OPTIMIZATION OF A GROUNDWATER HEAT PUMP

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Abstract: By analyzing the operating characteristics of the groundwater heat pump (GWHP), this paper puts forward an integrated optimal mathematical model with an objective function of the annual total costs according to technical and economic optimal principle. A computation program is also developed. On the premise of ensuring the unit to fulfil the heating and cooling requirements of buildings, this model is used to calculate and optimize some key operating parameters and components according to different temperature and depth of the groundwater. It combines the reliability with economy of heating and cooling and achieves the most optimal match of each part of the system. The present method will lead to a greater improvement in the performance of the heat pump than traditional method.

Keywords: groundwater heat pump; optimal design; mathematical model

Nomenclature:

a，b，c，d	coefficients
c_{el}	price of electricity, ¥/kWh
c_t	cost per unit weight of well pipe, ¥/kg
c_p	specific heat of water, kJ/(kg℃)
C_0	total cost per unit time, ¥/y
C_{eva}	cost of evaporator, ¥
C_{con}	cost of condenser, ¥
C_{com}	cost of compressor, ¥
C_{pum}	cost of well pump, ¥
C_{tub}	cost of well pipe, ¥
C_{el}	cost of additional electricity distribution installations, ¥
COP	coefficient of performance
CRF	capital recovery factor
d	diameter of well pipe, m
E	annual electricity consumption, kWh/y
F	area of heat exchanger, m^2
H	well pump head, kPa
h_1	enthalpy of refrigerant at the inlet of compressor，kJ/kg
h_{2a}	theoretical enthalpy of refrigerant at the outlet of compressor, kJ/kg
h_2	practical enthalpy of refrigerant at the outlet of compressor, kJ/kg
h_3	enthalpy of refrigerant at the outlet of condenser, kJ/kg
i	interest rate
K_e	heat transfer coefficient of evaporator, kW/(m^2 ℃)
K_c	heat transfer coefficient of condenser, kW/(m^2 ℃)
L	total length of well tube，m
n	depreciation time，y
N_e	power input of electric motor, kW
Q_0	specified cooling capacity of unit, kW
Q_h	specified heating capacity of unit, kW
T	total running time in a year（h/y）
t_e	evaporation temperature, ℃
t_c	condensation temperature, ℃
t_{m1}	design return water temperature, ℃
t_{m2}	design supply water temperature, ℃
t_{w1}	well water temperature, ℃
t_{w2}	well water temperature at the outlet of GEHP, ℃
V_h	theoretical volume flow of compressor, m^3/s
W_t	weight of well water pipe per unit length, kg/m
ΣC	total cost per unit time (exclude electricity cost), ¥/y
Δt_e	temperature difference in the evaporator, ℃
Δt_c	temperature difference in the condenser, ℃
τ	equivalent full-load hours, h/y
η_{el}	electric efficiency of compressor
η_p	efficiency of well pump
ρ_w	density of water, kg/m^3

Subscripts:

r	refrigerating
h	heating

0. Introduction

In recent years, the groundwater heat pump (GWHP) is achieving its rapid development in some parts of China. This type of heat pump that uses groundwater drawn from a deep well and injected into another simultaneously is an ideal unit for space heating and cooling applications. The groundwater is its heat sink for cooling in summer, and heat source for heating in winter. [1,2] Because of the effect of heat insulation and accumulation of the soil, the temperature of groundwater is nearly constant in a year and is generally 1～2℃ higher than the local annual average air temperature[3]. Therefore, the groundwater is an ideal heat source/sink for heat pumps. In practice, the performance of GWHP is effected seriously by such facts that the property of the well (temperature of groundwater, water level, etc) varies with areas and the proportion of heating load to cooling load also varies with buildings. In order to ensure its reliability and improve its economy, it is important to design the system properly according to the property of the wells and the actual needs of the buildings.

