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DIRECT AND INDIRECT LOW TEMPERATURE GEOTHERMAL DISTRICT HEATING SYSTEMS

Chuanshan Dai and Jun Liang
Tianjin Geothermal Research and Training Center, Tianjin University, 300072, Tianjin, P.R.China

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ABSTRACT

The selection of direct or indirect use of low temperature geothermal district heating systems is based on the quality of geothermal water. An indirect geothermal heating district system is strongly recommended if the concentration of some corrosion inducing chemicals such as chlorite in geothermal water is high. In the paper, a theoretical method capable of quantitatively evaluating the thermodynamic differences between direct and indirect geothermal district heating systems is proposed. The contents include: the main factors effecting the difference between the two kinds of heating systems; the determination of a best operation mode for an indirect geothermal district heating system and the method of finding optimum design parameters. Since plate and frame heat exchangers (PHEs) are commonly used in indirect geothermal district heating systems; the advantages of using the PHEs with variable plate spacing are also discussed. For comparing the thermal performance between direct and indirect geothermal district heating systems, a concept of relative thermal efficiency is proposed. Once the PHEs are designed in an indirect geothermal district heating system; the optimum flowrate ratio of geothermal water to circulating water can be found through an equation derived in the paper, which has been proved in practice.

1. INTRODUCTION

District heating systems with low temperature geothermal resources (< 100°C geothermal water) are different from those with the fossil fuels boilers. Since the wellhead temperature is different from well to well, the design supply water temperature cannot be set to a standard as it is in conventional boiler heating systems. Some features about the direct and indirect geothermal district heating systems have been discussed by Dai (1997). In this paper a further analysis is presented. To concise the paper, the detailed derivation of some equations and the assumptions upon which these are based on are not given. However readers are referred to the paper by Dai (1997).

2. THE THERMAL PERFORMANCE COMPARISON

The thermal performance comparison between direct and indirect geothermal district heating systems is based on the same wellhead temperature, heating load, and outdoor and indoor temperatures.

2.1 Thermal balance Equations

All the equations below are derived at the thermal steady state conditions, i.e. the ambient conditions are remained unchanged. The transient heat transfer or the heat storage effect of the building wall on the district heating system has not been considered.

The thermal balance equation for the direct heating system is:

$$Q_d = G_h C_p \Delta T$$

$$\Delta T = C_2^{\frac{1}{\beta}} \left[\ln\left(1 + \frac{\Delta T}{T_h - T_a - (1 + C_1)\Delta T}\right) \right]^{\frac{1+\beta}{\beta}} \qquad (1)$$

$$C_1 = \frac{G_h C_p}{C_v V} \qquad C_2 = \frac{G_h C_p}{A_r \alpha}$$

in which Q_d is the heat transferred or heat load for a direct heat system. G_h is the mass flowrate of geothermal water. C_p is the specific heat capacity of water at constant pressure (kJ/kg°C); ΔT is the temperature drop of geothermal water flowing through the consumer's in-building equipment (=T_h-T_{do}). T_h is the supply temperature of geothermal water; T_{do} is the outlet temperature of geothermal water used for a direct heating system. T_a is the outdoor temperature. C_v is the specific volumetric heat load capacity of building (W/m^3°C). V is the total volume of a heated building. α and β are empirical constants of heat radiators in the form of $k_r = \alpha \Delta T_m^{\beta}$. k_r is the overall heat transfer coefficient of radiators. ΔT_m is the mean temperature difference between the heat radiator surface and the air in rooms. A_r is the total heat transfer area of heat radiators. The above equation can be solved for ΔT by iteration method, beginning with a proper initial guess value. Then the heat load Q_d can be obtained.

The indirect heating system is different from the direct heating systems as it has PHEs for the heat transferred from geothermal water to the circulating water. The thermal balance equation can be written as:

$$B_1 Q_{id} + B_2 Q_{id}^{\frac{1}{1+\beta}} = B_3 \qquad (2)$$

where
$$\begin{cases} B_1 = C_1 - \dfrac{R_2}{2} + \dfrac{1}{P_2} \\ B_2 = \dfrac{G_h C_p}{(\alpha A_r)^{\frac{1}{1+\beta}}} \\ B_3 = G_h C_p (T_h - T_a) \end{cases} \qquad (3)$$

in which R_2 is the heat capacity rate ratio. P_2 is the temperature effectiveness based on the stream of geothermal water. It is a function of Number Heat Transfer Unit (Ntu_2) and R_2 (Kays and London, 1984). R_2 can be approximately given as G_h/G_c since the fluids on both sides are waters.

$$P_2(Ntu_2, R_2) = \begin{cases} \dfrac{1-\exp(-Ntu_2(1-R_2))}{1-R_2\exp(-Ntu_2(1-R_2))} & \text{for } R_2 \neq 1 \\ \dfrac{Ntu_2}{1+Ntu_2} & \text{for } R_2 = 1 \end{cases} \tag{4}$$

