reach the temperature in the above analysis, is the cold heat still valuable? Is it possible to make the best use of the natural resources around the building through thermoactivated technology of the envelope structure, and at least achieve the purpose of reducing energy consumption under the premise of reasonable cost performance?

The concept of "dynamic insulation", that is, the use of different temperatures of the TAF temperature, forms a "dynamic U value" of the envelope structure, and finally achieves an energy-saving effect which is better than the conventional insulation method.

3.1. Introduction to Modelling Methods

Following the result of the modelling and simulation will be introduced, that the TAF using capillary technology to obtain the following three points on the Impact of thermal performance of the wall:

a) Water inlet temperature of the capillary;
b) Water flow rate in the capillary;
c) The position of the capillary TAF in the wall structure layer.

The TAF has the "equivalent insulation" characteristic, and by adjusting the water inlet temperature of the capillary, the thermal characteristics of the TAF will be changed. The "equivalent insulation" of TAF was simulated through numerical simulation analysis and comparison of multiple variants, correspondence between the characteristics and the insulation layer used in conventional perimeter envelop structures. And by selecting reasonable energy-saving technical indicators, the energy-saving effect of TAFs is studied.

The simulation mainly uses a different capillary technique than other research simulations as embedded piping system, and makes 3 assumptions about the location of the TAF as shown in Figure 9 and Table 3:

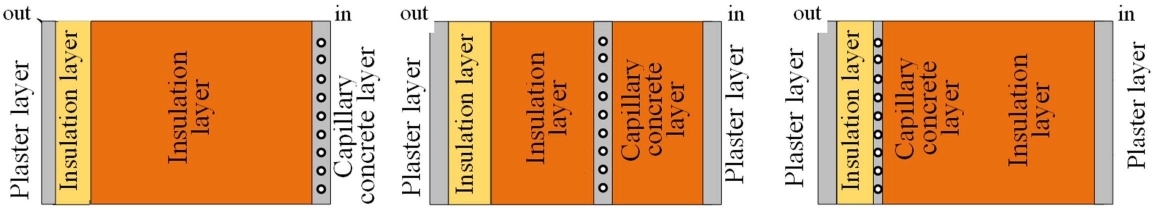
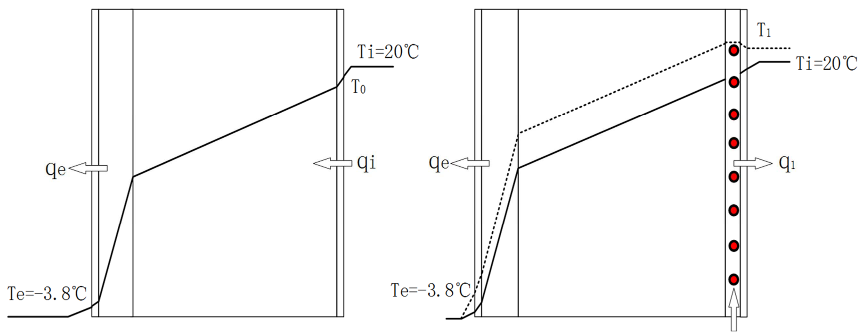


Figure 9. *3 wall forms with different locations of the TAF.*

Table 3. *Table of Thermal Parameters for Walls.*

material	Clay bricks	Insulation layer	Plaster layer	Capillary concrete layer
Thermal conductivity (W/(m·K)).	0.81	0.20	0.93	1.51
Specific heat (J/(kg·K))	1050	670	837	920
Density (kg/m^3).	1800	250	1800	2400
240 brick wall structural layer thickness (mm).	240	40	10	0
TAF structural layer thickness (mm).	240	40	10	20

Taking the winter working conditions in Zhengzhou as an example, the calculated heating temperature of outdoor in Zhengzhou is T_e= -3.8°C, and the indoor temperature is set to T_i= 20°C, then the wall temperature distribution of the natural state of the 240-brick wall is shown in Figures 2-5 (a), the surface temperature of the wall is T_0, and the surface heat flow density of the wall is q_i. Corresponding to Figure 10 (b), after the low-temperature hot water is passed into the TAF (given the input heat flow density q_{in}), the temperature distribution of the original natural state of the wall will change, at this time, the surface temperature of the wall is T_1, and the heat flow density of the surface of the wall is q_1.



(a) 240 Temperature distribution of the common wall (b) Temperature distribution of the thermoactivated wall

Figure 10. *240 Temperature distribution of walls compared with temperature distribution of TAFs.*

As shown in Table 4, according to the value relationship between q_{in} and q_i, there are four states of TAF.

