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Optimal Design and Operation of a District Heating System Coupled with a Deep Open Loop Geothermal Well and Heat Pumps

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ABSTRACT

A thermal and economic analysis of a district heating system coupled with a deep open-loop geothermal well (DOLGW) and heat pumps was performed in the paper. A typical configuration of such a geothermal district heating system was proposed, and the steady heat transfer equations at each equipment from the source of geothermal well to the terminal radiator of heat users were listed. The Newton's iteration method was adopted to find the steady supplied heat capacity of the system. Based on the analysis and the tested data of a real DOLGW, an optimal size selections of heat pumps, plate heat exchangers, and terminal heat radiators for such a system can be determined theoretically.

1. INTRODUCTION

Geothermal district heating represents about 59% of annual energy use in the categories of direct uses of geothermal energy according to the latest worldwide review by Lund and Toth (2021). China has been the top leading country with the largest annual use of geothermal energy for more than two decades. However, it is noticed that the annual energy use with geothermal (ground source) heat pump takes more than half of the total geothermal energy use in China, and most of those geothermal heat pump systems were coupled with shallow borehole heat exchanger (BHE). One of reasons is that the local official government was very cautious about geothermal energy exploration in a manner of pumping and reinjection, especially for the case with a difficulty in reinjection. In recent years, the technology of a heat extraction system so called "no water withdrawn but heat only" has been given more and more attentions recently in many countries (e.g. Alimonti, *et al*. (2018)). In fact, this technology is not new, which can be traced back to about 40 years ago. Downhole heat exchanger (DHE) (e.g. Allis and James (1980), Freeston and Pan (1984)) and borehole heat exchanger (BHE) (e.g. Rybach and Eugster (2010)), for examples, are the two typical representatives of heat extraction only. One of main differences between BHE and DHE is that BHE should be used alternatively for both heating and cooling with season changing in a year, while DHE can only be used for heating in winter. Another difference is that BHE is generally installed in shallow ground formation of soil, but DHE is generally suspended in a shallow geothermal well drilled in a permeable aquifer with a large geothermal gradient. In a thermodynamic point of view, the shallow underground layer where BHEs are installed does not belong to the geothermal resources, e.g. Dai and Chen (2008). With the technical development in well drilling, the heat stored in deep reservoir is accessible economically. This leads to an increasing research interest on the heat extraction from a deep layer underground recently. It is noticed that, up to date, most of previous studies were on the heat extraction performance of a closed deep borehole heat exchanger (DBHE) (e.g. Alimonti, et al. (2018), Kohl, et al. (2002), Collins and Law (2017)) and a few on the open looped design, e.g. Westaway, (2018), and Dai, et al., (2019). In the present paper, the thermal behavior of a geothermal district heating system coupled with a deep open loop geothermal well will be investigated.

The doublet well system with both production and reinjection wells has been the most typical way of geothermal energy uses. However, a difficulty might be encountered in reinjection, particularly provided that the reinjection well was drilled in a shallow sandstone reservoir e.g. Chai, et al., (2022). In order to provide a sustainable and efficient way for extracting heat underground, the studies on the geothermal district heating by using DBHE as the heat source have been increased in recent years (e.g. Alimonti, et al., (2020)). Numerous of papers can be found in the literature in various aspects of DBHE, such as the sustainable maximum heat extracted rate, the performance of CO_2 as the circulation fluid, the inner pipe material insulation, the thermal influence distance, etc. (e.g. Kong, et al., (2017), Roman, et al., (2008), and Henrik, et al., (2015)). Since the outlet fluid temperature from DBHE is generally low except at a very low circulation mass flowrate (e.g. Laszlo, et al., (2020)), which cannot meet the lowest limit of district heating so that heat pumps have to be coupled with in a real application. Zhang (2021) performed a numerical study by taking into the accounts of well depth, inner pipe material insulation, circulation flowrate, etc.. and the results showed that an optimal circulation flowrate did exist for getting a maximum COP, which increases with the well depth. Wang et al. (2022b) also gave a numerical modeling of DBHE coupled with heat pumps based on a thermodynamic analysis of working fluid. The thermal performances of such a district heating system with different intermittence operation modes are also simulated and analyzed (e.g. Wang, et al., (2022b)). However, it is noticed that even though the geothermal district heating system coupled with DBHE and heat pumps shows its sustainability and a relatively larger COP comparing with that of shallow BHE, the demonstration projects showed that the sustainable heat extraction rate per wellbore length was low, which was only about 100W/m (e.g. Kong, et al., (2017), Wang, et al., (2017), and Cai, et al., (2021)). Therefore, various types of subsidies and grants from the government at the local or national level might be essential for some private entrepreneurs to pursue this line of investment considering the high capital investment in well drilling and surface equipment and construction at the wellhead (e.g., Maciej, et al., (2015)). Therefore, it is crucial to increase the heat extraction rate from a single geothermal well without net water withdrawn for the economic assessment of such a project.

