An Optimization Strategy for the Control of Small Capacity Heat Pump Integrated Air-conditioning System

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Abstract: This paper studies the optimization of a small-scale central air-conditioning system, in which the cooling is provided by a ground source heat pump (GSHP) equipped with an on/off capacity control. The optimization strategy aims to optimize the overall system energy consumption and simultaneously guarantee the robustness of the space air temperature control without violating the allowed GSHP maximum start-ups number per hour specified by customers. The set-point of the chilled water return temperature and the width of the water temperature control band are used as the decision variables for the optimization. The performance of the proposed strategy was tested on a simulation platform. Results show that the optimization strategy can save the energy consumption by 9.59% in a typical spring day and 2.97% in a typical summer day. Meanwhile it is able to enhance the space air temperature control robustness when compared with a basic control strategy without optimization.

Keyword: Ground source heat pump; Air-conditioning system; Optimization; Robustness; Maximum start-ups number; On/off control

Nomenclature

a_0 - a_3 , b_0 - b_1 , c_0 - c_3	Coefficients	C	Thermal capacitance (kJ/K)
c	Specific heat {kJ/(kg·K)}		
c_{pw}	Specific heat of water at constant pressure {kJ/(kg·K)}		
h	Sampling interval (s)	J	Objective function
M	Mass flow rate (kg/s)		
$\bar{M}$	Maximum allowable bound of the mass flow rate (kg/s)		
$\underline{M}$	Minimum allowable bound of the mass flow rate (kg/s)		
N_c	Samplings in one cycle time	NTU	Number of transfer units
N_{start}	Allowed maximum start-ups number of the GSHP per hour		
PLR	Partial load ratio	Q_{HP}	Cooling capacity of heat pump (kW)
$\hat{Q}_{HP}$	Mean cooling capacity over the on time during on/off cycling period (kW)		
q	Cooling load (kW)		
$\hat{q}$	Mean value of the cooling load in one GSHP on/off cycling period (kW)		
T	Temperature (°C)	$\bar{T}$	Upper bound of the temperature (°C)
$\underline{T}$	Lower bound of the temperature (°C)		
T'	Temperature after introducing dynamic effects (°C)		
ΔT_w	Width of the CHWR temperature control band (°C)		
t	Time (s)	UA	Overall heat transfer coefficient (kW/K)
V	Volume (m ³)		
$\hat{W}$	Mean power consumption in one GSHP on/off cycling period (kW)		

Greek letters

ρ_w Water density (kg/m³)

Subscripts

a	Air	cd	Condenser
cl	Cooling water	cyc	One cycle time
down	Lower threshold	ev	Evaporator
fan	Fan	GS	Ground source heat exchanger
HP	Heat pump	in	Inside or inlet
<i>k</i>	Current time instant	<i>l</i>	Total cooling load
min	Minimum	off	Off
on	On	out	Outlet
pump	Pump	rated	Rated
re	Return	<i>sen</i>	Sensible heat load
set	Set value	soil	Soil
sup	Supply	sys	System
up	Upper threshold	w	Chilled water

1. Introduction

Air-conditioning (A/C) systems are widely used in buildings to provide safe and comfortable environment for building occupants. It has been shown that more than 40% of the building energy consumptions are mainly used for the heating, ventilation, and air-conditioning (HVAC) systems (ASHRAE 2009; Shen and Lukes 2015). In order to reduce the energy use of A/C systems, a number of optimization controls have been developed since 1980s, reviewed by ASHRAE handbook (2011). New developments since 2000 were surveyed by Afram and Janabi-Sharifi (2014). Generally, the basic components of optimizing A/C system framework are cost function, decision variables, and constraints (Huang and Wang et al. 2009; Sun et al. 2013).

Many optimization methods, including evolutionary algorithms (Sayyaadi et al. 2009, Kusiak et al. 2011), branch and bound (Chang et al. 2005, Ferreira et al. 2012) and simulated annealing (Chang et al. 2006, Spindler and Norford 2009), have been shown to be successful in minimizing energy consumption and reducing operation cost for A/C systems. However, those optimization methods focus on multi-chiller plants with large capacity (Sun et al. 2013, Ling et al. 2012) or on commercial refrigeration plants (Sarabia et al 2007). Such systems are typically characterized by continuous capacity control, for example using optimal chiller sequencing control.

This paper studies the optimization of a small-scale A/C system integrated with a ground source heat pump (GSHP). Usually, a GSHP refers to a heat pump that uses the ground as a heat source and provides heat in cold winter, but it also refers to a heat pump that uses the ground as a heat sink and provides cooling in hot summer (Corberan et al. 2011). In this study, the cooling condition of the GSHP is concerned. Such a system is one common and fast

growing system for heating/cooling small-scale buildings (Arteconi et al. 2013; Schibuola et al. 2013). Different from chiller plants with large capacity, the most common control method of small-scale GSHPs is an on/off control (Montagud et al. 2014). Many studies show that the annual efficiency of the intermittently operated heat pump may be higher than the variable-speed controlled heat pump (Karlsson and Fahlen 2007; Madani et al. 2011; Zhao et al. 2003). The on/off controlled GSHP is usually controlled according to the chilled water return (CHWR) temperature (i.e., the water temperature at the inlet of the GSHP evaporator) (Corberan et al. 2011). When the CHWR temperature is higher than a pre-defined upper threshold, the compressor will be switched on; while it will be switched off when the CHWR temperature is lower than a pre-defined lower threshold. Generally, the on/off threshold is set according to the set-point of the CHWR temperature $T_{w,set}$ and the allowed width of the differential between the on and off temperature ΔT_w (see Eq. (1) for details) (Beghi and Cecchinato 2011).

There are two important issues in the on/off control of GSHP in practice. One is the GSHP start-ups number per hour, which should be concerned since frequent switch on/off will speed up the wear and aging of the device (Beghi and Cecchinato 2011). Another one is the significant disturbances introduced by the on/off control on the control loops of the A/C system, which may deteriorate the stability of the control loops especially when the on/off switch is frequent (Corberan et al. 2011). Therefore, both issues should be taken into account when studying the energy efficiency of a GSHP integrated A/C system.