Table 4. *Working status of TAFs in winter heating.*

numbering	Winter status	The temperature of the inner surface of the wall is T_1	The heat flow density of the inner surface of the wall is q_1
Status 1	Heating status	$T_1 > T_i > T_0$	$q_1>0$, $q_{in}>q_i$ (heat is transferred from the wall to the room).
Status 2	Balanced state	$T_1 = T_i > T_0$	$q_1=0$, $q_{in}=q_i$ (no heat exchange between the wall and the room).
Status 3	Equivalent insulation status	$T_i > T_1 > T_0$	$q_1< q_i <0$ (heat is transferred from the room to the wall).
Status 4	Invalid status	$T_i > T_0 > T_1$	$q_i< q_1<0$ (heat is transferred from the room to the wall).

Generally, there are four working states of TAF:

1) Equivalent thermal resistance (ER-Value) state: The TAF improves the thermal insulation capacity of the wall, the existence of the TAF in this case is equivalent to increasing the thickness of the thermal insulation layer.
2) balanced state: The wall balanced the external wall load, which generated by the change of external temperature. That means, following the change of the external ambient temperature, the corresponding adjustment of the cold and hot water temperature into the TAF, so that the heat transfer between the inner surface of the wall and the room is 0 W/m^2.

3) Heating (cooling) status: The TAF uses natural energy as an auxiliary heating (cooling) system to transfer heat (cooling capacity) to the room and reduce air conditioning power consumption.
4) Invalid state: The invalid water supply temperature not only fails to take a gain effect, but also aggravates the heat loss from the room to the TAF.

In summary, different water supply temperature ranges will make the wall in different working states. According to different outdoor environmental conditions, dynamical adjustment of the TAF as the working state can achieve the purpose of maintaining indoor thermal comfort and energy saving. Therefore, exploring the correspondence between water supply temperature and the working state of TAFs is the focus of this study.

In order to visually reflect the impact of water temperature changes on the thermal characteristics of the wall, and to exclude the interference of other factors, the simulation is set up under steady-state conditions. The outdoor air temperature is set to the average calculated outdoor temperature of -3.8°C for winter heating in City Zhengzhou, and the indoor temperature is set to 20°C. The walls are set to a north facing, the capillary layer is built between the inner plaster layer and the brick layer, and the water temperature at the capillary inlet changes from 16-28 ° C. The length of the capillary within the wall is one linear metre (1000 mm).

The following is a summary of the simulation results of the thermal characteristics of the TAF at different inlet water temperatures into capillary tubes.

3.2. Effect of Inlet Water Temperature on Thermal Characteristics of TAF

According to the case study in Table 4, in order to analyse the influence of the water inlet temperature of the capillary tubes on the inner and outer surface temperature, working state and heating capacity of the TAF, the concepts of invalid water temperature (IVMT), ideal water temperature, and ER-Value state water temperature range are defined, and the following conclusions can be drawn through the analysis of each working condition:

3.2.1. The Temperature Change Between the Inner Surface of the Wall and the Outer Surface of the Wall

From Figure 11, it can be seen that when the water inlet temperature changes from 16°C to 28°C, the average surface temperature of the wall increases from 17.7°C to 23.3°C, that is, for every 1°C increase in the water inlet temperature, the surface temperature of the wall increases by about 0.5°C, and the temperature change is obvious. Because of the presence of the external insulation layer of the wall, the external surface temperature of the wall is only increased from -3.1°C to -2.9°C, and the change is not obvious.

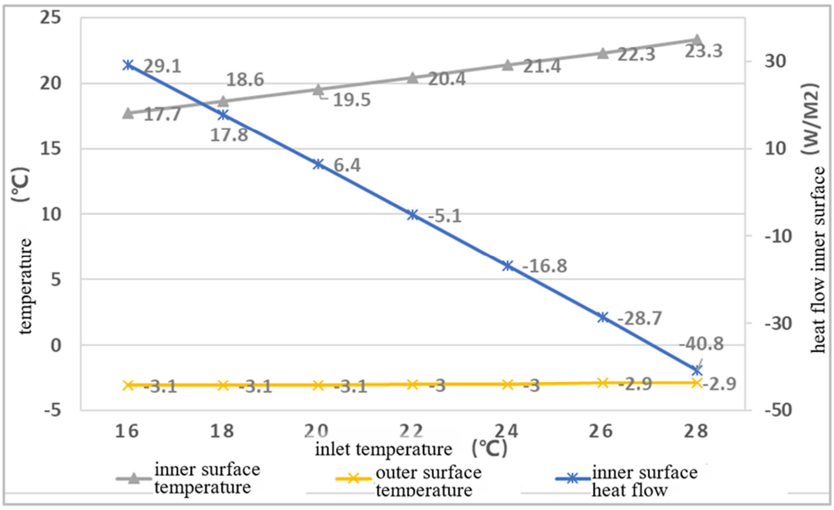


Figure 11. *The temperature of the inner and outer surfaces of the wall and the heat flow density of the inner surface of the wall change with the water temperature of the capillary inlet.*

3.2.2. Invalid Water Temperature (IVMT)

The invalid water temperature (IVMT) is the dividing line between the ER-Value state and the invalid state of the TAF, and the water inlet temperature of the capillary tubes is equal to the outlet water temperature, which is also equal to the IVMT.

