Key technology study on thermal balance of ground source heat pump in territories with a cold winter and a warm summer

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Abstract: The imbalance of heat extracted from the soil by the ground heat exchangers (GHEs) in winter and its rejection in summer is expected to affect the long term performance of ground source heat pump (GSHP) in cooling load dominated area of China. This paper presents a new method based on traditional HGSHP system that the parallel operation mode of GHE and cooling tower (CT) to share the condensing load during low cooling load period and the series operation mode of GHE and CT to achieve cool-stage into the soil in transition season. According to this technology, the evaluation index for GHE system and cooling tower are constructed on the base of finite line heat source theory, the superposition principle and the heat and humidity transfer theory of cooling tower. Then the year-round suitable conditions of CT is analysed through theoretical analysis, practical measurement and numerical simulation. Finally, an actual GSHP project in Nanjing is given as example for system optimization and the predictive result using this new method is studied. The results show that using the new method, the "thermal accumulation" problem can be well restrained, which can provide reference on optimized system design and energy-saving operation when using GSHP system in the territories with a cold winter and a warm summer of China.

Keywords: Ground source heat pump; soil "thermal accumulation"; cooling tower for cool-stage; coupling operation condition; soil temperature field

1 INTRODUCTION

The imbalance of heat extracted from the soil by the ground heat exchangers (GHE) in winter and its rejection into the soil in summer will lead to the "thermal accumulation" for ground source heat pump (GSHP) system in cooling load dominated areas of China[1]. Traditionally, in order to resolve the "thermal accumulation" problem, a hybrid ground-source heat pump (HGSHP) was proposed with its configuration of combining the GSHP with cooling tower(CT), which in summer the CT works as auxiliary heat rejecter for GSHP system to undertake part of the cooling load. For HGSHP system, the ASHRAE[2] introduced the method of the application of HGSHP system to large public buildings. Honghee Park[3] et al. made a performance comparison between the GSHP and the HGSHP with parallel and serial configurations through experiment. Yavuzturk and Spitler[4] studied the control method and gave out the control model of HGSHP system. Jinggang Wang[5] built an experiment platform to study control strategy of HGSHP system, and presented the strategy of controlling the CT according to the difference between the outlet temperature of condenser and ambient wet bulb temperature. Zhongyi Yu [6] introduced the design method of HGSHP system and explored the optimal design strategy according to a real project. Wenjie Gang[7] used artificial neural networks(ANN) to model the ground heat exchanger instead of the ground source heat pump systems, which provided reference for optimal operation of HGSHP.

However, the thermal performance of CT is significantly restricted by the high relative humidity and temperature in hot summer and cold winter zone of China, especially during peak cooling load period (Jul., Aug.). That will lead to the higher outlet water temperature from CT than the outlet water temperature from GHE, and that will degrade the energy-saving effect of GSHP. As for this problem, the common strategy is to enlarge the design volume of cooling tower, which lead to the lower technical economy.

In order to solve the "thermal accumulation" problem, this paper presents a new method based on traditional HGSHP system that the parallel operation mode of GHE and CT to share the condensing load during low cooling load period and the series operation mode of GHE and CT to achieve cool-stage into the soil in transition season. According to this new method,the evaluation index of coupling operation mode between GHE and CT is established. And the suitable operating conditions of CT are proposed. At last, a case study in Nanjing is given as an example to analyse whether this new method is fesasible to this project.

Nomenclature		*Greek symbols*	
r	radial distance(m)	λ	thermal conductivity(W/(m·K))
r_j	distance between each borehole and the wanted point (m)		
z	depth(m)	α	thermal diffusivity(m^2/S)
τ	time(s)		
M	mass flow rate of circulating fluid(kg/h)	*Dimensionless groups*	
N	numbers of borehole		
c_p	specific heat capacity (kJ/kg·ºC)	θ	excess temperature
T_{f1}	inlet circulating fluid temperature(ºC)	ε	cooling efficiency
T_{f2}	outlet circulating fluid temperature (ºC)	β	relative cooling capacity
T_b	temperature of borehole wall(ºC)	ω	relative energy efficiency coefficient
H	depth of borehole (m)	η_q	the ratio of latent heat
q_l	heat transfer rate per meter depth of borehole(W/m)	EER	energy efficiency coefficient
Q	cooling capacity(kW)	μ	water-air ratio
t	temperature (ºC)	*Subscripts*	
W	Water flow rate（m^3/h）	W	Water
P	motor power (kW)	1	inlet
h	enthalpy (kJ/kg)	2	outlet
G	air volume (m^3/min)	S	wet bulb
d	humidity ratio (g/kg)	0	rated conditions
		f	fan
		p	pump
		a	air

2 THE PRINCIPLE AND EVALUATION INDEX OF NEW METHOD

2.1 Principle

The schematic diagram of this new method HGSHP system is given in Fig.1. The basic operation principle is given below: in peak cooling load period (Jul., Aug.), the GHE undertake all the condensing load, and during low cooling load period (Jun., Sep.), the CT work in parallel with GHE to share the condensing load, and in transition season the heat pump system stop working, meanwhile, considering the effect of outside air parameter on the performance of CT, the CT work in series with GHE during this period to achieve the cool-storage into the soil.