However, current relative studies did not consider both issues. For example, Corberan et al. (2011) studied the on/off control of a GSHP integrated A/C system and showed that the energy consumption of the GSHP reduces when the set-point of the CHWR temperature

increases. Madani et al. (2013) compared three common methods to control an on/off controlled GSHP system and showed that the degree-minute method achieves the lowest energy use. Cervera-Vázquez et al. (2015) proposed an energy optimization strategy for a GSHP system and showed that a higher chilled water temperature set-point leads to more frequent on/off of the heat pump (Beghi and Cecchinato 2011). As the compressor on/off cycling frequency depends on the on/off threshold, Beghi and Cecchinato (2011) proposed an adaptive control to deal with the similar problem. In their study, ΔT_w and $T_{w,set}$ are regulated to improve the energy efficiency rating (EER) of the chiller based on the estimated cooling load subject to the allowed maximum number of compressor start-ups per hour. However, in their algorithm, the energy consumption of the air side system was not considered.

In this paper, a strategy is proposed for a GSHP-integrated A/C system to minimize the overall system energy consumption subject to the allowed GSHP maximum start-ups number per hour specified by customers. Two steps are necessitated. Firstly, the water volume inside the chilled water loop is checked whether it is enough to satisfy the allowed GSHP maximum start-ups numbers per hour at the designed condition (i.e. ΔT_w and $T_{w,set}$ are the rated values). If not, a water tank is added to increase the water volume to the required minimum volume. Secondly, ΔT_w and $T_{w,set}$ are optimized to minimize the overall system energy consumption according to the system operating load condition without violating the allowed GSHP maximum start-ups number and degrading the indoor temperature control performance.

2. System Description and Control Issues

2.1 System description

The GSHP-integrated central A/C system is illustrated in Fig. 1, where the GSHP is controlled by an on/off capacity controller C1. The cooling water is provided by a ground

source heat exchanger (GSHE). The water flow rate in both the chilled water side and the cooling water side is constant. The supply air is conditioned by an air handle unit (AHU), which consists of the fresh air from outdoor and the returned air from the conditioned space. When the GSHP is on, the water is chilled down and delivered to the AHU to cool down the supply air. The supply air temperature is not under control and will vary with the variation of the chilled water supply temperature, which oscillates following the on/off operation of the GSHP. The supply air flow rate is controlled by a variable frequency drive (VFD) fan according to the room temperature measurement and its set point. The room is a common A/C zone, and its temperature is controlled by a bilinear feedback controller C2 (Huang 2011), which tracks the set-point by regulating the supply air flow rate. Details are referred to the reference (Gao et al. 2015).

Fig. 1 Schematic diagram of the GSHP integrated central air-conditioning system

2.2 Water volume and its influence on system performance

It is known that the water volume inside the chilled water loop affects the on/off frequency of the compressor when the system operates at partial load. Generally at the same load condition, the on/off frequency will be higher when the water volume is less. Normally, the water volume is designed according to the cooling capacity of the system. Its design range is from about 3.9 to 7.8 litres per kilowatt in a typical HVAC system or up to 7.8 to 12.9 litres per kilowatt where temperature control accuracy is critical (Dawson 2007). This design does not

consider the allowed maximum start-ups number of compressor per hour that may be specified by customers to protect their device (e.g. three or two times per hour). The specification of the allowed maximum start-ups number poses a constraint (a minimum value) on the water volume inside the chilled water loop. If the water volume is smaller than the required minimum water volume, a water tank is needed for increasing the water volume inside the chilled water loop to guarantee the allowed maximum start-ups number of compressor per hour specified by customers. It should be noted that when a water tank is added, it is fully integrated into the existing system without additional water volume control.

2.3 Importance of the setting for the on-off capacity control of the GSHP

When the water volume inside the chilled water loop is fixed, the performance of the system is mainly affected by the system control settings. In the on/off capacity control of a GSHP, the set-point of the CHWR temperature $T_{w,set}$ and the width of the CHWR temperature control band ΔT_w are normally used to determine the upper threshold (i.e. when the CHWR is higher than $T_{w,up}$, the GSHP is switched on) and the lower threshold (i.e. when the CHWR is lower than $T_{w,down}$, the GSHP is switched off) as described in Eq. (1) (Beghi and Cecchinato 2011).

$$T_{w,up} = T_{w,set} + \frac{\Delta T_w}{2}, \quad T_{w,down} = T_{w,set} - \frac{\Delta T_w}{2} \quad (1)$$

The set of $T_{w,set}$ will affect the whole system energy and the number of the GSHP start-ups per hour. When ΔT_w is a constant, a higher $T_{w,set}$ increases the cooling capacity and reduces the on time operation of the heat pump (Corberan et al. 2011), which can be exploited to reduce the heat pump energy use (Beghi and Cecchinato 2011). However, the mean supply air temperature increases following the increment of $T_{w,set}$. In order to maintain the space air temperature close to its set-point, the average operation frequency of the fan has to be enlarged to increase the supply air flow rate. Thus, the energy consumption of the fan

increases. Therefore $T_{w,set}$ should be set by considering the overall system energy consumption. Besides, given a load the value of $T_{w,set}$ will also affect the frequency of the GSHP switched on/off. A larger $T_{w,set}$ will result in a larger start-ups number of the GSHP per hour. This is because for a given load the on time duration of the heat pump reduces with the increase of $T_{w,set}$ while the off-time is relatively constant, and the period for one on/off cycle is reduced.

The set of ΔT_w will also affect the number of the GSHP start-ups per hour. Given a $T_{w,set}$ and the load, ΔT_w will affect the oscillating period of the CHWR temperature and hence the start-ups number of the GSHP (Beghi and Cecchinato 2011). A larger ΔT_w value leads to a smaller number of the GSHP start-ups.

Fig. 2 Space temperature control performance with different settings of $T_{w,set}$

It is worth noticing that $T_{w,set}$ and ΔT_w will also affect the terminal room temperature control. An example from a test on the test facility shown in Section 4.1 is given in Fig. 2, where the load was fixed and ΔT_w was set to 6 °C and maintained a constant. When $T_{w,set}$ was 14 °C, the space air temperature can be maintained at the set-point very well; while it fluctuated significantly when $T_{w,set}$ was changed to 17 °C. Therefore, in order to improve the whole system energy performance, reduce the on/off cycling and enhance the room temperature

control performance, $T_{w,set}$ and ΔT_w should be optimized, and an optimization scheme will be discussed in the next section.