Figure 12 shown the different working conditions. From working conditions 5 and 3, it can be seen that when the water inlet temperature of the capillary tubs is 20°C, the water temperature at the outlet of the capillary is 19.3°C, the water temperature in the capillary tubes is reduced, and the capillary transfers heat to the wall. The direction of heat flow density on the surface of the wall is transmitted from the room to the wall, and it can be determined that the TAF is in an ER-Value state. When the water inlet temperature of the capillary tubes is 20°C, the water temperature at the outlet of the capillary is 18.1°C, the water temperature in the capillary tubes rises, and the wall transfers heat to the capillary tubes, in this case, the presence of the TAF increases the heat flow density transmitted from the room to the wall (greater than the heat flow density on the surface of the wall in the natural state), and the TAF is in an invalid state. Therefore, when the capillary inlet and outlet temperature is equal, the water inlet temperature of the capillary tubes in this moment is the IVMT. When the inlet water temperature is 18.5°C, the capillary outlet water temperature is also 18.5°C, and the IVMT is 18.5°C.

3.2.3. Ideal Water Temperature

The ideal water temperature (IDMT) is the dividing line between the e state and the heating state of the TAF.

From working conditions 5 and 7 can be seen, when the water inlet temperature of the capillary tubes is 20°C, the water temperature at the outlet of the capillary is 19.3°C, the water temperature in the capillary is reduced, the heat flow density direction of the surface of the wall is transmitted from the room to the wall, and the TAF is in an ER-Value state. When the water inlet temperature of the capillary tubes is 22°C, the water outlet temperature of the capillary tubes is 20.5°C, the water temperature in the capillary is reduced, and the heat flow density direction of the surface of the wall is from the wall to the room, and the TAF is in a heating state. When the heat flow density transmitted from the surface of the wall to the room is 0 W/m^2, the inlet water temperature of the capillary tubes is defined as the ideal water temperature, and the IDMT is also the dividing line between the ER-Value state and the heating state of the TAF. In this case, the comprehensive heat transfer between the internal surface of the TAF and the indoor environment is zero, which is equivalent to completely eliminating the load generated by the external wall, so this is also the most ideal state for the TAF. From Figure 12 is known that when the water inlet temperature of the capillary tubes is 21.1°C, the heat flow density on the surface of the wall is 0.1 W/m^2, and the comprehensive heat transfer between the wall and the indoor environment is close to zero, so it can be concluded that the IDMT at this time is about 21.1°C.

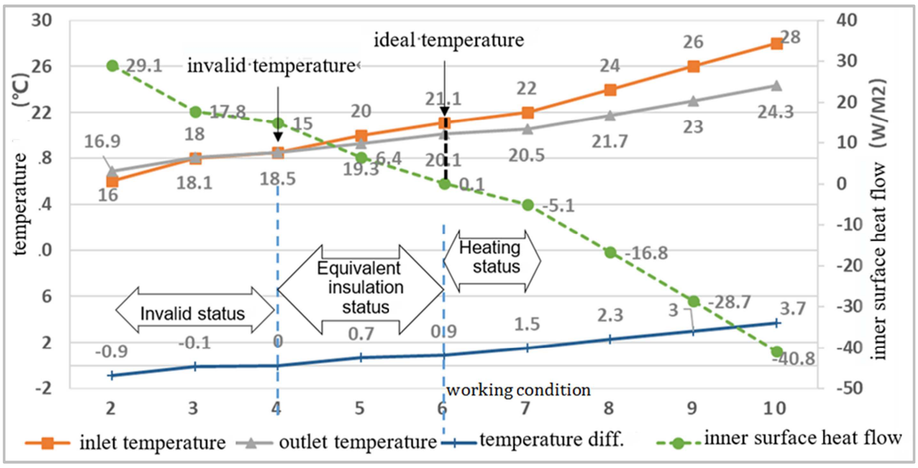


Figure 12. *Variation of water temperature at the inlet and outlet of capillary and heat flow density on the surface of the wall.*

3.2.4. Water Temperature Range as Equivalent Insulation State (ER-Value)

The inlet water temperature range when the TAF is in the ER-Value state, that means the range from the IVMT to the IDMT is called the water temperature range of the ER-Value state of the TAF.