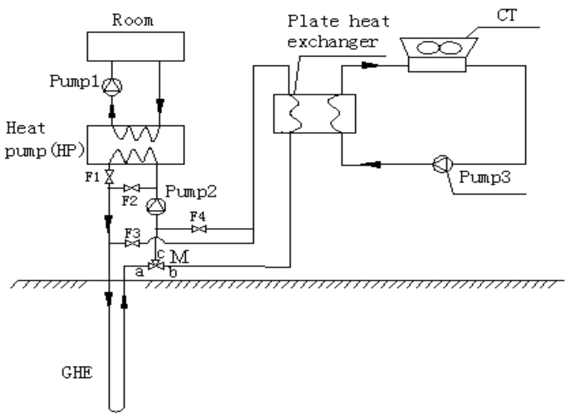


Fig.1: Schematic diagram of the new method HGSHP system

According to this new method, there are two significant parts that will affect the operation performance of HGSHP system. One is the GHE system, another one is the CT system. So it is necessary to find the optimal coupling operation condition of GHE and CT based on the air conditioning load condition and thermal characteristic of GHE and CT system.

Firstly, the ground heat exchanger (GHE). The structure and heat transfer process of a single U-tube exchanger can be expressed as Fig.2. There are many factors that have influence on thermal performance of GHE, such as properties and configuration of tubes, thermal parameters of the soil mass(conductivity, thermal capacity, density of soil), and air-conditioning load characteristic, which of the last is more important in an actual project.

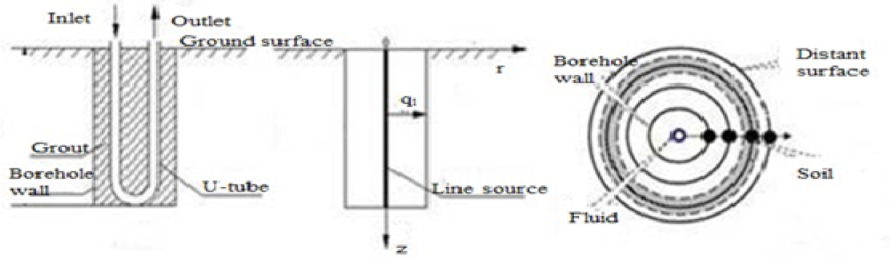


Fig.2 : Physical model in the GHE

Secondly, CT is an important heat and mass transfer equipment for HGSHP system. Its heat and mass transfer performance is not only related to structure, material characterization, but also effected by wet bulb temperature of inlet air(t_{S1}),inlet water temperature t_{W1}, water-air ratio(μ=W/G), etc. According to the heat and mass transfer principle, the closer outlet water temperature (t_{W2}) to inlet air wet bulb temperature (t_{S1}), the more sufficient heat and mass transfer process in CT, which will produces a better cooling effect (Fig.3).

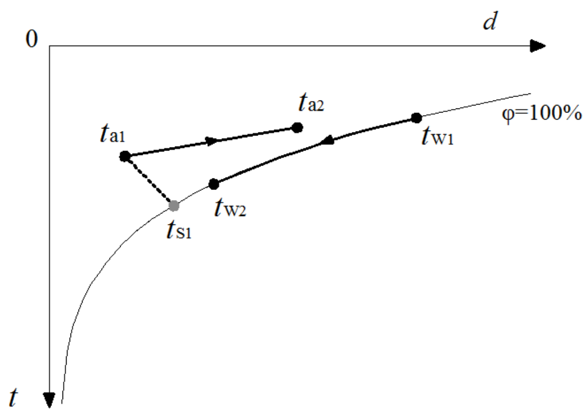


Fig.3 : Thermal process of cooling tower in *h-d* chart

2.2 Evaluation index for the new method

In order to establish the optimal coupling operation strategy of the new HGSHP system, the evaluation index of GHE and CT are built respectively.

2.2.1 Evaluation index for GHE system

In this study, the finite line heat source model is used for calculating the soil temperature field to evaluate the performance of GHE system. As shown in Fig.2, according to the finite line heat source theory of GHE [8~10], the excess temperature of any point in soil mass around GHE (one borehole) is expressed as Eq. (1). Where efrc(x) represents the complementary error function.

$$\theta = \frac{q_l}{4\pi\lambda}\int_0^H \left\{ \frac{\mathrm{erfc}[\frac{\sqrt{r^2+(z-h)^2}}{2\sqrt{\alpha\tau}}]}{\sqrt{r^2+(z-h)^2}} - \frac{\mathrm{erfc}[\frac{\sqrt{r^2+(z+h)^2}}{2\sqrt{\alpha\tau}}]}{\sqrt{r^2+(z+h)^2}} \right\} dh \qquad (1)$$

On the base of formula (1), the multiple boreholes model is used in this study in order to evaluate soil temperature field around multiple boreholes, which is more close to actual project condition. The superposition principle [11] is used for calculating the soil temperature field around multiple boreholes on the base of the assumption that the thermophysical properties of soil mass are constant. According to superposition principle, the excess temperature response at any point of the soil mass around multiple boreholes can be calculated as a superposition of the excess temperature response generated by each borehole, as shown in formula (2).