3. The Proposed Optimization Strategy

Given a GSHP integrated A/C system and the specified allowed maximum start-ups number of the GSHP per hour by customers, the proposed optimization scheme is briefly illustrated below:

- Firstly, the water volume inside the chilled water loop will be checked whether it is enough to satisfy the allowed GSHP maximum start-ups number per hour specified by customers when the rated $T_{w,set}$ and ΔT_w are used. To this end, the required minimum water volume $V_{w,sys,min}$ will be calculated according to the allowed GSHP maximum start-ups number and compared with the actual water volume V_{pip} . If the actual value V_{pip} is smaller than the required volume $V_{w,sys,min}$, a water tank will be used to increase the water volume to the required minimum water volume.
- Secondly, $T_{w,set}$ and ΔT_w will be optimized online to minimize the overall system energy use according to the system operating load condition. To this end, the system energy model will be established, and an efficient way to solve the optimization will be identified. The GSHP maximum start-ups per hour specified by customers and the room temperature control performance will be taken as constraints in the optimization.

3.1 Minimum chilled water volume of the system

For a single GSHP with the on/off capacity control, the maximum number of the GSHP start-ups occurs when the cooling load is 50 % of the GSHP rated cooling capacity (Beghi and Cecchinato 2011) (which can be derived from Eq.(8) as well), based on which the minimum chilled water volume is given by Eq. (2), where $t_{cyc,min}$ (in second) is the

minimal on/off cycle time corresponding to the allowed GSHP maximum start-ups N_{start} per hour, defined by Eq. (3).

$$V_{w,\text{sysmin}} = \frac{t_{\text{cyc,min}} Q_{\text{HP, rated}}}{4\Delta T_{w,\text{rated}} c_{pw} \rho_w} \quad (2)$$

$$t_{\text{cyc,min}} = \frac{3600}{N_{\text{start}}} \quad (3)$$

Equation (2) is developed as follows. Firstly, according the energy balance of the chilled water, the CHWR temperature satisfies (Beghi and Cecchinato 2011)

$$C_w \frac{dT_{w,\text{re}}}{dt} = q_l - Q_{\text{HP}}, \quad C_w = c_{pw} \rho_w V_{w,\text{sys}} \quad (4)$$

where C_w is the thermal capacitance of the water inside the chilled water system (kJ/K), c_{pw} is the water specific heat (kJ/(kg·K)), ρ_w is the water density (kg/m³), $V_{w,\text{sys}}$ is the water volume of the chilled water system (m³), q_l is the total cooling load of the zone (kW), and Q_{HP} is the cooling amount of the heat pump (kW).

Using a sampling interval h , Eq. (4) is discretized as Eq. (5), where k indicates the time instant. For a given total cooling load q_l , the on half-cycle t_{on} , i.e. the time used for the CHWR temperature dropping from its upper threshold to its lower threshold, is calculated by Eq. (6). When the GSHP is switched off, $Q_{\text{HP}}=0$ and the off half-cycle t_{off} is calculated by Eq. (7), where $\hat{Q}_{\text{HP}}$ is the mean cooling capacity over the on time during on/off cycling period, $Q_{\text{HP, rated}}$ is the rated cooling capacity of the GSHP at the designed condition, and PLR is the partial load ratio.

$$\frac{T_{w,\text{re}}^{k+1} - T_{w,\text{re}}^k}{h} = \frac{q_l^k - Q_{\text{HP}}^k}{C_w} \quad (5)$$

$$t_{\text{on}} = \frac{\Delta T_w C_w}{\hat{Q}_{\text{HP}} - Q_{\text{HP, rated}} PLR}, \quad PLR = \frac{q_l}{Q_{\text{HP, rated}}} \quad (6)$$

$$t_{\text{off}} = \frac{\Delta T_w C_w}{Q_{\text{HP,rated}} PLR} \quad (7)$$

As $T_{w,set}$ and ΔT_w are at their rated values, the GSHP will output its rated cooling capacity when it is on (i.e. $\hat{Q}_{HP} = Q_{HP,rated}$), the full cycle is given by Eq. (8). When $PLR=0.5$, the shortest cycle is reached as shown by Eq. (9), based on which the minimum water volume inside chilled water loop can be calculated by Eq. (2).

$$t_{\text{cyc}} = t_{\text{on}} + t_{\text{off}} = \frac{\Delta T_{\text{w, rated}} C_{\text{w}}}{Q_{\text{HP, rated}}} \left(\frac{1}{1 - PLR} + \frac{1}{PLR} \right) \geq \frac{2\Delta T_{\text{w, rated}} C_{\text{w}}}{Q_{\text{HP, rated}}} \sqrt{\frac{1}{(1 - PLR)PLR}} \quad (8)$$

$$t_{\text{cyc}} = t_{\text{cyc,min}} = \frac{4\Delta T_{\text{w,rated}} C_{\text{w}}}{Q_{\text{HP,rated}}} \quad (9)$$

If $V_{w,sys,min} > V_{pip}$, a water tank is needed and the system structure becomes Fig. 3. The minimum size of the water tank for the system can be computed by Eq. (10). When $T_{w,set}$ and ΔT_w are the rated values, this size of the water tank can satisfy the allowed maximum start-ups number of the compressor at any cooling load condition.

$$V_{w,tank} = V_{w,sys,min} - V_{pip} \quad (10)$$

Fig. 3 Schematic diagram of optimization

3.2 Optimization of $T_{w,set}$ and ΔT_w

After the water volume in the chilled water loop is checked and retrofitted (if necessary), the

next step is to optimize $T_{w,set}$ and ΔT_w . The objective of this optimization is to minimize the overall energy consumption of the system subject to the constraints (1) the GSHP maximum start-ups per hour specified by customers and (2) the allowed variation of the chilled water return temperature. Thus, the optimization is formulated as

$$(T_{w,set}^*, \Delta T_w^*) = \arg \min J(T_{w,set}, \Delta T_w) \quad (11)$$

subject to the constraints (12) - (16):

$$T_{w,set} + \Delta T_w / 2 \leq \bar{T}_{w,re} \quad (12)$$

$$T_{w,set} - \Delta T_w / 2 \geq \underline{T}_{w,re} \quad (13)$$

$$t_{cyc} = t_{on} + t_{off} \geq t_{cyc,min} \quad (14)$$

$$t_{on} \geq t_{on,min} \quad (15)$$

$$t_{off} \geq t_{off,min} \quad (16)$$

where J is the objective function; $t_{on,min}$ and $t_{off,min}$ are the allowed minimum on time and the allowed minimum off time which are given by the compressor manufacturer to safeguard the compressor; $\bar{T}_{w,re}$ and $\underline{T}_{w,re}$ will be specified according to the room temperature control, detailed in the subsection ‘constraints of optimization’.