2. A DISTRICT HEATING SYSTEM COUPLED WITH A DOLGW AND A HEAT PUMP

In general, the outlet fluid temperature from the DBHE or DOLGW is low and not suitable for space heating directly, so that a heat pump has to be coupled with. A typical district heating system coupled with a DOLGW and a heat pump, as shown in Fig. 1, can be designed. The setup consists of three parts: (1) the open circulation loop of geothermal water from the DOLGW to plate heat exchanger; (2) clean water circulation loops through the evaporator and condenser of the heat pump, respectively, and (3) clean water circulation loop through terminal radiators in buildings.

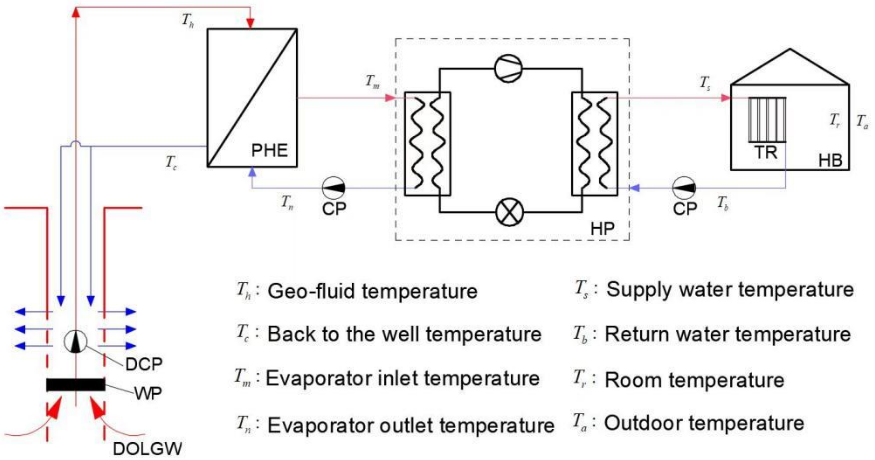

Figure 1: The district heating setup coupled with a DOLGW and a heat pump.

The following assumptions are made in order to simplify the thermal analysis. Firstly, all of the equipment installed in the system are at thermal steady state, no change with time. Secondly, The heat storage effect of pipelines, equipment, and walls of buildings is ignored. The heat transfer balance equations in a chain from the DOLGW to the building can be written down. which include:

(1) The heat extracted rate from the geothermal well:

$$Q_1 = (9517.428 - 589.769T_c)G_h - 610600 + 59155T_c \tag{1}$$

The equation above was a fitting curve based on the field test data of DOLGW obtained recently. The correlation indicates that the heat extracted is proportional to the circulation flowrate, G_h, and increases with the decrease of fluid temperature back to the well, T_c.

(2) The heat released from the geothermal water:

$$Q_2 = C_p G_h (T_h - T_c) \tag{2}$$

where T_h is the outlet temperature of geothermal water from the DOLGW.

(3) The transferred heat at the plate heat exchanger can be expressed as:

$$Q_3 = \frac{C_p (T_h - T_n)(S-1)}{\dfrac{S}{G_h} - \dfrac{1}{G_m}} \tag{3}$$

where G_m is the clean water circulation flowrate through the plate heat exchanger and the evaporator of heat pump. T_n is the water temperature back to the plate heat exchanger from the evaporator. S is a coefficient related to the design and arrangement of plate heat exchanger as:

$$S = \exp\left[\frac{KAF}{C_p}\left(\frac{1}{G_h} - \frac{1}{G_m}\right)\right] \tag{4}$$

where K is the total heat transfer coefficient of plate heat exchanger, "A" is the effective heat transfer area, F is a correction factor while the plate heat exchanger is different from the design of "one pass / one pass" counter current arrangement.