$$\theta_{\mathrm{m}}(\tau) = \sum_{i=1}^{N} \theta_{\mathrm{i}}(\tau) \qquad (2)$$

Which the $\theta_{\mathrm{i}}(\tau)$ represents the excess temperature response generated by each borehole. Adding formula (1) into formula (2) come the excess temperature response at any point of the soil mass around N boreholes (formula 3).

$$\theta(\mathrm{r}, \mathrm{z}, \tau) = \frac{1}{4\pi k}\sum_{j=1}^{N} \mathrm{q}\int_0^H \left\{ \frac{efrc\left[\frac{\sqrt{r_j^2+(z-h)^2}}{2\sqrt{\alpha(\tau-\tau_{i-1})}}\right]}{\sqrt{r_j^2+(z-h)^2}} - \frac{efrc\left[\frac{\sqrt{r_j^2+(z+h)^2}}{2\sqrt{\alpha(\tau-\tau_{i-1})}}\right]}{\sqrt{r_j^2+(z+h)^2}} \right\} dh \qquad (3)$$

It can be known that once the thermophysical properties of soil mass and the geometry of boreholes are known, the key point for calculating the temperature response at any point of the soil mass is the q value, which is the heat transfer rate per meter along borehole depth. In this paper, the hourly q value can be calculated as formula (4) through either measurement data (such as the inlet and outlet water temperature and water flow rate on source side), or through simulation of GSHP system using TRNSYS software.

$$Mc_p(T_{f2} - T_{f1}) = q_l H \qquad (4)$$

However, in actual project condition, formula (3) can not be used directly because the q value is constant in formula (3) , while the hourly heat flow value (q) in actual project are mostly transient. In order to make formula (3) applicable for calculating the temperature response of soil mass in actual project, the superposition principle of heat flow[12] are introduced. As shown is Figure 4, the basic ideology of superposition principle of heat flow is that a duration heat flow which varies continuously is equivalent to the superposition of many step heat flow (Figure 4) . As for single borehole model, the temperature response of soil mass can be calculated using superposition principle of heat flow as formula (5).

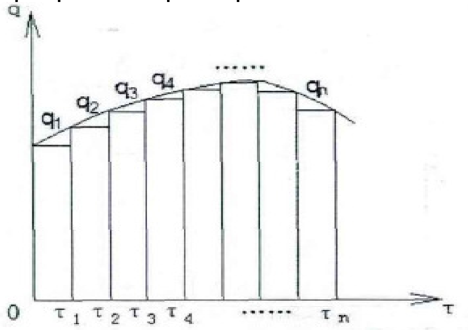


Fig.4 Schematic diagram of continuous heat flow decomposition

$$T = \frac{1}{4\pi\kappa_s}\sum_{i=1}^{n}(q_{li} - q_{li-1})g(\tau_n - \tau_{i-1}) + T_0 \tag{5}$$

Adding formula (5) to formula (3), the excess temperature response at any point of soil mass around multiple boreholes under varied continuous heat flow are given as formula (6).

$$\theta(\mathrm{r},\mathrm{z},\tau) = \frac{1}{4\pi k}\sum_{j=1}^{N}\sum_{i=1}^{n}(q_i - q_{i-1})\int_0^H \left\{ \frac{efrc\left[\frac{\sqrt{r_j^2 + (z-h)^2}}{2\sqrt{\alpha(\tau-\tau_{i-1})}}\right]}{\sqrt{r_j^2 + (z-h)^2}} - \frac{efrc\left[\frac{\sqrt{r_j^2 + (z+h)^2}}{2\sqrt{\alpha(\tau-\tau_{i-1})}}\right]}{\sqrt{r_j^2 + (z+h)^2}} \right\} dh \tag{6}$$

2.2.2 Evaluation index for CT system

Traditionally, as an evaluation index of CT, the cooling efficiency (ε) is defined as actual cooling capacity of CT (Q) over ideal maximum cooling capacity (Q_{max}) (Eq.(7)). Apparently, the bigger the ε is, the closer outlet water temperature t_{W2} to the theoretical limit temperature t_{S1}, which means higher heat and mass transfer efficiency .

$$\varepsilon = \frac{\mathrm{Q}}{\mathrm{Q}_{\max}} = \frac{t_{\mathrm{W1}} - t_{\mathrm{W2}}}{t_{\mathrm{W1}} - t_{S1}} \times 100\% \tag{7}$$

But in this paper, in order to evaluate the actual thermal efficiency of CT, three other indexes are used. First is called the relative cooling capacity (β), which means a ratio of actual cooling capacity and rated cooling capacity(Q/Q_0), β is for evaluating the extent to which the actual cooling capacity of the CT close to rated operating condition.The second is called the relative energy efficiency coefficient(ω), which represents a ratio of actual comprehensive energy efficiency coefficient and rated comprehensive energy efficiency coefficient (EER/EER_0), ω is for evaluating the extent to which the actual comprehensive energy efficiency coefficient of the CT close to rated operating condition. The last is called the ratio of latent heat (η_q), which means a ratio of latent heat and the total heat gained by the air .the bigger the η_q is, the more of the evaporation of water(Eq.(7)~(9))

$$\boldsymbol{\beta} = \frac{\mathrm{Q}}{\mathrm{Q}_0} = \frac{c_{\mathrm{p\cdot w}}\mathrm{W}\ (\ t_{\mathrm{W1}} - t_{\mathrm{W2}}\ )}{c_{\mathrm{p\cdot w}}\mathrm{W}_0(\ t_{\mathrm{W1,0}} - t_{\mathrm{W2,0}}\ )} \tag{8}$$

$$\omega = \frac{EER}{EER_0} = \frac{\dfrac{Q}{P_f + P_P}}{\dfrac{Q_0}{P_{f0} + P_{P0}}} \tag{9}$$