Objective function

The average power consumption of the overall system in one GSHP on/off cycle is used as the objective function, which is defined as

$$J = \widehat{W}_{sys} = \widehat{W}_{HP} + \widehat{W}_{pump} + \widehat{W}_{fan} \quad (17)$$

where $\widehat{W}_{sys}$ is the average power of the overall system in one cycle (kW). Since the power is correlated to the CHWR temperature set-point, the space air temperature set-point and the PLR, $\widehat{W}_{sys}$ is calculated using the following empirical formula (titled as the GSHP mean

power model in one on/off cycle).

$$\hat{W}_{\text{sys}} = c_0(T_{\text{a,set}} - T_{\text{w,set}})^2 + c_1(T_{\text{a,set}} - T_{\text{w,set}})PLR + c_2\left(\frac{PLR}{T_{\text{a,set}} - T_{\text{w,set}}}\right)^2 + c_3 \quad (18)$$

where $T_{\text{a,set}}$ is the space air temperature set-point, which is fixed in this study; $T_{\text{w,set}}$ is the CHWR temperature set-point; and c_0 - c_3 are coefficients that should be identified using the operation data.

Constraints of optimization

The constraints shown by Eqs. (12) and (13) are developed based on the assumption that when the GSHP is on, the cooling provided by the AHU can fulfil the required cooling demand (only sensible load is concerned) and local controller can well track the room temperature set-point. In this case, the supply air temperature satisfies

$$M_a c_a (T_{\text{a,set}} - T_{\text{a,sup}}) = q_{\text{sen}} \quad (19)$$

The room temperature can be maintained at its set point by adjusting the supply air flow rate. As the supply air flow rate is limited into the range $(\underline{M}_a, \bar{M}_a)$, the supply air temperature should also be limited into the range of

$$\underline{T}_{\text{a,sup}} \leq T_{\text{a,sup}} \leq \bar{T}_{\text{a,sup}} \quad (20)$$

where the lower and upper bounds are given by

$$\underline{T}_{\text{a,sup}} = T_{\text{a,set}} - q_{\text{sen}} / (\underline{M}_a c_a) \quad (21)$$

$$\bar{T}_{\text{a,sup}} = T_{\text{a,set}} - q_{\text{sen}} / (\bar{M}_a c_a) \quad (22)$$

As the supply air temperature is affected by the CHWR temperature, an empirical formula (i.e., the simplified AHU model) is used to describe the relationship between the CHWR temperature and the AHU supply air temperature (Ma and Wang 2009), shown below

$$T_{\text{w,re}} = a_0 T_{\text{a,sup}}^{a_1} q_l^{a_2} M_a^{a_3} \quad (23)$$

where a_0 - a_3 are the parameters that should be identified by the system operating data. Then,

$\bar{T}_{w,re}$ and $\underline{T}_{w,re}$ are calculated by

$$\bar{T}_{w,re} = a_0 \bar{T}_{a,sup}^{a_1} q_l^{a_2} \bar{M}_a^{a_3} \quad (24)$$

$$\underline{T}_{w,re} = a_0 \underline{T}_{a,sup}^{a_1} q_l^{a_2} \underline{M}_a^{a_3} \quad (25)$$

where $\bar{T}_{a,sup}$ and $\underline{T}_{a,sup}$ are the upper and lower bound of the supply air temperature catering to the sensible heat load.

The constraint specified by Eq. (14) is developed to limit the start-ups number of the GSHP. The constraints specified by Eqs. (15) and (16) are used to safeguard the compressor, which are given by the compressor manufacturer. When $T_{w,set}$ changes, the mean cooling capacity over the on time during on/off cycling period, $\hat{Q}_{HP}$, will change, which affects the length of the heat pump operating time as shown in Eq. (6). Therefore, the relationship between $\hat{Q}_{HP}$ and $T_{w,set}$ is important for the constraints of Eqs. (14)-(16).

In this study, the heat accumulation in the soil is ignored in a short time period, and the thermal process of the GSHE is assumed to be constant. Thus the cooling water temperature at the GSHE outlet in the GSHP on period is relatively steady, and the variation may be within 0.5 °C. The effect of the cooling water temperature on $\hat{Q}_{HP}$ is not obvious and may be neglected. $\hat{Q}_{HP}$ is seriously affected by $T_{w,set}$, and can be predicted by using the model of Eq. (26) (called as the mean cooling capacity model over the on time during on/off cycling period).

$$\hat{Q}_{HP} = b_0 T_{w,set}^{b_1} \quad (26)$$

3.3 Load estimation

As shown in Eqs. (4), (18), (19) and (23), $T_{w,set}$ and ΔT_w are optimized according to the load

condition. Since the optimization works based on the GSHP on/off cycling period, the average cooling load in one cycle calculated by Eq. (27) is used to assess the load condition, where $\hat{q}_l$ indicates the mean value of the total cooling load in one cycle, $\hat{q}_{sen}$ indicates the mean value of the sensible heat load in one cycle, and N_c is the samplings in one cycle and its value is determined by the length of the cycle period.

$$\hat{q}_l = \frac{1}{N_c} \left(\sum_{i=0}^{N_c-1} q_{l,k-i} \right), \quad \hat{q}_{sen} = \frac{1}{N_c} \left(\sum_{i=0}^{N_c-1} q_{sen,k-i} \right) \quad (27)$$

The real-time total cooling load $q_{l,k}$ is calculated by Eq. (28) based on the energy balance between the water side and the air side of the AHU. This simple way is used based on the assumption that the thermal loads affecting the GSHP system is only produced by AHU.

$$q_{l,k} = M_w c_{pw} (T_{w, re, k} - T_{w, sup, k}) \quad (28)$$

where M_w is the chilled water flow rate which is constant, $T_{w, sup}$ is the water temperature at the inlet of the AHU (i.e. the chilled water supply temperature). $T_{w, re}$ is the CHWR temperature assumed to be the water temperature measured at the outlet of the AHU. The subscript k indicates the current time instant. Similarly, one simple way to calculate the real-time sensible heat load $q_{sen,k}$ at steady state is used based on the circulating air energy balance as shown in Eq. (29), which will be used in Eq. (27) and then in Eqs. (21-22).

$$q_{sen,k} = c_a M_{a,k} (T_{a, re, k} - T_{a, sup, k}) \quad (29)$$

3.4 Optimization algorithm

The optimization algorithm is summarized below:

Offline

Initialize the counter N_c to zero, and set the initial values of $\hat{q}_l$ and $\hat{q}_{sen}$.