(4) The heat transferred at the evaporator can be given by:

$$Q_4 = C_p G_m (T_m - T_n) \tag{5}$$

in which T_m is the outlet water temperature from the plate heat exchanger.

(5) The heat transferred at the condenser side can be given by:

$$Q_5 = \mathrm{COP} \cdot W \tag{6}$$

where COP is the coefficient of performance of the designed heat pump, and W is the input electric power of the heat pump. In general, the COP of a heat pump depends on the manufacturer and the product standard. In a thermodynamic point of view, COP can be related to the temperature difference between the averaged fluid temperatures of condenser and evaporator. In the present analysis, a simple linear function of COP was adopted according to the field test data, which is

$$COP = 12.271 - 0.329\theta \tag{7}$$

in which θ can be written as:

$$\theta = \frac{T_s + T_b}{2} - \frac{T_m + T_n}{2} \tag{8}$$

The measured COP of the heat pump shows approximately a negative linear function against the average temperature difference, θ, between condenser and evaporator, as given by Eq. (8).

(6) The heat released from the circulation water through the condenser to the buildings can be given by:

$$Q_6 = C_p G_c (T_s - T_b) \tag{9}$$

where T_s and T_b are the circulation water temperatures supplying to and back from the buildings, respectively. G_c is the mass flowrate of the circulation water through the condenser and terminal radiators in the building.

(7) The terminal radiators inside the buildings take the duty of releasing heat of circulation water to the buildings in a natural convection heat transfer model, i.e.:

$$Q_7 = A_t \alpha \left(\frac{T_s + T_b}{2} - T_r\right)^{\beta+1} \tag{10}$$

where A_t is the heat transfer area, and α, β are the coefficients given by the radiator manufacturer reflecting the heat transfer performance of the product.

(8) The heat supplied to the buildings finally emits to the environment at temperature of T_a, while the buildings is maintained at a comfortable temperature of T_r.

$$Q_8 = C_q V (T_r - T_a) \tag{11}$$

where C_q is the volumetric heat loss coefficient of buildings, V is the total volume of heated buildings, and T_a is the ambient temperature. According to the given data in Table 1, the equivalent heating load per floor area of the buildings is about 60W/m^2, which corresponds approximately to the case of building in the northern part of China.

Table 1: Parameters used in the thermal analysis

DOLGW	Circulation mass flowrate, G_h	38 t/h or 10.56kg/s
Heat Pump	Arrangement	one pass / one pass counter current
	Effective heat area per plate	0.8 m2
	Number of plates, Np	80
	Channel spacing	5 mm
	Plate thickness	0.5 mm
	Material of Plate	Stainless steel
	Thermal conductivity of plate	16.28 W/(m2.K)
	Fouling thermal resistance	0.00017 (m2.K)/W
	Electric Power	180 kW
Terminal Radiator	Cast iron	
	Effective heat transfer area, A_t	6000 m^2
	Coefficient, α	2.15
	Coefficient, β	0.35
Building	Total Volume of Building, V	26000 m^3
	Specific heat load per volume of building, C_q	1.0 W/(m^3·K)

3. SOLUTION METHODS

According to the assumption made and the energy balance equations above Eqs. (1) to (11), we have $Q_1 = Q_2 = Q_3 = Q_4$, and $Q_4 + W = Q_5 = Q_6 = Q_7 = Q_8$. By some necessary simplification operations, a complicated transcendental equation about the heat transfer rate Q_5 can be derived. The Newtonian iteration method was applied for finding the root of Q_5 (e.g. Dai, (1997)). The iteration equation is:

$$Q_{n+1} = Q_n - \frac{Q_n - 11.271W + 0.329W\theta}{1 + 0.329W\left[\frac{1}{1+\beta}\frac{1}{A_t\alpha}\left(\frac{Q_n + W}{A_t\alpha}\right)^{\frac{-\beta}{1+\beta}} + \frac{1}{C_qV} + \frac{T_h}{C} - \frac{1}{2C_pG_m}\right]} \tag{12}$$

where C is given by:

$$C = \frac{C_pT_h(S-1)}{\frac{S}{G_h} - \frac{1}{G_m}} \tag{13}$$

After getting the heating power of Q_5, or Q_2, the temperatures at each section can be obtained simultaneously.