The key point is to get the outlet water temperature(t_{W2}) for comparing heat and mass transfer performance of CT on the basis of Eq.(7)~(9). This research adopts the heat and mass transfer model——four-variable model [13] which is proposed by former soviet scholars. Then the finite difference method is used to discrete model equation and calculate through Matlab2006 [14].

3 SUITABLE CONDITION ANALYSIS FOR YEAR-ROUND OPERATION OF CT

The heat and mass transfer performance of CT is affected by many factors, including season characteristic, ambient meteorological parameters、inlet water temperature 、water flow rate、air volume, etc. Manufactures usually only provide cooling capacity Q_0 under rated operating condition, water flow rate W_0 and air volume G_0 under its corresponding operating condition. However, for a real project, as the variation of seasons and outdoor meteorological parameter, the operating condition of CT will deviate from its original rated operating condition in most situations. That lead to the actual cooling capacity Q is different from its rated cooling capacity Q_0 (Which is usually ignored). This research will discuss the suitable conditions for CT under off-rating condition in all year round based on evaluation indexes from section 2.2.2.

3.1 Calculation condition

For the sake of convenience, this research focuses on meteorological parameters in Nanjing, China. Figure 5 shows distribution of outdoor air status on *h-d* diagram for the city of Nanjing in the typical meteorological year. Table 1 shows the major technical parameters of CT for researching.

From fig. 4, it illustrates that the peak cooling load period is basically in the days of hot and humid weather (Jul., Aug.). Outdoor wet bulb temperature reaches up to 25℃, which is not beneficial for CT operation. While it is feasible to CT operation when outdoor wet bulb temperature dramatically down to 21℃ during the low cooling load period (Jun., Sep.). What is more, the outdoor wet bulb temperature is below 13℃ in transition seasons and winter.

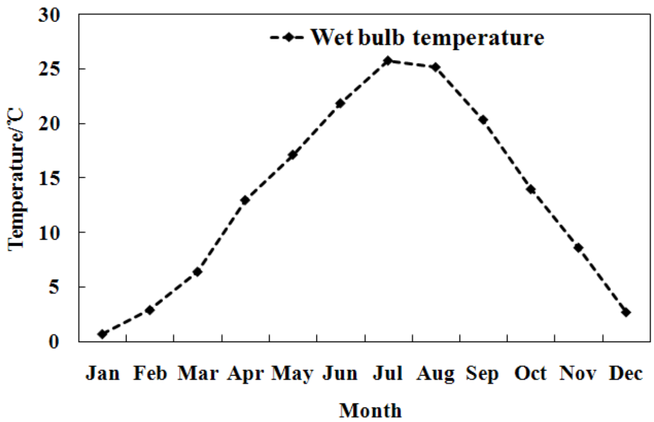


Fig.5 : Variation of outdoor wet bulb temperature in Nanjing,China

Tab. 1 Parameter description of CT

G_0 (m^3/min)	W_0 (m^3/h)	$t_{w1,0}$ / $t_{w2,0}$ (℃)	t_{S1}. (℃)	Q_0 (kW)	EER_0	μ
3560×2	390	37/32	28	2275	43.75	0.76

In order to analysis the effect of ambient meteorological parameters (especially wet bulb temperature t_{S1}), inlet water temperature t_{W1}, water-air ratio (μ=W/G) on heat and mass transfer performance for CT and the extent of cooling capacity deviate from rated working condition, two cases of calculation condition (Case 1, Case 2) are used(table 2) . The inlet water temperature of CT is 32□ in Case 1 (by reference the inlet temperature of CT in Jun. and Sep. from the real project on chapter 4). And in transition season, when operation mode is the coupling state of CT in series with GHE, considered the soil temperature will impact on inlet temperature of CT (outlet temperature of GHE), setting inlet temperature should accord with soil temperature in corresponding months (by reference the soil temperature in corresponding months from the real project on chapter 4). Furthermore, VWV (variable water volume) operating contributes more for improving the energy efficiency than VAV (variable air volume) operating [12]. Therefore, this research is mainly focus on the effectiveness of VWV operating on CT's heat and mass transfer performance.