Online operation (at each sampling time)

Step 1: Measure $T_{a, re, k}$, $T_{a, sup, k}$, $m_{a, k}$, $T_{w, re, k}$, $T_{w, sup, k}$ and calculate the real-time total cooling load

$q_{l, k}$ by Eq. (28) and sensible heat load $q_{sen, k}$ by Eq. (29), and increase N_c by 1;

Step 2: Check the status of the GSHP:

If the status changes from ‘off’ to ‘on’, calculate the mean total cooling load $\hat{q}_l$ and the mean sensible heat load $\hat{q}_{sen}$ by Eq. (27), and reset N_c to zero. Calculate the optimal values of $T_{w, set}^*$ and ΔT_w^* by Eq. (11);

If the status does not change or the status changes ‘on’ to ‘off’, maintain the values of $\hat{q}_l$, $\hat{q}_{sen}$, N_c , $T_{w, set}^*$ and ΔT_w^* ;

Step 3: Apply $T_{w, set}^*$ and ΔT_w^* to the on/off controller and wait for the next sampling time.

For this optimization algorithm, the inputs include the cooling load calculated by Eq. (28) and the sensible heat load calculated by Eq. (29). The outputs are $T_{w, set}$ and ΔT_w , which will be used to control the on/off of the GSHP by resetting the on/off threshold according to Eq. (1).

4. Construction of the simulation platform

4.1 Test facility

A simulated virtual system representing an actual GSHP-integrated central A/C system, as shown in Fig. 4, was used to test the performance of the optimization strategy. The air-conditioning room has the area of 50 m², which may represent an ordinary A/C zone. The load conditions of the room are presented in Section 5.1. The technique data of the GSHP-integrated A/C system are given in Table 1. According to the rated CHWR temperature set-point and the rated band width of the CHWR temperature control band, the GSHP will be switched on when the CHWR is larger than 15 °C while off when the CHWR is smaller than 9 °C. The supply air fan is equipped with a variable frequency drive, and the operation frequency can be adjusted. This test rig is also installed with a set of monitoring

and control system. The parameters of sensors and actuators are listed in Table 2. By this building automation system (BAS), chilled/cooling water supply/return temperature and air supply/return temperature can be measured and recorded automatically with specified time interval.

Fig. 4 Layout of the GSHP integrated A/C system

Table 1 Technique data of the GSHP-integrated A/C system

GSHP	
Compressor type	Scroll compressor
Evaporator, condenser type	Tube-in-tube heat exchangers
Expansion valve	A common thermostatic expansion valve
Refrigerant	R22
Rated cooling capacity	17.2 kW
Rated input power	3.91 kW
Rated CHWR temperature set-point	12 °C
Rated band width of the CHWR temperature control band	±3 °C

Rated chilled water flow	0.822 kg/s
Rated cooling water flow	1.006 kg/s
Chilled water pump rated power	0.292 kW
Cooling water pump rated power	0.475 kW
Air-handling unit	
Rated air flow rate	3000 m ³ /h
Supply air fan rated power	1.1 kW
Supply air fan frequency	15 Hz to 50 Hz

Table 2 Description of sensor and actuator

Sensor/Actuator	Signal type	Range	Accuracy
Air temperature sensor	10K ohm	0-70°C	±0.1°C
Water temperature sensor	4-20mA	0-50°C	±0.1°C
Fan frequency drive	0-10V	0-50Hz	-

4.2 Simulation platform

The software TRNSYS (TRNSYS 2004) was used to construct the simulation platform as shown in Fig. 5. The models used in this simulation platform include a building model, a GSHP model, an AHU model, water pump models, a water tank model, a ground source heat exchanger (GSHE) model, a cooling coil control valve model, temperature and humidity sensor models and an actuator model etc. The optimization algorithm was coded using MATLAB, which are embedded in TRNSYS by the component Type 155. Two local controllers, an on/off controller and a bilinear feedback controller, were used control the heat pump and the room temperature respectively.

Fig. 5 Simulation platform of the GSHP integrated A/C system

4.3 Main models used in the simulation platform

Building model

The building model adopted in this study is a simplified model simulating the dynamic balance of energy and moisture of the building system. The model represents the A/C room using a thermal network with thermal resistance, thermal capacitance and air volume. The external and internal walls of the room are represented by a two-node model to link the space with outside or the neighbor rooms. The internal mass such as furniture in the room is represented by another two-node model. Detailed model description and the building configuration can be found in Ref (Gao et al. 2015).

Heat pump dynamic model

The GSHP dynamic model includes the steady-state model (includes the motor power consumption model and the evaporator and condenser heat exchanger model) and the first-order inertia process in the evaporator and condenser. The steady-state model is a

quasi-physical model (Bourdhouxhe et al. 1999). The power of the compressor W_{com} is calculated by Eqn. (30), where W_{in} is the internal work of the compressor, W_{lo} is a constant portion of the electromechanical losses, and α is a loss factor allowing for the definition another electromechanical loss which is assumed to be proportional to the internal power. Both the evaporator and the condenser were modeled using the ε -NTU method, described by Eqns. (31)-(34), where Q is the heat exchange amount, and UA is the overall heat transfer coefficient, which is considered to be constant.

$$W_{\text{com}} = W_{\text{in}} + \alpha W_{\text{in}} + W_{\text{lo}} \quad (30)$$

$$\varepsilon_{\text{ev}} = 1 - \exp(-NTU_{\text{ev}}), \quad NTU_{\text{ev}} = \frac{UA_{\text{ev}}}{c_{pw}M_w} \quad (31)$$

$$Q_{\text{ev}} = \varepsilon_{\text{ev}} c_{pw} M_w (T_{w,\text{in}} - T_{\text{ev}}) \quad (32)$$

$$\varepsilon_{\text{cd}} = 1 - \exp(-NTU_{\text{cd}}), \quad NTU_{\text{cd}} = \frac{UA_{\text{cd}}}{c_{pw}M_{\text{cl}}} \quad (33)$$

$$Q_{\text{cd}} = \varepsilon_{\text{cd}} c_{pw} M_{\text{cl}} (T_{\text{cd}} - T_{\text{cl},\text{in}}) \quad (34)$$