4. RESULTS AND DISCUSSION

4.1 Heat transfer characteristics of DOLGW against different outdoor temperature

Figs. 2 (a) to (d) show the calculated results at different given parameters. In addition to the calculated heating power or the heat load of DOLGW system, the temperatures at different equipment can also be obtained. Fig. 6(a) shows the heating power at a variable flowrate of G_m.

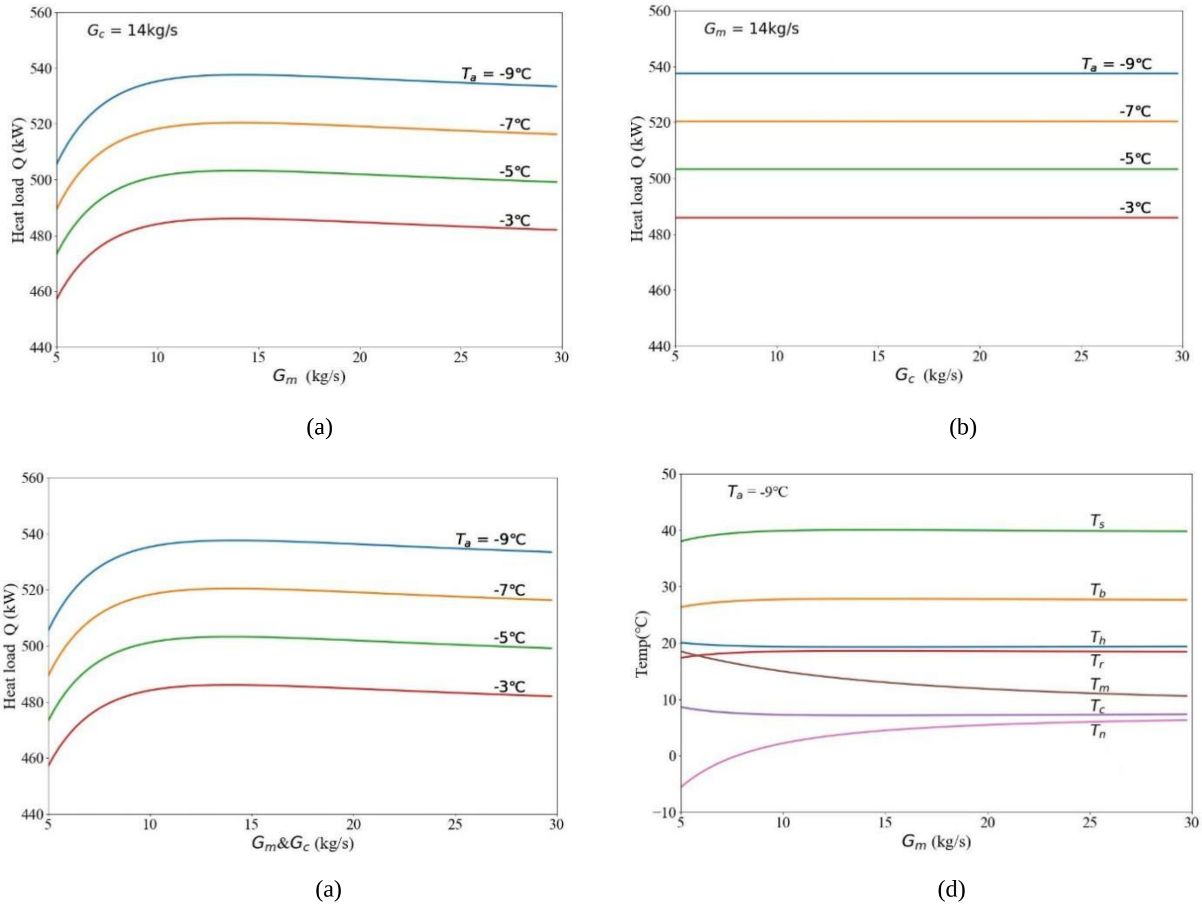

Figure 2: The calculated heat load at a variable flowrate of G_m and a fixed G_c (a), a variable flowrate of G_c and a fixed G_m (b), variable both G_m and G_c, and $G_m = G_c$ (c), the temperatures at different section with T_a = -9°C and a variable G_m , (d).