From the working conditions 4 to 6 in Figure 12, it can be seen that when the inlet water temperature of the capillary tubes is 18.5~21.1°C, the TAF is in an ER-Value state, and the direction of the heat flow density is from the indoor to the wall, and the heat flow density on the surface of the wall is reduced from 18.0 W/m^2 to 0 W/m^2. This is because as the water inlet temperature of the capillary tubes increases, the ER-Value capacity of the TAF gradually increases until the heat flow between the internal surface of the wall and the indoor environment decreases to 0 W/m^2.

3.2.5. Working State Conversion of TAF and Heating Capacity

As can be seen from Figure 12, when the water inlet temperature of the capillary tubes is lower than 18.5°C, the TAF is in an invalid state, and when the water inlet temperature of the capillary tubes increases to between 18.5~21.1°C, the TAF is converted to an ER-Value state, and when the water inlet temperature continues to increase to 21.1°C above, the TAF is converted to a heating state.

When the TAF is in a heating state, the water inlet temperature at the capillary tubes is between 21.1~28°C. Following the increase of water inlet temperature, the heat flow density value of the inner wall surface increases from 0 W/m^2 to 40.8W/m^2, and the direction of the heat flow density is from the wall to the room. Therefore, it can be concluded that when the TAF is in a heating state, with the increase of the inlet temperature of the capillary tubes, the heating capacity of the TAF will gradually increase.

3.2.6. The Temperature Difference Between the Supply and Return Water

As can be seen from Figure 12, when the inlet water temperature of the capillary increases from 16°C to 28°C, the water supply and return water temperature difference of the capillary increases from -0.9°C to 3.7°C. When the TAF is in the ER-Value state and heating state, the capillary tubes transfers heat to the wall, and the temperature difference between the supply and return water of the capillary tubes is higher than 0°C. When the temperature difference between the supply and return water of the capillary tubes is less than 0°C, that is, the outlet water temperature of the capillary is higher than the inlet water temperature, and the capillary tubes absorbs heat from the room at this time, increasing the heat flow density transmitted from the indoor to the outdoor, and the TAF is in an invalid state. Within a certain range, with the increase of the temperature difference between the supply and return water of the capillary tubes, the heat transferred by the hot water in the capillary tubes to the wall is also increasing, the temperature on the surface of the wall gradually increases, and the heat flow density gradually increases too, TAF gradually change from an ER-Value state to a heating state. That is to say, the temperature difference between the supply and return water of the capillary tubes reflects the working state of the TAF.

3.3. Effect of TAF Position on Thermal Characteristics of TAF

The basic structural size of the TAF used in the simulation is 20 mm plaster layer + 40 mm EPS insulation layer + 240 mm brick layer + 20mm inner plastering layer, the capillary layer thickness is 10 mm, the pipe spacing is 10 mm, the embedded pipe position changes inside the wall, the position of the TAF under each working condition is shown in Table 5, and the TAF position changes in the upper part of the wall as shown in Figure 13. Considering the influence of the TAF on the working state of the TAF in different positions, the following distinctions are also made between the range of the water inlet temperature at the capillary tubes: the water temperature of the capillary inlet changes from 16~28°C in the working condition 1-10, the water inlet temperature of the capillary tubes changes from 14~28°C in the working condition 11-21, and the water temperature of the capillary inlet changes from 12~28°C in the working condition 22-34.

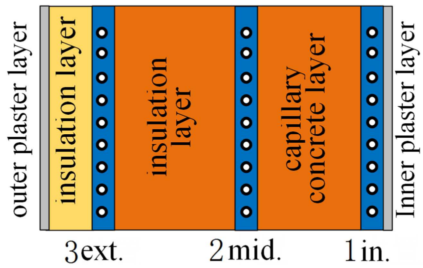


Figure 13. *TAF Position Change.*

Table 5. *Corresponding table for the location of the TAF for each operating condition.*

Condition number	TAF position	Wall structure
Working conditions 1-10	Position 1: TAF interior	The capillary layer is located between the inner plaster layer and the brick layer
Operating conditions 11-21	Position 2: TAF middle	The capillary layer is located in the middle of the brick layer
Operating conditions 22-34	Position 3: TAF exterior	The capillary layer is located between the brick layer and the insulation layer

The following is a summary of the simulation results of the thermal characteristics of the thermally activated wall structural elements at different TAF locations.