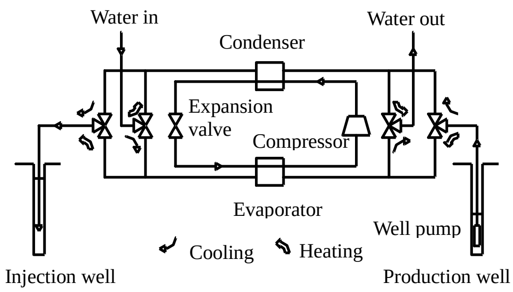

Fig. 1 Schematic of groundwater-source heat pump

Optimization design of GWHP is to realize optimal cooperation for all components and find improving direction at the same time by considering the running costs and first investment simultaneously under present technical and economic conditions.

1. Mathematical model

1.1 System and objective function

The GWHP studied in this paper is illustrated in figure 1. The groundwater is heat source in the heating mode, and heat sink in the cooling mode. The three-way valves in the pipeline fulfill the switch of different running modes.

The objective in this paper is to minimize the total cost per unit time C_0, which includes both the operating (electricity) cost and the capital cost, for a given heating and cooling capacity. The operating cost increases if the investments decrease and vice verse.

$$C_0 = c_{el} E + \sum C \qquad (1)$$

1.2 Variables and independent variables

The following decision variables are supposed to have been known: cooling capacity and chilled water temperatures as the system running in cooling mode; heating capacity and heated water temperatures as the system running in heating mode; water temperature of production well. So the state of the system could be determined by each set of the following seven variables: t_{er}、t_{eh}、t_{cr}、t_{ch}、t_{w2r}、t_{w2h}、d. The areas of the evaporator and condenser, the specific compressor displacement and power input to the compressor and the power input to well pump can now easily be determined with these variables. Therefore, these seven variables are chosen as independent variables in this paper.

1.3 Analysis of objective function

The annual consumption of electricity in equation 1 is composed by the consumption of compressor and well pump. The electric consumption of compressor is calculated with equivalent full-load method in this paper.

$$E = \tau_r \frac{Q_0}{h_{1r} - h_{3r}} \frac{h_{2ar} - h_{1r}}{\eta_{elr}} + \tau_h \frac{Q_h}{h_{2h} - h_{3h}} \frac{h_{2ah} - h_{1h}}{\eta_{elh}} + \frac{Q_0(h_{2r} - h_{3r})H_r T_r}{\rho_w c_p (h_{1r} - h_{3r})(t_{w2r} - t_{w1r})\eta_{pr}} + \frac{Q_h(h_{1h} - h_{3h})H_h T_h}{\rho_w c_p (h_{2h} - h_{3h})(t_{w1h} - t_{w2h})\eta_{ph}} \tag{2}$$

Reference is made in equation 2 to the equivalent full-load hours (τ_r, τ_h) and plant on-hours (T_r, T_h) for design conditions. These hours should be determined according to the building load, running time and part load performance of the GEHP. They will effect on the results of optimization directly and should be given careful consideration according to the practical values of existing similar buildings if necessary. The lifts of well pump, H_r and H_h, are consisted of pressure loss in the unit, pressure difference arising from the water level difference between production well and injection well, and pressure loss in the pipeline.

The costs for parts slightly affected by alternative construction of the system, such as pipes connection the components, are just added as constants as they have no effect on the optimization. So the total capital cost per year, ΣC, doesn't include the costs mentioned above, and is defined as follow:

$$\Sigma C = CRF(i,n)(C_{eva}+C_{con}+C_{com}+C_{pum}+C_{tub}+C_{el}) \tag{3}$$

$CRF(i,n)$ is annuity factor of the capital investment, defined as:

$$CRF(i,n) = i/[1-(1+i)^{-n}]$$

Therefore, C_0 in equation 1 may be defined as computational total cost per unit time in order to distinguished with actual total cost.