Tab. 2 Calculation condition

Calculation condition	t_{S1} (℃)	.t_{W1} (□)	G (m^3/min)	W (m^3/h)	μ
Case 1	18,19,20,21,22	32	7120 (G_0)	130~780 ($0.33W_0$~$2.0W_0$)	0.25~1.5
Case 2	6.1/8.5/10.8/12.6	19.8/24/24.2/19.1	7120 (G_0)	130~780 ($0.33W_0$~$2.0W_0$)	0.25~1.5

3.2 Results and analysis

3.2.1 Results from Case 1

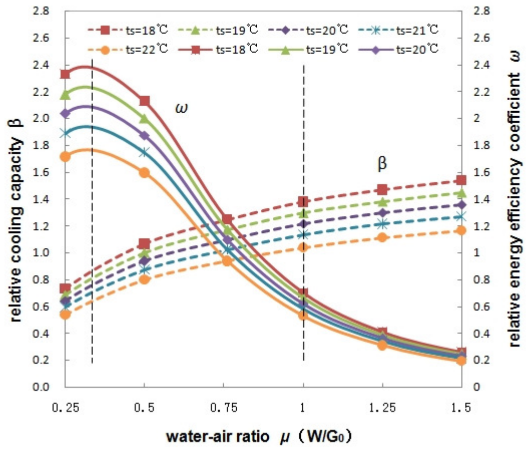
Figure 6 shows the variation of CT's relative cooling capacity (β), relative energy efficiency coefficient (ω), cooling efficiency (ε), outlet water temperature (t_{W2}), and the ratio of latent heat (η_q) by changing the water flow rate of CT(W) during low cooling load period in summer (Jun., Sep.).

Figure 6 (a) illustrates that: 1) CT's relative cooling capacity (β) increases as water-air ratio μ increases from 0.25 to 1.5 ($\mu=W/G_0$, $W=0.33W_0$~$2.0W_0$). The trend is steep at first then mild. When it reaches the maximum β=1.53, it shows increasing water flow rate will help to improve CT's cooling capacity. 2) Relative cooling capacity (β) tends to decrease while outdoor wet bulb temperature rises. It means outdoor wet bulb temperature directly impact on heat and mass transfer performance of CT. 3) When water-air ratio μ=0.3, CT's relative energy efficiency coefficient (ω) reaches its peak 2.4, then sharply decreases. This trend is not influenced by the variation of outdoor wet bulb temperature, which means an excessively increasing of water flow rate will lead to energy consumption increases for pump and degradation for system's energy efficiency coefficient.

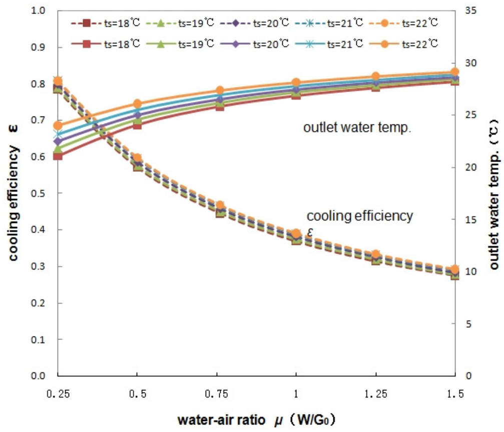
In addition, Figure 6 (b) illustrates that: 1) CT's cooling efficiency ε goes down when water-air ratio increases from 0.25 to 1.5. This trend is not impacted by variation of outdoor wet bulb temperature (t_{S1}=18□~22□). The reason is based on Eq.(3), solely increasing water flow rate without changing air volume will result in gradual rising of outlet water temperature. Numerator keeps decreasing while denominator almost constant.

Figure 6(c) reveals that changing water flow rate will affect the ratio of latent heat (η_q) in Case1. With water-air ratio increasing, the ratio of latent heat (η_q) decreases gradually and then tends to stable. This means the evaporation of water possess a large proportion in heat and mass transfer process.

Learning from results in Case1 (Fig. 6). In Jun. and Sep., relative cooling capacity β=0.6~1.4, relative energy efficiency coefficient ω=0.5~2.4, the difference temperature between inlet and outlet Δt =4~10□, the ratio of latent heat η_q=0.8~0.9 , t_{W2}- t_{S1}=2~8.8□ when water-air ratio μ=0.3~1(μ=W/G_0 , W=0.4W_0~1.33W_0).



(a)The impact of μ to β, ω



(b)The impact of μ to ε, t_{W2}

Fig.6: The impact on heat and mass transfer performance by changing the water flow rate in Jun. and Sep.(Case1)

3.2.2 Results from Case 2

Using the same method in Case 1 to analyze the heat and mass transfer performance in Case 2. The result shows that it has limited impact on improve cooling capacity by increasing water flow rate (water-air ratio μ). Conversely, relative energy efficiency coefficient (ω) is reduced. With outdoor wet bulb temperature dropping, the ratio of latent heat (η_q) is also reduced. Moreover, the heat and mass transfer performance is not only influenced by outdoor wet bulb temperature, but affected by inlet water temperature. So the suitable water-air ratio μ=0.5~0.76(μ=W/G_0 , W=0.67W_0~W_0), relative cooling capacity β=0.4~1.1, relative energy efficiency coefficient ω=0.4~1.9, temperature difference between inlet and outlet Δt = 2.2~7□ , the ratio of latent heat η_q=0.65~0.87 , t_{W2}- t_{S1}=3.6~10.1□.