The first-order inertia model in the Ref. (Wang et al. 2000) is used to simply represent the dynamic response of the GSHP with lumped thermal mass, shown by Eqs. (35)-(38), where T and T' are the water temperature at inlet and outlet of the thermal storage, respectively. C is the thermal capacitance, and only the thermal capacitances of the water volume in the evaporator and condenser are concerned.

$$C_{\text{ev},\text{out}} \frac{dT'_{w,\text{out}}}{dt} = c_{pw} M_w (T_{w,\text{out}} - T'_{w,\text{out}}) \quad (35)$$

$$C_{\text{cd},\text{out}} \frac{dT'_{\text{cl},\text{out}}}{dt} = c_{pw} M_{\text{cl}} (T_{\text{cl},\text{out}} - T'_{\text{cl},\text{out}}) \quad (36)$$

$$C_{\text{ev},\text{in}} \frac{dT'_{w,\text{in}}}{dt} = c_{pw} M_w (T'_{w,\text{in}} - T_{w,\text{in}}) \quad (37)$$

$$C_{cd,in} \frac{dT'_{cl,in}}{dt} = c_{pw} M_{cl} (T'_{cl,in} - T_{cl,in}) \quad (38)$$

GSHE model

The GSHE associated with the GSHP system is modeled by using a NTU model as Eq. (39), where T_{soil} is the soil temperature, and UA_{GS} is the overall heat transfer coefficient of the GSHE which is considered to be a constant. Since the operation of the air-conditioning system is not continuously, the hot stack is not considered in this GSHE model.

$$T_{GS,out} = T_{soil} + \frac{T_{GS,in} - T_{soil}}{\exp^{NTU_{GS}}}, NTU_{GS} = \frac{UA_{GS}}{c_{pw} M_{cl}} \quad (39)$$

AHU model

The adopted AHU model consists of a wet cooling coil model and a fan model (Wang 1999). The flow characteristic of the cooling coil valve is of equal percentage with dead bands (Xu et al. 2004). The basic law of fluid mechanics is used for the fan model (Gao et al. 2015). Dynamic sensor models are used to simulate the temperature, humidity sensors using time constant method. An actuator model is used to simulate the realistic characteristics of the actuators. The actuator is assumed to accelerate very quickly and then turn at constant speed (Haves and Dexter 1989). The water pump model is a constant speed pump model. A water tank provides thermal inertia within the system, and a simplified tank model from Ref. (Corberan et al. 2011) is used as shown by Eq. (40), where ρ_w is the water density. The temperature in the water tank $T_{w,tank}$ is considered to be uniform, and the heat loss from the tank is ignored.

$$\frac{dT_{w,tank}}{dt} = \frac{M_w (T_{w,out} - T_{w,tank})}{V_{w,tank} \rho_w} \quad (40)$$

4.4 Validation of the heat pump model and the GSHE model

Before testing the control algorithm, the heat pump model and the GSHE model were validated using the operation data collected from the test rig. To this end, the test rig was running in several summer days and the data collected in July 7 were used to identify the parameters of the heat pump model and the GSHE model. The outdoor temperature varied from 23°C to 32 °C and the relative humidity varied from 66% to 95% (a typical summer day in Wuhan). The indoor load varied from 5 kW (night) to 15 kW.

In the heat pump model, the input variables are the inlet chilled/cooling water temperature as well as the chilled/cooling water flow rate. The output variables are the outlet chilled/cooling water temperature and the power consumption. In the steady-state model, the parameters need to be determined are α , W_{lo} , UA_{ev} and UA_{cd} , and their identified values are listed in Table 3. For the dynamic model (Eqs. (35)-(38)), as the water volume in the evaporator was about 15 L and that in the condenser was about 20 L, the thermal capacitances (i.e., $C_{ev,in}$, $C_{ev,out}$, $C_{cd,in}$ and $C_{cd,out}$) used in the heat pump dynamic model were calculated and listed in Table 4.

Table 3 Parameters of the heat pump

Parameters	UA_{cd} (kW/K)	UA_{ev} (kW/K)	α	W_{lo} (kW)
Value	3.0371	2.6786	-0.3716	2.7762

Table 4 Thermal capacitances used in the heat pump dynamic model

Thermal capacitance	Inlet (kJ/K)	Outlet (kJ/K)
Evaporator side	31.5	31.5
Condenser side	42	42

The model was validated using the data collected in another summer day (other than July 7).

As an example, the measured CHWR temperature and cooling water return temperature, the input variables of the heat pump model in July 9, are plotted in Fig. 6(a). It can be seen that the cooling water temperature at the inlet of the condenser $T_{cl,in}$ was stable (around 22 °C). The CHWR temperature varied between 11 °C and 19 °C with the on/off of the GSHP, where the CHWR temperature set-point $T_{w,set}$ was set as 15 °C, and ΔT_w as 6 °C (Ideally $T_{w,in}$ should vary between 12 °C and 18 °C; and the slightly difference may due to inertia in the system dynamics). In Fig. 6(b), the predicted power consumption as well as the predicted chilled water supply temperature by using the GSHP model was compared with the measurements from the test rig. The maximum mean deviation of the compressor power consumption in one cycle is 2.46 %, and the maximum mean deviation of the chilled water temperature at the evaporator outlet is 4.45 %. The GSHP model may predict the outlet water temperature and power consumption accurately.

(a) Input variables of the GSHP model

(b) Comparisons of experimental data and model predicted data

Fig. 6 Evaluation of the GSHP model

Note: “ W_{com} ” indicates the power consumption, and “ $T_{w,out}$ ” indicates the chilled water temperature at the outlet of the evaporator. “-exp and -pre” indicate the experimental results and predicted results by using the model respectively. “-MD” indicates the mean deviation.