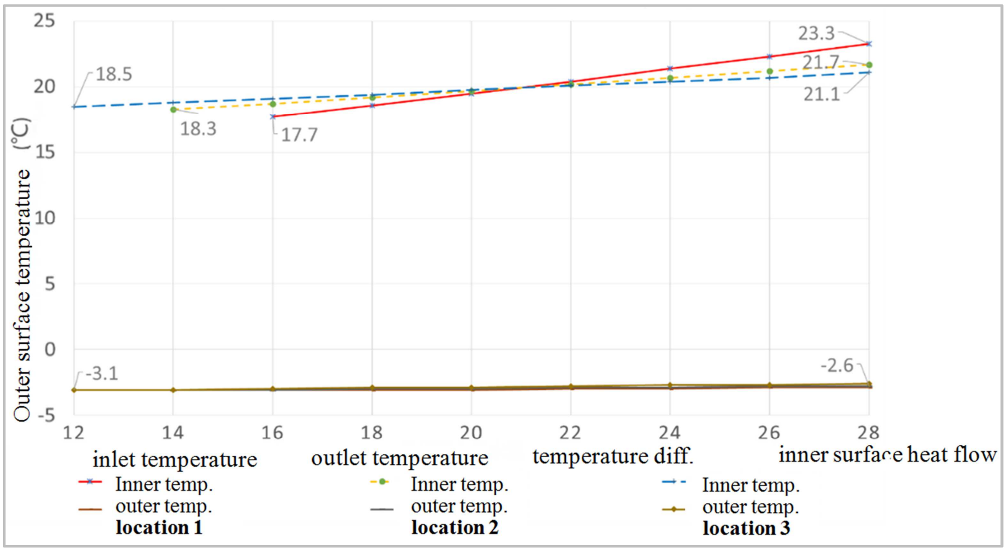


Figure 14. *Temperature changes in the interior and exterior surfaces of walls in three positions.*

The following conclusions can be drawn from the simulation data analysis:

3.3.1. The Temperature Change Between the Inner Surface of the Wall and the Outer Surface of the Wall

When the capillary layer position changes from 1-2-3, the capillary layer position changes inside the wall from the near-indoor side to the outdoor side (hereinafter referred to as the change from the inside to the outside). When the water inlet temperature is 28°C, the corresponding inside surface temperature of TAF is 23.3°C, 21.7°C, 21.1°C, respectively, and the corresponding outside surface temperature TAF is -

2.9°C, -2.8°C, -2.6°C. It can be seen that under the same inlet water temperature of 28°C, as the position of the capillary layer changes from inside to outside, the surface temperature of the TAF is reduced from 23.3°C to 21.1°C, and the temperature change is obvious; however, due to the existence of the external insulation layer of the wall, the outside temperature of the TAF is only increased by -2.9°C to -2.6°C, the temperature change is only 0.3°C, and the change of capillary position has little effect on the surface temperature outside the wall. From Figure 14 and synthesizing the above analysis, the following conclusions can be drawn: When the water inlet temperature of the capillary tubes is fixed, as the position of the TAF changes from the inside to the outside, the inside surface temperature of the TAF gradually decreases, and the outside surface temperature gradually increases, but the temperature change of the inside surface is greater than the outside surface.