As for the transition season, when may be not all the time are suitable for cool-stage because of outdoor wet bulb temperature and soil temperature, two indexes are introduced in order to find the optimal operation strategy during this period. The first is relative cooling capacity(***β)***, in this paper, the critical value of relative cooling capacity is defined as 0.6, which means it is an unsuitable cool-stage time when relative cooling capacity β less than 0.6. The second index is temperature difference between soil temperature and the outlet water temperature of CT. Similarly, the critical value of temperature difference is defined as 5 □, which means it is an unsuitable cool-stage time when temperature difference less than 5 □.

According to the above analysis result and two important indexes, the suitable operation period for CT during transition season are Oct.16 to Oct.31, November and March.

Tab. 3 t_{W1}, t_{W2}, β, $\Delta t'$ of cooling tower in Nanjing of China (μ=0.5~0.76)

Month	t_{W1} □	t_{W2} □	Soil temp. □	β	$\Delta t'$ □
Mar.	19	13.6 ~ 15	19.8	0.72 ~ 0.81	4.8 ~ 5.2
Nov.	24	17 ~ 18.7	24	0.94 ~ 1.1	5.3 ~ 7
Oct.16 to Oct.31	24.2	17.9 ~ 19.4	24.2	0.84 ~ 0.96	4.8 ~ 6.3

4 APPLICATION ANALYSIS

4.1 Project description

A high-rise residential building with 7,000m^2 area located in Nanjing of China is used as the sample building. The cooling season is from May to October, and heating season is from December to February. The indoor temperature and humidity are maintained about 23 □to 24□and 50% to 55% respectively. The system is shown in solid line of Fig.7. Correspondingly, the GHE has 1111 boreholes arranged in liner configuration, and each borehole with a depth of 35 m and the borehole spacing is 5 m. The initial temperature of soil is 18□. And the soil mass has thermal conductivity of 1.5 W/ (m·K), density of 1500kg/m^3, and specific heat value of 1800 J/ (kg·K). The buried pipe are made of PE, which has a thermal conductivity of 0.35 W/ (m·K).

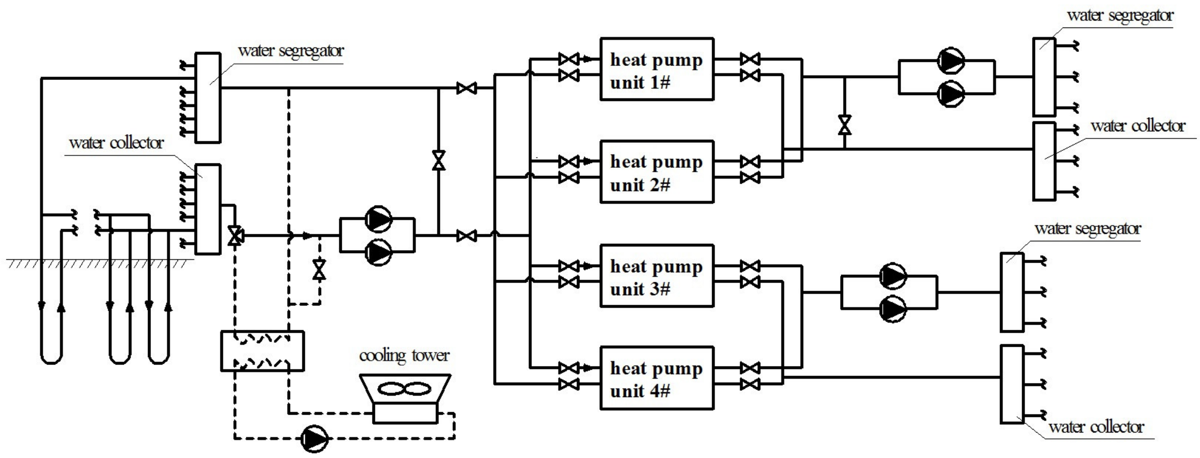


Fig.7 Schematic diagram of GSHP system

According to the analysis of the measured data on GSHP system in 2009, the heat rejected to soil in summer accounts for 61% of the annual total accumulated heat, and the heat extracted from soil in winter accounts for 39% of the annual total accumulated heat. The imbalance rate of the heat rejected to soil in summer and that extracted from soil in winter has reached up to 55.6 %. Especially, as shown in Figure 8, during peak cooling load period in summer, the average inlet/outlet water temperature can be up to 36~38□/ 32 ~35□, which leads to the performance degradation of heat pump.