For the GSHE model, UA_{GS} is the only parameter that needs to be identified. As the annual average temperature of the soil under the ground of about 10 meters maintains in Wuhan is around 18 °C (Yang et.al 2014), T_{soil} was set to be 18 °C. Using the operating data of the GSHE in July 7, UA_{GS} was identified as $UA_{GS}=2670$ W/K. Once again, the operating data in other summer days were used for validation. An example is shown in Fig 7 where the data were collected in July 9. It can be seen that the predicted water temperature matches well with the measurements. The relative error varied between -5 % and 5 %. It also shows that the GSHE model may predict the outlet cooling water temperature accurately.

Fig. 7 Evaluation of the GSHE model

5. Results and Analysis

After the heat pump model and the GSHE model were validated, the simulation platform was used as an ‘actual system’ to test the optimization algorithm. The results and analysis were detailed below.

5.1 Boundary conditions for the results

In the simulation tests, occupancy, lighting and equipment loads and weather conditions were specified according to the actual operation conditions of the test facility shown in Fig. 4. The profile of total heat gain is shown in Fig. 8, in which from the lunch time was set from 12:00 to 14:00. The outdoor air temperature and humidity in a summer day and a spring day were used and shown in Fig. 9(a) and 9(b) respectively. The occupancy number was set to 20, and the outdoor air flow rate was designed to be 30 m³/h for each occupancy according to the ASHRAE standard (ASHRAE 2004). The cooling loads (including total cooling load and sensible heat load) of the summer day and the spring day are given in Fig. 10(a) and 10(b) respectively. The set-point of the indoor air temperature was set 25 °C. The air-conditioning worked from 8:00 am to 20:00 pm, and the sampling interval for the local control algorithms

was 15 seconds. The boundary conditions for each test were the same.

Fig. 8 Profile of the total heat gain of the occupancy, lighting and equipment

(a) A summer day

(b) A spring day

Fig. 9 Weather profiles of a summer day and a spring day

(a) A summer day

(b) A spring day

Fig. 10 Cooling load profiles of the summer day and the spring day

5.2 Identification of the models used in the optimization algorithm

Before validating the models used in the optimization algorithm, the minimal water volume inside the chilled water loop was firstly investigated. As the allowed maximum number of the GSHP start-ups per hour was two times and both the minimum values of t_{on} and t_{off} were 180 seconds (given by the manufacture), the minimum size of the water tank for the designed operation condition was 0.250 m^3 . Therefore, a water tank with the capacity of 0.250 m^3 was added to the chilled water flow loop.

As the optimization algorithm needs a mean power model of GSHP in one on/off cycling period in the form of Eq. (18), an AHU model in the form of Eq. (23) and a mean cooling capacity model of the heat pump over the on-time in one on/off cycle in the form of Eq. (26), they were identified and validated using the data generated from the simulation platform.

To identify the GSHP mean power model (i.e. the coefficients c_0 - c_3), the training data were generated by the simulation platform (with the configuration specified in Section 4.1) when the room temperature set point $T_{a,set}$ was fixed at 25 °C and the PLR was varied from 30% to 100% (including both summer and spring days). These four coefficients were identified as $c_0=-0.0008$, $c_1=0.2372$, $c_2=253.4778$, $c_3=1.2957$. Different data were used for validation and the result is shown in Appendix A.1.

To identify the AHU model in the form of Eq. (23) (i.e. the coefficients a_0 - a_3), the total cooling load q_l was changed from the rated load (full load) to 30% of the full load (corresponding to the PLR from 100% to 30%). These four coefficients were identified as $a_0=1.6538$, $a_1=0.8126$, $a_2=0.0242$ and $a_3=0.0608$. Similarly, the validation was done using different data and the result is shown in Appendix A.2.

To identify the mean cooling capacity model of the heat pump over the on-time in one on/off cycle in the form of Eq. (26) (i.e. the coefficients b_0 , b_1), the CHWR temperature set-point $T_{w,set}$ was changed from 12°C to 16 °C. These two coefficients b_0 and b_1 were identified as $b_0=8.7967$, $b_1=0.2541$. Once again, the validation was done using different data and the result is given in Appendix A.3.

5.3 Performance evaluation of the optimization

The optimization algorithm was assessed from the room temperature control performance, GSHP on/off frequency and the total energy consumption. To facilitate the discussion, the proposed optimization strategy was compared with a basic control strategy, in which $T_{w,set}$ and ΔT_w are fixed on their rated values. Note that both strategies were compared when they were applied to the same system.

Space temperature control performance

In the summer day, the space air temperature was plotted in Fig. 11. Both the optimization strategy and the basic control strategy achieved good control robustness. The average absolute tracking error from 8:00 to 20:00 was 0.030 °C, while it was 0.023 °C when the basic control strategy was used. In the optimization strategy, $T_{w,set}$ and ΔT_w were adjusted according to the cooling load in order to minimize the overall energy consumption, and the variation of $T_{w,set}$ and ΔT_w were shown in Fig. 12(a). Note that when the cooling load increased sharply (which can be seen from Fig. 10(a)) at 9:00 and 14:00, the optimization strategy cannot track the change timely so that the large spikes were observed.

Fig. 11 The space air temperature in a summer day

(a) Optimization strategy

(b) Basic control strategy

Fig. 12 The CHWR temperature profile in a summer day

In the spring day, the thermal response of the space air was more sensitive to the interference of the GSHP on/off. Fig. 13 shows the space air temperature in the spring day. It can be seen that the space air temperature control was more robust when the optimization strategy was used. The average absolute tracking error from 8:00 to 20:00 was 0.059 °C, while it was 0.376 °C when the basic control strategy was used. The number of the large spikes by using the optimization was much less than that by using the basic control strategy.

Under the light cooling load condition, the cooling amount delivered to the room was larger than the cooling demand. In this case, the basic control strategy cannot adjust the variation

range of the CHWR temperature (as shown in Fig. 14(b)) to maintain the space air temperature at the set-point well. However, the optimization strategy was able to increase the minimum value of the CHWR temperature to reduce the minimum cooling amount to match the cooling demand as shown in Fig. 14(a). Similarly, when the cooling load decreased sharply as shown Fig. 10(b) at 12:00 and 18:00, the optimization strategy cannot track the change timely, and the large spikes were also observed.

