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Research Paper

Simulation of the performance of a hybrid ground-coupled heat pump system on the basis of wet bulb temperature control

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HIGHLIGHTS

- A wet bulb temperature control strategy for heat pump system is proposed.
- A method of determining the control parameters is introduced.
- Performance of the new strategy and two common strategies is compared.

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ABSTRACT

Considering the characteristics of a cold source of air and soil, an operation control strategy of a hybrid ground-coupled heat pump system (HGCHPS) was proposed based on wet bulb temperature control. A method of determining the control parameters was introduced, and a mathematical model of the main components of the system was developed. Dynamic simulation of the system performance was studied under this operation strategy and two additional commonly used control strategies. The results showed that the system had good heating and air conditioning safeguards under the control of the wet bulb temperature. The annual average coefficients of performance (*COP*s) of the cooling and heating system were 3.93 and 4.09, respectively. The annual average *COP* of heating and air conditioning was 3.96. The long-running performance of this operation strategy was better than those of the two additional control strategies and was able to maintain the soil's thermal balance around soil heat exchangers (SHEs) with an annual cycle.

1. Introduction

Ground source heat pump system (GSHPS) is an efficient heating and air conditioning system that utilizes surface soil thermal energy to meet the heating-cooling requirements of a building [1]. In areas that are dominated by building cooling loads, a single GSHP system may fail to meet the cooling demand, and in this case, auxiliary cooling devices (such as cooling towers) are required to eliminate the excessive heat of condensation. This form of system is usually called a hybrid ground-coupled heat pump system (HGCHPS). Compared with a single GSHPS, an HGCHPS not only ensures the soil's thermal balance around the soil heat exchangers (SHEs), but also reduces the initial investment and operating costs of the system [2–7].

Because the system can achieve comprehensive utilization of the ambient air and surface soil thermal energy, it is vital important to optimize the system design and operation control to achieve better overall performance and economic efficiency of the system. Researchers from various countries have carried out many studies on such systems [8–15]. The design process and methods were provided firstly by ASHRAE [8], and Kavanaugh and Gilbreath further fine-tuned and improved the process [9,10]. Depending on the connection mode between the SHEs and cooling towers, hybrid heat pump systems are mostly divided into parallel and tandem forms. Based on life-cycle cost analysis, Ramamoorthy et al. and Hackel et al. [11,12] optimized HGCHPS in the tandem form. The capacity of the auxiliary heat abstractor was optimized by Chiasson et al. based on the balance of the cooling and heating loads of soil heat exchangers (SHEs) [13]. The parallel type of HGCHPS was optimized through experiments by Honghee Park et al., who obtained the best values of refrigerant charge and secondary fluid flow rate within SHEs [14]. A cooling tower heat and mass transfer mathematical model was established by Sagia et al., who calculated and analyzed the performance of the cooling tower of a parallel-form hybrid heat pump system at