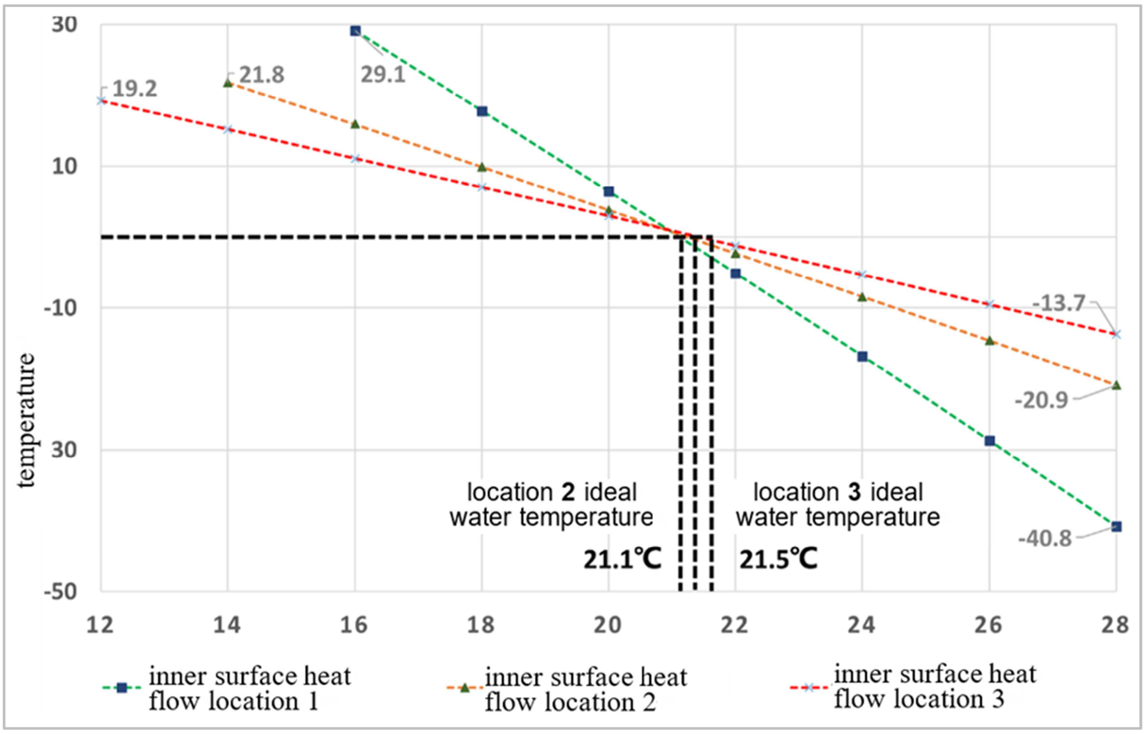


Figure 15. *The heat flow density and IDMT of the wall surface change with the position of the TAF under different capillary inlet water temperatures.*

3.3.2. Heating Capacity

From Figure 15 is known that when the water inlet temperature of the capillary tubes is set to the maximum value of 28°C, the TAF under the three working conditions is in a heating state, and the heat flow density transmitted by the wall to the room reaches the maximum value corresponding to the three positions. As the position of the capillary layer changes from 1-2-3, the distance between the capillary layer and the inside surface of the wall is from increased, and the absolute value of the heat flow density transmitted by the wall to the room is also reduced from 40.8 W/m^2 to 13.7 W/m^2. It means, it can be concluded that when the water inlet temperature is certain and the TAF is in a heating state, the heating capacity of the TAF is gradually reduced as the position of the TAF changes from the inside to the outside.

3.3.3. Ideal Water Temperature (IDMT)

The ideal water temperature (IDMT) is the boundary between the ER-Value state and the heating state of the TAF.

Analysis of working conditions 6, working conditions 17, working conditions 29 know that when the capillary layer position is located in position 1, position 2, position 3, the IDMT corresponding to the TAF is 21.1°C, 21.3°C, 21.5°C, respectively. It can be seen that as the position of the TAF changes from the inside to the outside, the IDMT of the TAF is gradually increasing, but the IDMT change is not large.

3.3.4. Invalid Water Temperature (IVMT)

Invalid water temperature (IVMT) is the boundary between the ER-Value state and the invalid state of the thermally activated wall.

From Figure 16, it is known that when the capillary layer position is located at position 1, position 2 and position 3, the IVMT corresponding to the TAF is 18.5°C, 15.8°C and 12.9°C, respectively. It can be seen that as the position of the TAF changes from the inside to the outside, the IVMT is gradually decreasing, that means the water temperature range available to the TAF is gradually increasing. (This suggests that when the water inlet temperature at the capillary available to the system is low, we should consider placing the capillary layer on the outside to match the system.).

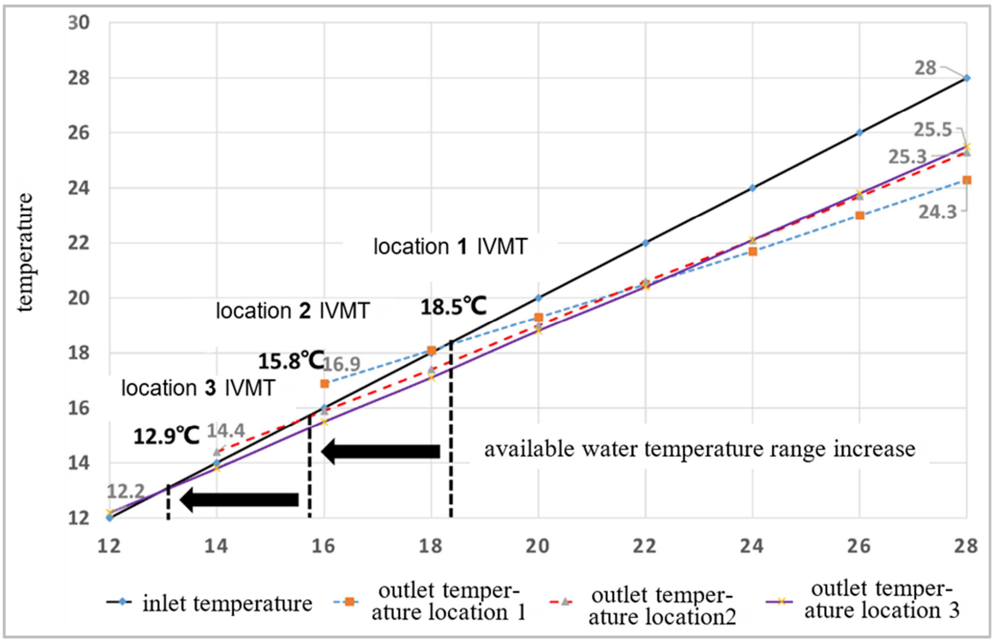


Figure 16. *Variation of water temperature and ineffective water temperature of capillary inlet and outlet under different activation layer positions.*

3.3.5. Water Temperature Range of Equivalent Insulation State

The water inlet temperature range of the ER-Value state means the range from the IVMT to the ideal water temperature.