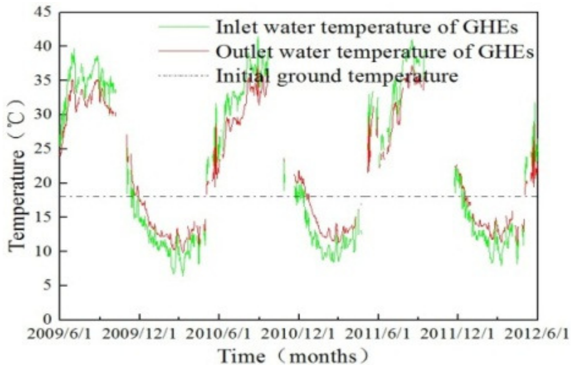
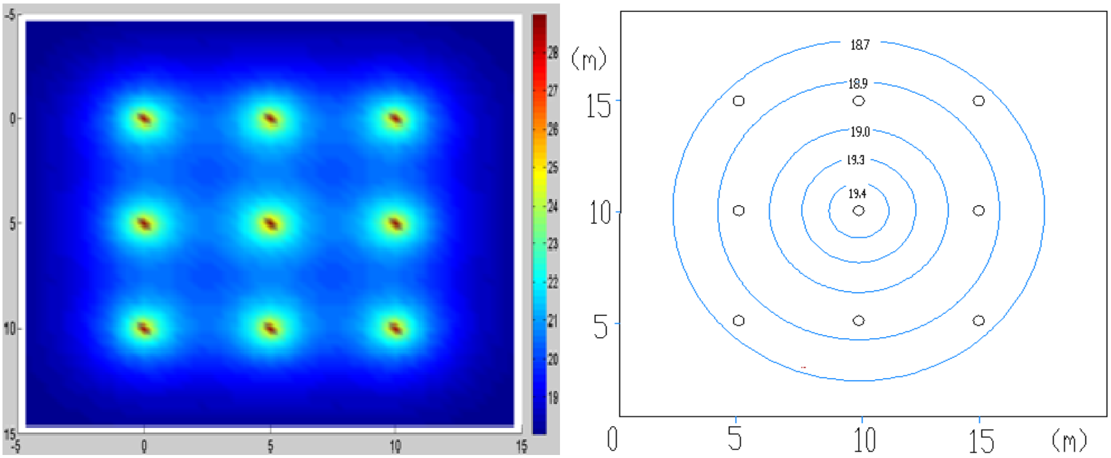


Fig.8：Inlet and outlet water temperature of GHE in 2009~2012

Using the multiple boreholes model, a 9 boreholes model (3×3) is constructed. And the soil temperature field around this 9 boreholes under Z=17.5m plane is predicted according to formula (4) and (6) through the measurement data (inlet and outlet water temperature and water flow rate on source side). Figure 9 are the temperature field which is given as isotherm. The results show that the temperature nearby boreholes reached to 28℃ at the end of the cooling season(October 8th); and after one year's operation(May 31th), the temperature nearby boreholes reached to 19.4℃,which is 1.4 ℃ higher than the initial value. The result with concentric circles size in Fig.9 b indicates that the thermal disturbance at the mid borehole is stronger than outside boreholes, although these 9 boreholes have had a spontaneous recovery period (from March to May).



a) End of cooling season (October 15 th) **b) One year later(May 31th)**

Fig.9：Predictive temperature field at Z=17.5m plane without the new method

4.2 System optimization using the new method

Using the new method based on traditional HGSHP system, the schematic diagram of new HGSHP system is shown as Fig.7, which the dotted line part is the adding equipment. The added CT has a model No. of DBNL3-80, which has an air volume of 56,000 m^3/h, 80 m^3/h of its water flow rate and the motor power of 2.2 KW. Figure 10 is the schematic diagram of the proposed operation method. Namely, A1 represents the heat rejected into the GHE in summer; A2 represents the heat extracted from soil in winter; A3 represents the heat rejected into the air through CT in low cooing load period in summer; A4 represents the heat removed by the coupling operation combining the CT with GHE in transition seasons. According to this actual project, the imbalance rate of the heat rejected to soil in summer and that extracted from soil in winter has reached up to 55.6 %((A1-A2)/ A2). Accordingly, base on the optimization strategy in Fig.7, the heat imbalance can be solved by the contribution of part A3 and A4, with the relationship between 4 parts can be expressed as Eq.(7):

$$A1\text{-}A2 = A3 + A4 \tag{7}$$

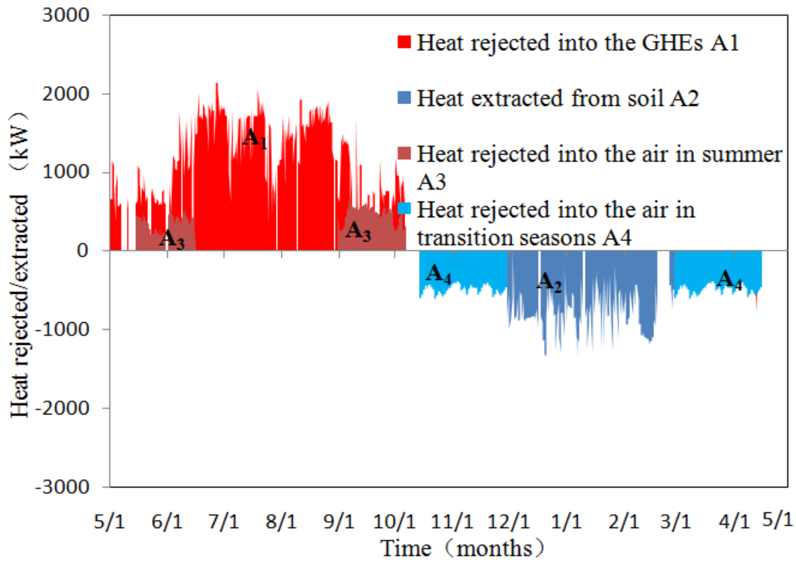


Fig.10: Schematic diagram of the new method

4.3 Feasibility analysis

According to the optimization strategy in chapter 4.2, a TRNSYS model is built as Fig.9 for calculating the outlet and inlet water temperature of GHE. All the input data are from measurement statistics in chapter 4.1 and 4.2. And the soil temperature field around the same 9 boreholes above is predicted also by the multiple boreholes model in chapter 2.