Fig. 13 Control performance of the space air temperature in a spring day

(a) Optimization strategy

(b) Basic control strategy

Fig. 14 CHWR temperature profile at the light load condition

GSHP start-ups number

The start-ups number of the GSHP in the summer day and the spring day were listed in Table 5. In both control strategies, the constraint on the GSHP start-ups number was satisfied. In the summer day, as ΔT_w was reset to a smaller value than the rated value during most of the operation time in the optimization strategy, the start-ups number of the GSHP was larger when compared with the basic control strategy as shown in Fig. 15. In the spring day, ΔT_w was reset to a larger value than its rated value from 15:00 to 19:00 as shown in Fig. 14, which results in a slightly smaller on/off number when compared with the basic control strategy.

Table 5 Start-ups number in 12 hours

Load conditions	Summer day	Spring day
Num-bas	13	24
Num-opt	18	22

Note: “-bas and -opt” indicate the basic control strategy and the optimization strategy respectively.

(a) Optimization strategy

(b) Basic control strategy

Fig. 15 Start-ups number of the GSHP in the summer day

Energy performance

The energy performance of the optimization strategy was also compared with that of the basic control strategy. A typical spring day and summer day were considered, representing 35 % and 80 % of the full load condition respectively. The daily average energy consumption of the system was listed in Table 6. It can be seen that the optimization strategy achieved a better energy performance both in summer day and in spring day. In the summer day, the energy consumption of the GSHP from 8:00 to 20:00 decreased 2.68 kWh (i.e., 6.34 %), while that of the fan increased 1.04 kWh (i.e., 27.59 %). The total energy consumption reduced 1.64 kWh, i.e., 2.97 % of the total energy consumption. In the spring day, the energy consumption of the GSHP from 8:00 to 20:00 decreased 4.31 kWh (i.e., 21.15 %), while that of the fan

increased 1.25 kWh (i.e., 53.42 %). The total energy consumption was reduced 3.06 kWh (i.e., 9.59 % of the total value) when compared with that by using the basic control strategy. The results show that the optimization strategy can efficiently reduce the energy consumption of the overall system, especially under partial load conditions.

Table 6 Energy consumption of 12 hours in the summer day and the spring day

Load conditions	Equipment	Energy consumption (kWh)		Energy saving (%)
		Basic control strategy	Optimization strategy	
Summer day	Heat pump	42.25	39.57	6.34
	Fan	3.77	4.81	-27.59
	Pumps	9.2	9.2	-
	Overall system	55.22	53.58	2.97
Spring day	Heat pump	20.38	16.07	21.15
	Fan	2.34	3.59	-53.42
	Pumps	9.2	9.2	-
	Overall system	31.92	28.86	9.59

Note: “+” indicates energy saving, and “-” indicates energy increasing.

6. Application issues

The proposed strategy is not difficult to use. For an existing GSHP integrated A/C system, the water volume in the chilled water loop can be easily calculated. When the allowed GSHP maximum start-ups number is specified, the required volume is calculated by Eq. (2), which should be compared with the current water volume to decide whether a water tank is needed.

The real-time optimization of $T_{w,set}$ and ΔT_w needs a mean power model of GSHP in one on/off cycling period (Eq. (18)), an AHU model (Eq. (23)) and a mean cooling capacity

model of the heat pump over the on-time in one on/off cycle (Eq. (26)). Those models should be identified using the data collected from commissioning. As the system is in a small-scale, data generation will not be difficult. Another application issue is the online load estimation, which can be done using energy meters based on the calculation shown in Eqs. (28)-(29). Nowadays building automation system (BAS) is widely used to monitor and control A/C systems. A feasible way to implement the proposed strategy is to integrate it into a BAS system.

7. Conclusion

This paper presents an optimization strategy for a GSHP-integrated A/C system in order to optimize the overall system energy consumption without violating the allowed GSHP maximum start-ups number and degrading the space air temperature control performance. This optimization strategy is schemed with two steps. Firstly, the water volume inside the chilled water loop is checked whether it is enough to satisfy the allowed GSHP maximum start-ups number per hour at the design condition. Secondly, an optimization algorithm is developed to optimize the system settings (ΔT_w and $T_{w,set}$) by minimizing the total power of the system. The performance of the optimization strategy is compared with the basic control strategy (i.e., the system operates at the predefined rule). The results show that the optimization strategy can improve the robustness of the space air temperature control significantly and reduce the system energy consumption obviously in spring days, i.e. under light load conditions. The energy consumption of the GSHP was reduced by 21.15 % and the overall system was reduced by 9.59 %. In the summer day, both the optimization strategy and the base control strategy can achieve a robust control of the space air temperature, while the energy consumption of the overall system is slightly smaller (reduced by 2.97 %) when the optimization strategy is used. The results also show that when the water volume in the chilled

water loop is sufficient, both strategies can guarantee the GSHP start-ups number specified by customers being satisfied.

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Appendix A

A.1 Validation of the mean power consumption model of the overall system

Figure A.1 presented the validation of the mean power consumption model of Eq. (18) by using the ‘operation data’ collected from a simulation platform as shown in Fig. 5, which was constructed based on an actual test facility as shown in Fig. 4. The ‘measurement data’ are the power consumptions of the heat pump, pumps and fan. These data were collected in widely varying conditions, which covered the possible operation conditions of the actual system. It can be seen that the model may predict the overall energy consumption of the GSHP system well. The largest absolute prediction error is 2.63 %, and the average absolute error is 0.67 %.

Fig. A.1 Validation of the mean power consumption model of the overall system in one GSHP

on/off cycling period

A.2 Validation of the simplified AHU model

The simplified AHU model, described by Eq. (23), was validated by using the ‘operation data’ collected from the simulation platform. The ‘measurement data’ include CHWR temperature, water temperature at the inlet of the AHU, supply air temperature and supply air flow rate. The validation results were given by Fig. A.2. It can be seen that the model may predict the CHWR temperature accurately in a large variation range. The largest absolute prediction error is 1.01 %, and the average absolute error is 0.42 %.

Fig. A.2 Validation of the simplified AHU model

A.3 Validation the mean cooling capacity model over the on time during on/off cycling period

The mean cooling capacity model of Eq. (26) was validated by using the ‘operation data’ collected from the simulation platform as shown in Fig. A.3. When the CHWR temperature set-point changes from 10 °C to 17 °C, the largest absolute prediction error is 0.18 %, and the average absolute error is 0.1 %. The reliability and accuracy of the model is acceptable in this study.

Fig. A.3 Validation of the mean cooling capacity model

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