It can be seen that the total heating power increases with the decrease of ambient temperature, T_a, and a maximum heating power exists while G_m is at approximately 14.0 kg/s, or the flowrate ratio of G_h/G_m is 0.75. A similar phenomenon had been observed for the indirect geothermal heating system by doublet well (e.g. Dai, (1997), and Dai and Liang, (1999)). However, in this case the optimal flowrate ratio is 0.75 instead of 0.83 obtained by Dai and Liang (1999). By changing the circulation mass flowrate of G_c in a range of 5 to 30 kg/s does not bring any changes to the heating power at a fixed G_m as shown in Fig. 2(b). Fig. 2(c) shows that simultaneously varying both G_m and G_c has an identical heat output with the case of varying G_m alone.

Fig. 2(d) shows that a stable room temperature, T_r, can be obtained while G_m is larger than 10kg/s, which is over 18°C, the lower temperature limit of heating standard. It is worthy to point out that the inlet temperature to the evaporator T_n should not be lower than 6°Cto prevent the possible ice formation in the evaporator of heat pump. Therefore, a relatively large flowrate of G_m, for example, 20 kg/s is recommended. Similar output curves can be observed with a larger ambient temperature. Even the heating power decreases with increasing T_a, the room temperature meet the standard requirement and larger than the case of T_a = -9 °C.

4.2 Optimal selection of HP and operation of DOLGW

As shown in Fig. 3, the open circles connected with the dash line indicates the minimum total investment, C_t, including the costs of heat pump C_{HP}, plate heat exchange C_{PHE}, and terminal heat radiator C_{TR}. In the present economic analysis, the specific costs per heating watt were adopted, i.e., the specific cost ratio of $c_{HP} : c_{PHE} : c_{TR}$ was given by 850 : 30 : 1.6, where c_{HP} is equal to C_{HP}/Q_5, c_{PHE} is equal to C_{PHE}/Q_3, and c_{TR} is equal to C_{TR}/Q_7. It shows that the total investment of C_t is mostly sensible to the selection of heat pump capacity. The optimum heat pump capacity is about 160 kW, and correspondingly the optimal heat transfer areas of PHE and TR are 63 plates, and 6300 m^2, respectively, which is a little different from the case of minimizing C_{HP}+ C_{TR}, as shown in Fig. 6(f).

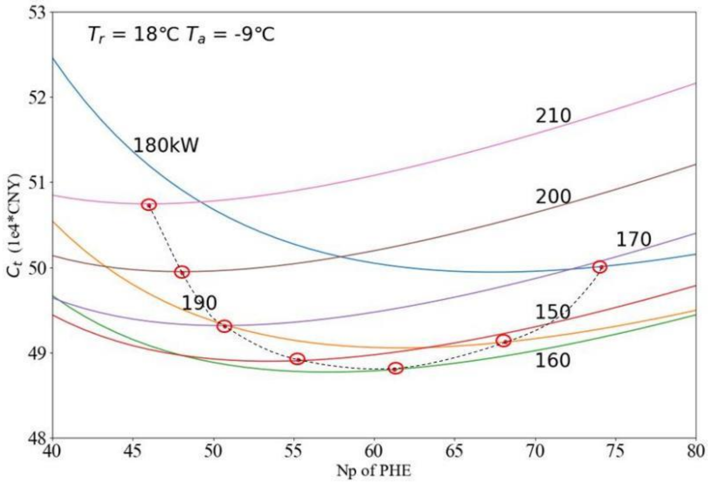

Figure 3: The total cost of Ct vs the number of plate Np of PHE at various input electric powers of HP.

5. CONCLUSIONS

The thermal analysis of a district heating system coupled with a deep open looped geothermal well (DOLGW) and a heat pump was performed in the present paper. Based on the empirical correlation of the heat extraction rate of DOLGW and the COP of heat pump, the stable heating power and the temperatures at different equipment of those circulating loops were obtained. An optimum mass flowrate ratio 0.75 of hot side to cold side of plate heat exchanger exists for such a district heating system, which is relatively smaller than the case of doublet well system. In addition, the changing flowrate at the condenser side does not show much effect on the total heating power.

By giving the specific costs per heating power of heat pump, plate heat exchanger, and terminal heat radiator, in a ratio form of 850:30:1.6, the total investment of the three equipment can be minimized with the lowest limitation of heating duty. It shows that since the heat pump is the most expansive one among the three, the optimum size selection of these equipment was dominated mainly by the cost of heat pump, and the optimum size of terminal heat radiator seems unchanged with both the number of plates of PHE and the electric capacity of HP.

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