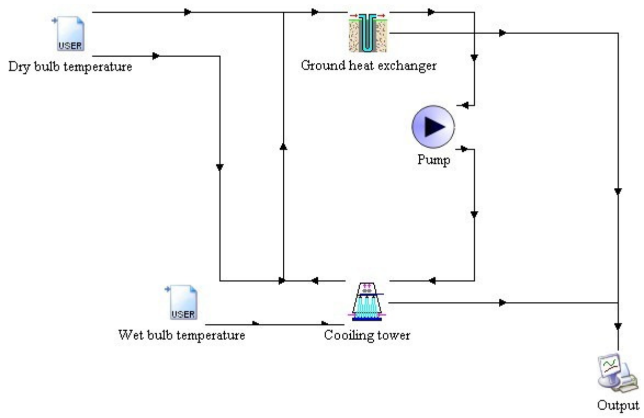
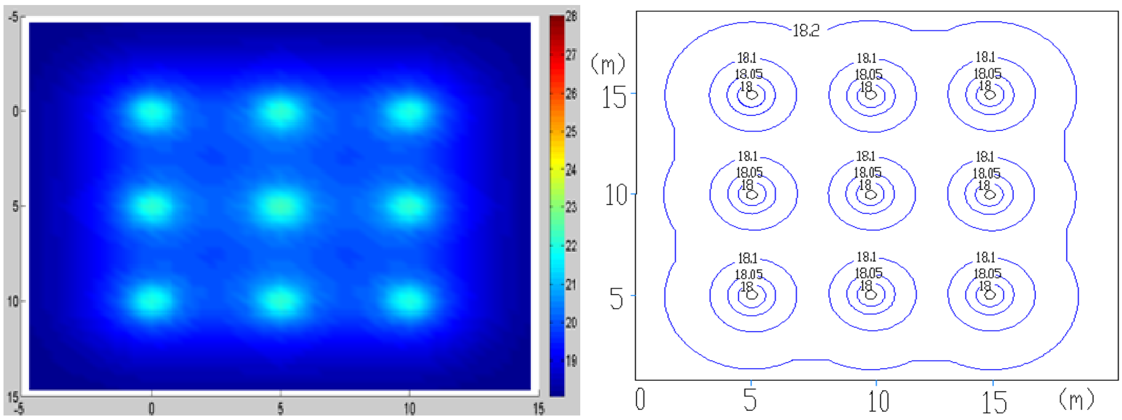


Fig. 11: TRNSYS model for calculating analysis

Combining the calculating result of TRNSYS model with Eq.(1) and Eq.(2), the soil temperature field under the same plane above are predicted. The results are shown in Fig.12. Using the new method, the temperature nearby boreholes reached to only 22℃ at the end of the cooling season(October 8th), which is about 6℃ lower than the same time when this method isn't used; and after one year operation, the temperature field are almost turn back to its original value(18℃) at whole plane. And using TRNSYS software, the three year operation simulation using the new method shows in Fig.13, the outlet water temperature of GHE is decreased from 35□ (peak cooling load period in 2011 summer) to 29□. The thermal accumulation problem is controlled effectively.



a) End of cooling season (October 15 th) **b) One year later(May 31th)**

Fig.12：Predictive temperature field at Z=17.5m plane using the new method

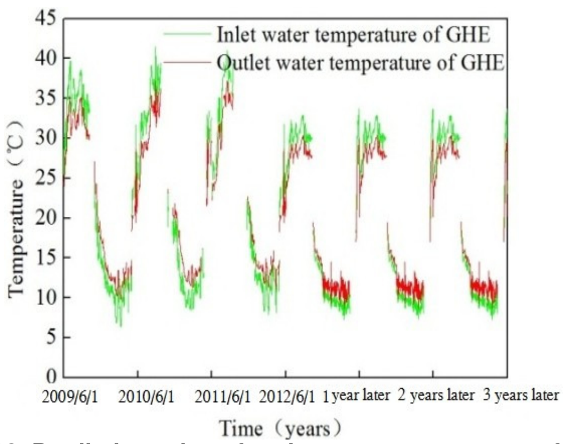


Fig.13 Predictive value of outlet water temperature of GHE

5 CONCLUSION

1) In order to solve the “thermal accumulation” for ground source heat pump (GSHP) system in cooling load dominated areas of China, this paper presents a new method based on traditional HGSHP system that the parallel operation mode of GHE and CT to share the condensing load during low cooling load period and the series operation mode of GHE and CT to achieve cool-stage into the soil in transition season.

2) Base on the four-variable model of CT, the year-round suitable condition of CT is studied. The results show that for Nanjing in China, during low cooling load period in summer, the suitable water-air ratio μ =0.3~1, and in transition season (Oct.16 to Oct.31, November and March), μ=0.5~0.76.

3) Using the multiple boreholes model in chapter 2, the predictive performance of GHE system by using the new method in an actual project in Nanjing is studied. The results show

that the new method can effectively solve the imbalance problem of the rejected/extracted heat, and the soil temperature field is almost falling back to its original value.

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