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Nomenclature

COP	coefficient of performance
COP_c	average cooling coefficient of performance
COP_h	average heating coefficient of performance
COP_s	the annual average coefficients of performance
D_{eq}	equivalent diameter of buried pipe (m)
K_d	cooling tower coefficient
N	number of characteristics of cooling tower
P_{hp}	power consumption of heat pump units (J)
P_{cwp1}	circulation pump power consumption on user side (J)
P_{cwp2}	circulating pump power consumption on the heat source side (J)
P_{cwp3}	circulating pump power consumption on cooling tower (J)
P_{fan1}	power consumption of cooling tower fan (J)
P_{fan2}	power consumption of fan coil (J)
Q_h	heat supply (J)
Q_c	cold supply (J)
Q_{cl}	cumulative cooling load (J)
Q_{hl}	cumulative heat load (J)
Q_{se}	heat extracted from soil (J)
R_0	the thermal resistance of heat exchanger wall (m^2 °C/W)
T_{am}	annual ground surface temperature fluctuations (°C)
T_{ci}	inlet water temperature of condenser (°C)
T_{co}	outlet water temperature of condenser (°C)
T_{ctfi}	inlet water temperature of cooling tower (°C)
T_{ctfo}	outlet water temperature of cooling tower (°C)
T_{ei}	inlet water temperature of evaporator (°C)
T_{eo}	outlet water temperature of evaporator (°C)
T_o	ambient temperature (°C)
T_{os}	ambient wet bulb temperature (°C)
T_{osd}	wet bulb temperature control parameter (°C)
T_s	soil temperature (°C)
T_{s0}	undisturbed soil temperature (°C)
T_{sf}	heat transfer fluid temperature within the soil heat exchanger (°C)
T_{cid}	inlet water temperature control parameter of condenser (°C)
T_{sm}	the annual average temperature of ground surface (°C)
T_{sr1}	nodal temperatures of the first layer of soil (°C)
a_s	thermal conductivity of soil (m^2/s)
c_{ps}	specific heat of soil (J/(kg °C))
c_{pf}	specific heat of heat transfer fluid (J/(kg °C))
c_{psf}	specific heat of exchange fluid within soil heat exchangers (J/(kg °C))
h_a	convection heat transfer coefficient of the surface air (W/(m^2 °C))
h_{sf}	convection heat transfer coefficient of heat transfer fluid (W/(m^2 °C))
i_{ai}	inlet air enthalpy of cooling tower (J/kg)
i_{ao}	outlet air enthalpy of cooling tower (J/kg)
i^*_{ai}	saturated air enthalpy of the cooling tower inlet water temperature (J/kg)
i^*_{ao}	saturated air enthalpy of the cooling tower outlet water temperature (J/kg)
l_0	length of borehole heat exchanger (m)
m_{cta}	air volume of cooling tower (kg/s)
m_{ctw}	water volume of cooling tower (kg/s)
p_c	power consumption in cooling conditions (W)
p_{cd}	rated cooling power consumption of heat pump unit (W)
p_h	power consumption in heat conditions (W)
p_{hd}	rated heating power consumption of heat pump unit (W)
q_c	cooling capacity (W)
q_{cd}	rated cooling capacity of heat pump unit (W)
q_h	heat capacity (W)
q_{hd}	rated heating capacity of heat pump unit (W)
q_l	heat exchange capacity per unit length (W/m)
r	radial coordinate (m)
r_0	borehole heat exchangers' spacing (m)
r_{ct}	the proportion of cooling capacity in Mode 3
t	time (h)
t_c	time constant (h)
t_0	time from initial time of simulation to eventual time of maximum temperature for ground surface (h)
v_{sf}	borehole heat exchangers' inlet velocity (m/s)
z	axial coordinate (m)
z_0	depth (m)
v_s	density of soil (kg/m^3)
ρ_{sf}	density of heat transfer fluid (kg/m^3)
λ_s	thermal conductivity of soil (W/(m^2 °C))
ΔT	temperature difference (°C)
a_1, b_1, c_1, d_1, a_2, b_2, c_2, d_2	curve-fit coefficients

different SHE inlet temperatures [15]. The operation performances of the parallel and tandem forms were compared through experiments by Park et al. [16]. The three most commonly used control methods are borehole outlet water temperature control, temperature difference control between the borehole outlet water temperature and the ambient air wet bulb temperature, and run-time control. The advantages and disadvantages of these three control strategies in different climatic conditions were studied by Yavuzturk et al., who determined the optimal operating control strategy under different system conditions with the lowest total life-cycle cost as the goal [17]. Fan et al. used TRNSYS software to establish a mathematical model of a hybrid heat pump system in parallel form and conducted optimization analysis for the system operation control strategy. The results showed that the control strategy consumed the lowest energy based on the combination of heat pump inlet water temperature and wet bulb temperature difference [18].

Both in the temperature control strategy and the temperature difference control strategy, the soil source is considered as the primary source and the cooling tower as an auxiliary cooling method, which means the system cannot take advantage of the ambient air source in the early and late cooling periods. So this paper proposes an operation strategy based on the control of wet bulb temperature. The operation performances under different operating control strategies are simulated by establishing a mathematical model of the system, and the feasibility of the control strategy based on wet bulb temperature control can be analyzed by comparison.

2. Mathematical model of hybrid ground-coupled heat pump system

Fig. 1 is a schematic of the parallel-form HGCHPS. The system consists of four parts: the heat pump unit, closed cooling towers, SHEs, and switching valves. The cooling tower or SHEs, to be served as the cold source heat pump, can be switched among each other by opening and closing the valves. The HGCHPS consists of multiple components. In order to study the dynamic characteristics of