By the analysis above is known, when the capillary layer is located in position 1, position 2 and position 3, the corresponding equivalent insulation state of the TAF has a water temperature range of 18.5-21.1°C, 15.8-21.3°C, 12.9-21.4°C, and the corresponding temperature difference is: 2.6°C, 5.5°C and 8.5°C. It can be seen from Figure 16 that when the TAF is in the ER-Value state, the water temperature range of the wall is gradually increased as the position of the TAF changes from the inside to the outside. This means the water temperature range available to the TAF is also increasing.

3.3.6. The Temperature Difference Between the Supply and Return Water

Figure 16 is shown, that in the working conditions 10, 21 and 33 can be seen, that when the capillary layer position is located at position 1, position 2, position 3, corresponding to the capillary inlet temperature of 28°C, the temperature difference of the supply and return water reaches the maximum value under the corresponding position, which is 3.7°C, 2.7°C, 2.5°C, respectively. It means in a heating state, under the same water inlet temperature, as the position of the TAF changes from the inside to the outside, the temperature difference between the supply and return water will gradually decrease, and the ability of the unit length capillary to provide heat to the wall will decrease accordingly. However, it is worth noting that the temperature drop of the unit length is limited, and within a certain range, the greater the temperature drop of the unit length, the stronger the heating capacity of the TAF.

On the other hand, under the same water temperature difference, the capillary provides the same amount of heat to the wall. As the position changes from the inside to the outside, the TAF reaches the same temperature difference required more capillary tubes. When the working state of the wall is the same, as the position of the TAF changes from the inside to the outside, the same capillary tubes can cover more of the exterior wall area to provide same amount of heat.

3.4. Effect of Water Flow Rate in the Piper on the Thermal Characteristics of TAF

When the capillary inlet temperature is 20°C, the water flow rate in the capillary changes from 0.1 m/s to 0.5 m/s, the temperature of the inner surface of the wall changes by 0.1°C, and the temperature of the outer surface does not change. When the capillary inlet temperature is 28°C, the water flow rate in the capillary changes from 0.1 to 0.5 m/s, the surface temperature of the wall changes to 0.1°C, and the outer surface temperature does not change, the heat flow density on the surface of the wall changes by only 0.3 W/m^2. It means, the change of the flow rate of the water has little impact on the thermal characteristics of the wall.

3.5. Overview of Simulation Results

For TAFs, the following indicators divide their effective mode of action:

1) The heat flow density of the wall as an indicator: from the outside to the inside is positive, from the inside to the outside is the negative value, when the heat flow density is greater than 0 for the heating condition, equal to 0 for the ideal working condition, and less than 0 for the ER-Value condition or invalid working condition. The water inlet temperature is called the ideal water temperature, when the heat flow value is equal to 0;

2) The temperature difference between the supply and return water as an indicator: the temperature difference greater than 0 is the ER-Value condition – heating condition, less than or equal to 0 is the invalid working condition - reverse (cooling) condition, and the water inlet temperature is called IVMT when the temperature difference is equal to 0.

From the above simulation results, it can be concluded that TAF can lead to the following thermal performance improvements:

1) IDMT is much smaller than the water temperature of any other heating system (21.1°C~21.5°C);
2) IVMT can be much lower than room temperature (position 3, 12.9°C);
3) In the external insulation structure, the closer the TAF is to the outer wall surface, the greater the ER-Value temperature range (position 3, 12.9°C~21.1°C).

Since the simulation uses a unit extension meter, it can be further inferred that in the TAF longer than 1 extension meter, the thermal activity effect will gradually attenuate along the length and tend to be ineffective from the ideal. The further outward the position, the smaller the attenuation, i.e., a larger area of the wall can be activated with the same amount of water and the inlet water temperature. If the inlet is the ideal water temperature, cool down the course until the IVMT is not available. According to the simulation results, it can be assumed that the inlet water temperature of 20°C is used in position 3, and the water temperature continues to decay along the length until it approaches the IVMT as shown in Figure 17, but its equivalent thermal resistance does not change much.