The following simple cost relationships are assumed for the evaporator and condenser to be valid in the region of optimization (these relationships may be changed due to other assumptions).

$$C_{eva} = aF_{eva}{}^{b} \qquad C_{con} = aF_{con}{}^{b}$$

Where,

$$F_{eva} = \frac{Q_0}{K_{er}\Delta t_{er}} \qquad F_{eva} = \frac{Q_h(h_{1h} - h_{3h})}{K_{eh}(h_{2h} - h_{3h})\Delta t_{eh}}$$

$$F_{con} = \frac{Q_h}{K_{ch}\Delta t_{ch}} \qquad F_{con} = \frac{Q_0(h_{2r} - h_{3r})}{K_{cr}(h_{1r} - h_{3r})\Delta t_{cr}}$$

The cost of compressor and its driven device is assumed to be a function of specified power input.

$$C_{com} = c\max(N_{er}, N_{eh})^{d}$$

The power inputs as heating and cooling are calculated respectively. The specified power input of the compressor is determined according as the bigger.

$$N_{er} = \frac{Q_0}{h_{1r} - h_{3r}} \frac{h_{2ar} - h_{1r}}{\eta_{elr}} \qquad N_{eh} = \frac{Q_h}{h_{2h} - h_{3h}} \frac{h_{2ah} - h_{1h}}{\eta_{elh}} \tag{6}$$

The cost of well pump and its driven device is attained in the same method as compressor. The cost of well water pipeline could be rough estimated according to its weight:

$$C_{tub} = L \cdot W_t \cdot c_t \tag{7}$$

1.4 Constraints

In the process of optimization, the changes of variables shouldn't overstep some practical bounds. Thereby, the constraints determined according to practical conditions are:

(1) The condenser temperature is greater than evaporator temperature: $t_{cr}>t_{er}$, $t_{ch}>t_{eh}$;

(2) In condenser, the condensation temperature of refrigerant is greater than outlet temperature of water, and outlet temperature of water is greater than its inlet temperature: $t_{cr}>t_{w2r}>t_{w1r}$, $t_{ch}>t_{m2h}>t_{m1h}$;

(3) In evaporator, the evaporation temperature of refrigerant is less than outlet temperature of water, and outlet temperature of water is less than its inlet temperature: $t_{er}<t_{m2r}<t_{m1r}$, $t_{eh}<t_{w2h}<t_{w1h}$;

(4) The outlet temperature of water in evaporator is greater than 5℃ to avoid freezing.

(5) Either the area of evaporator or the area of condenser is constant whatever conditions the system is in. Then

$$F_{evar} = F_{evah} \qquad F_{conr} = F_{conh}$$

In conclusion, the optimal design of GEHP is a multivariable nonlinear constrained minimization problem. In order to find the global minimum, common sense and insight into how the system works must be used to determine the value of a solution.

2. Example

The approaches discussed here are illustrated with the aid of a HVAC system of an office building. The designed heating and cooling loads for this building is 400kW and 340kW respectively. Plant on-hours for cooling and heating in a year are 1410h/a and 1350h/a respectively. Equivalent full-load hours for cooling and heating are 570h/a and 650h/a respectively. The electric price is 0.4 ¥/kWh, and additional electric establishment is about 300 ¥/kW. Four same size GEHPs are designed to service for the building with 45~50℃ hot water in winter and 12~7℃ chilled winter in summer. The four units use a same production well and a same injection well which depths are both 100m. The water temperature of production well, 20℃, is nearly constant in a year. The refrigerant used in the GEHPs is R22. Evaporator and condenser are both shell-and-tube heat exchangers with enhanced surface tubes. A semi-hermetic reciprocating compressor is utilized in the plant. The depreciation for the system is 10 years. The interest rate is 10%.

A computer program has been developed for finding the optimum system using a Variable Metric algorithm. The program begins by calculating the thermodynamic data for the assumed refrigerant R12 (other refrigerants may also be used.) with CSD equations of state. The optimization results are given in Table 1. Results with a conventional design method are also given in Table 1 for comparison.