Based only on the results of the winter heating conditions provided by the simulation, it can be considered that the IDMT value has little correlation with the outdoor temperature, while the IVMT changes with the outdoor temperature, and the lower the outdoor temperature, the lower the invalid temperature, which also means that in the case of lower outdoor temperature, the invalid temperature can even approach 0°C. In the case of considering the use of antifreeze, it is conceivable that the ER-Value effect played by the thermoactivated of the envelope structure can replace the conventional thermal insulation material that cannot be infinitely thickened, so that the building energy-saving methods are more reasonable, and only need to invest in various low-grade heat sources (LGTE), what cannot be applied in other opportunities, or exhaust the winter heat collection capacity of solar collectors.

On the other hand, it can also be inferred that when the outdoor temperature changes, it is absolute possible to adapt to the change by controlling the temperature of the water supply until the circulation is stopped, so as to avoid the inability to discharge the indoor heat generated by excessive insulation, so that indoor overheating occurs, and even artificially causes unnecessary demand for cooling energy consumption. Similarly, simulations of circulating cold water in the summer should lead to similar conclusions, that the return of forming dynamic U-values will be much better than conventional insulation and mechanical refrigeration.

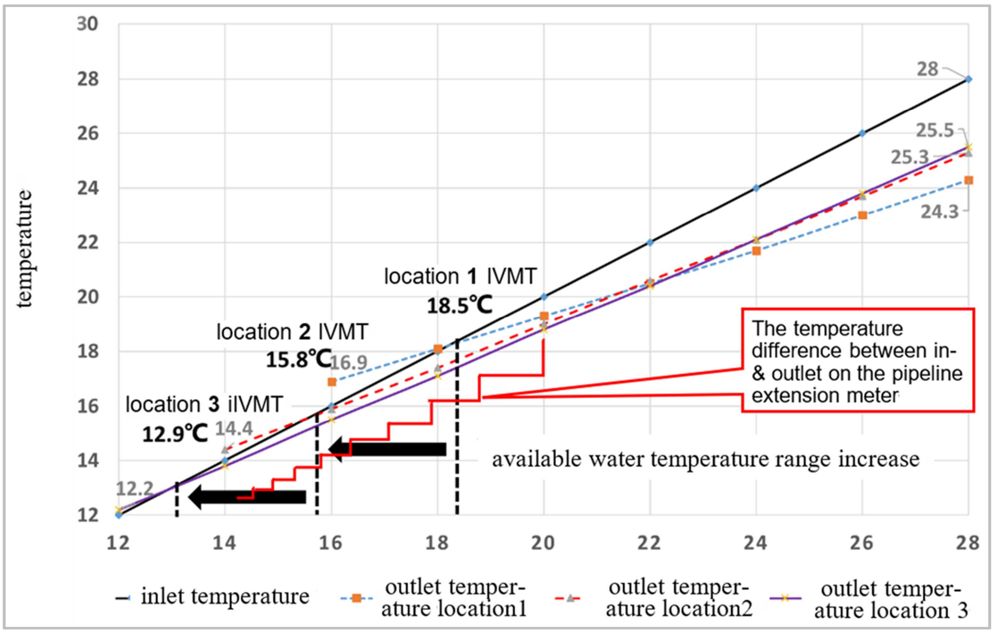


Figure 17. Changes in water temperature along with the length of capillary tubes.

4. Conclusion

The TAF has brought a new perspective to the creation of a hot and humid environment in the building. After introducing the theory of entransy, and comparing it with conventional systems, it can be concluded that TAF is the least costly way to achieve energy efficiency for external disturbances.

Without seeking to completely replace the original indoor system, the use of thermoactivated technology of the envelope can completely eliminate the impact of external disturbances on the indoor environment, and only need to provide a cooling and heat source that is between indoor and outdoor temperatures.

Through simulation analysis, it can be further concluded that even for the heating conditions in which the heat transfer of the building envelop accounts for the main part, the thermoactivated building envelop can also replace the traditional thermal insulation effect in a relatively wide range of low – temperature hot water supply, and achieve equivalent insulation of low-quality energy. In the existing various types of wall transformation technology more or less encountered fire, shedding problems, while excessive insulation caused other negative effects such as excessive heating in the transition season, high insulation cost etc., it is also worth studying and trying to use this technology as a method of energy-saving transformation with guaranteed low-grade cold and heat source (LGTE).

Declarations

1) The datasets generated during and/or analysed during the current study are available in the [CNKI] [DOI: 10.27061/d.cnki.ghgdu.2020.000880] repository, [https://kns.cnki.net/kcms/detail/detail.aspx?dbcode=CMFD&dbname=CMFD202101&filename=1020398509.nh&uniplatform=NZKPT&v=VVU0WWWKAeIWewdjszNdihv26PV7f5sfALQA-4570b3UFEQTUpTeKAmpk1hD4-5v].
2) We declare that we do not have any commercial or associative interest that represents a conflict of interest in connection with the work submitted.

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