Tab.1 Comparison of the results between the optimization and convention design methods.

	d(mm)	F_{eva}(m^2)	F_{con}(m^2)	V_h(m^3/s)	N_{er}(kW)	N_{eh}(kW)	COP_r	COP_h
Optimization	150*4.5	8.1	9.1	0.022	11.6	21.1	7.3	4.7
Convention	150*4.5	3.4	4.6	0.026	16.1	25.5	5.3	3.9
	C_0(¥/y)	$c_{el}E$(¥/y)	ΣC(¥/y)	E(kWh/y)	E_r(kWh/y)	E_h(kWh/y)	E_{pr}(kWh/y)	E_{ph}(kWh/y)
Optimization	60009	36862	23147	92156	26537	54814	5182	5623
Convention	67227	46078	21149	115194	36637	66194	7390	4973

Comparing with conventional design, optimal design could save the expenses of 7218 yuan every year (about 11%), because the greater savings of operating cost and the investments of compressor and well pump offset the increase of investments caused by increased areas of evaporator and condenser.

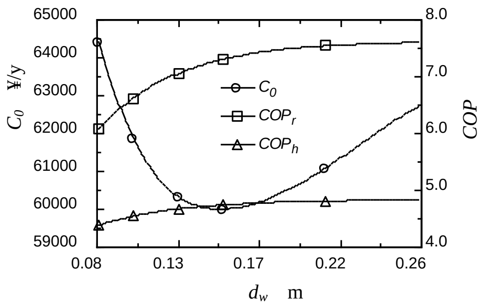

Fig.2 Effect of the well tube diameter on annual total cost and performances of heat pump.

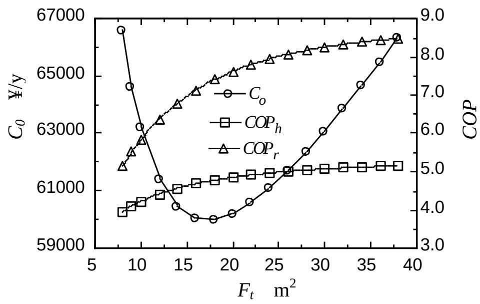

Fig.3 Effect of the total heat exchanger area on annual total cost and performances of heat pump

In Figure 2 the total cost per unit time and coefficient of performance (COP) for the system are shown as functions of the diameter of well water pipe. The electric power consumption of well pump decreases and COP increases with the increase of pipe diameter, which explain the rapid decline in the total cost per unit time in the initial stage. But the investment caused by increased diameter of well water pipe exceeds the decrease of operating cost in the later stage, which explain the increase in the total cost per unit time after the minimum (optimal) point.

In Figure 3 the total cost per unit time and COP for the system are shown as functions of the total heat exchanger area (F_{eva}+F_{con}) of one unit. The COP increases and, therefore, the electric power consumption of compressor decreases with the increase of total heat exchanger area, which explain the rapid decline in the total cost per unit time in the initial stage. But the improvement of COP becomes smaller gradually, the investment caused by increased heat exchanger area exceeds the decrease of operating cost in the later stage, which explain the increase in the total cost per unit time after the minimum (optimal) point.

In addition, the mathematical models developed above don’t deal with the size of plant, which is usually an important guideline to evaluate the performance of a plant. Lesser heat exchanger area could reduce the size of plant. From Figure 3 it is seen that the total cost per unit time is not sensitive to the change of total heat exchanger area near the optimal point. So the total heat exchanger area chosen finally should be suitable smaller than optimal value. For example, another calculation is performed when the total heat exchanger area is given as 13m^2. The new results are shown in Table 2. Comparing with Table 1, we can find that the total heat exchanger area for each unit reduces 5.4m^2 (about 29%) but the increase of the total cost per unit time is slight

Tab.2 Results of optimization design when total exchanger area is 13 m^2.

	Units	Results
C_0	¥/y	56666
$c_{el}\,E$	¥/y	39084
ΣC	¥/y	17582
F_{eva}	m^2	6.1
F_{con}	m^2	6.9
V_h	m^3/s	0.023
N_{er}	kW	12.9
N_{eh}	kW	22.3

3. Conclusions

It is feasible for the optimization design of the groundwater source water-to-water heat pump with computer. Solving the set of equations in the mathematical model that minimize the objective function, i.e., lowest cost, will establish the most cost-effective design parameters for the specific design configuration analyzed. The workload of design could also be reduced greatly.

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