2.2 The Relative Thermal Efficiency η

Suppose two wells have the same wellhead parameters and used for district heating with the same heating duty. But one is a direct heating system and the other is an indirect heating system. The geothermal water flowrates should be different. The direct geothermal district heating system needs less. But how the difference changes, for example with the outdoor temperature, the heating load factor and the circulating water flowrate of the indirect heating system. The concept of **relative thermal efficiency** η is proposed, which can be written as:

$$\eta = \frac{T_h - T_{ido}}{T_h - T_{do}} \tag{5}$$

in which T_{ido} and T_{do} are the outlet temperatures for indirect and direct geothermal district heating systems, respectively. For a specific designed indirect geothermal district heating system, an optimum running condition exists on which has the best flowrate match of geothermal water and circulating water (Dai, 1997). Therefore for the indirect heating system itself should have a thermal efficiency which named as **thermal efficiency of indirect system** η_{id} to evaluate the optimal running conditions. We have:

$$\eta_{id} = \frac{T_h - T_{ido}}{T_h - (T_{ido})_{opt}} \tag{6}$$

The optimum indirect district heating system gives a maximum relative thermal efficiency called **relative optimum thermal efficiency** η_{opt}. Since it supplies the same amount of heat as the direct heating system ($Q_{id} = Q_d$), η_{opt} can be written as:

$$\eta_{opt} = \frac{T_h - (T_{ido})_{opt}}{T_h - T_{do}} = \frac{(G_h)_d}{(G_h)_{id}} \tag{7}$$

subscript "opt" means optimum and "d" and "id" are for direct and indirect heating systems, respectively. Accordingly, the equation (5) can be written as $\eta = \eta_{opt} \times \eta_{id}$.

3. THE ACCESS OF BEST OPERATION FOR INDIRECT SYSTEMS

Note that the first part of equation (2) is mainly the consideration of the heated building and the PHEs, the second part the terminal heat radiators, and the third part the outdoor temperatures. Further inspection of equation (2) reveals that if B_2 and B_3 are known, the minimum of B_1 is the point where the maximum of Q_{id} can be achieved. B_1 is a function of P_2 and R_2. Therefore, the best operation problem of this kind of heating systems becomes that of finding a suitable flowrate of circulating water G_c or R_2 (G_h is constant) in order to get the minimum B_1 or the maximum heat Q_{id}. Differentiating equation (2) with respect to G_c and let it equal to zero, we have:

$$\frac{1}{P_2^2}\frac{dP_2}{dG_c} = \frac{G_h}{2G_c^2} \tag{8}$$

where dP_2/dG_c can be written in the following form:

$$\frac{dP_2}{dG_c} = \frac{A}{G_h C_p}\frac{\partial P_2}{\partial Ntu_2}\frac{dk}{dG_c} - \frac{G_h}{G_c^2}\frac{\partial P_2}{\partial R_2} \tag{9}$$

above A is the effective heat transfer area of PHEs. The overall heat transfer coefficient k of PHEs is:

$$\frac{1}{k} = \frac{1}{\alpha_h} + \frac{1}{\alpha_c} + r \tag{10}$$

$$\text{or} \quad k = \frac{\alpha_h}{\overline{C}_2 R_2^m + \overline{C}_1} \tag{11}$$

if ignore the thermal properties difference due to the temperature.

$$\text{where} \quad \overline{C}_1 = 1 + r\alpha_h \tag{12}$$

$$\text{and} \quad \overline{C}_2 = \begin{cases} 1 & \text{for ordinary PHEs} \\ (\varphi|\tilde{A})^{m} & \text{for PHEs with unequal flow cross-section area} \end{cases} \tag{13}$$

in equation (10) α_h, α_c are the film convective heat transfer coefficients of PHEs at the hot and cold sides of PHEs, respectively. r is the thermal resistance due to fouling. The power index m is an empirical constant of heat transfer performance of PHEs in the form of $Nu = CRe^m Pr^n$. φ and γ are the flow channel cross-section area ratio and the equivalent diameter ratio of wide channel side to that of narrow side. The derivative of k with respect to G_c is

$$\frac{dk}{dG_c} = \frac{\alpha_h \overline{C}_2 m R_2^{m+1}}{G_h(\overline{C}_2 R_2^m + \overline{C}_1)^2} \tag{14}$$

Substitution into equation (9) and using equation (10) we get finally a differential equation,

$$\frac{\partial P_2}{\partial Ntu_2} \cdot Ntu_2 \cdot \frac{mR_2^{m-1}}{(R_2^m + \overline{C}_1/\overline{C}_2)} = \frac{P_2^2}{2} + \frac{\partial P_2}{\partial R_2} \tag{15}$$

The above equation can be simplified by using the following definitions,

$$\varepsilon_1 = \frac{Ntu_2}{P_2}\frac{\partial P_2}{\partial Ntu_2} \tag{16}$$

where ε_1, is the sensitivity of effectiveness to Ntu_2, and

$$\varepsilon_2 = \frac{R_2}{P_2}\frac{\partial P_2}{\partial R_2} \tag{17}$$

is the sensitivity of effectiveness to the heat capacity rate ratio R_2. These definitions are the same as described by Kovarik (1989). Consequently, equation (15) can be written as:

$$\frac{mR_2^m}{(R_2^m + \overline{C}_1/\overline{C}_2)} = \frac{0.5P_1 + \varepsilon_2}{\varepsilon_1} \tag{18}$$

Equation (18) is a nonlinear equation of R_2. Even though it is difficult to exhibit the solution of this equation in explicit form, numerical analysis provides a means to find a solution. Note that the right side of the above equation is a general function of the heat transfer units Ntu_2, the capacity rate ratio R_2 and the arrangement of a heat exchanger. It can be calculated with the same procedure as the temperature effectiveness.

4. THE OPTIMUM DESIGN OF PHEs AND HEAT RADIATORS

There are two heat transfer areas that are needed to be determined in the equation (2). One is the heat transfer area of the plate heat exchanger, A, which is included in the temperature effectiveness P_2. And the other is that of the terminal heat exchanger, A_r. When the flow arrangement of the PHE is selected, P_2 is a function of the number of heat transfer units Ntu_2 and the heat capacity rate ratio R_2, noted as P_2 (Ntu_2, R_2) (see equation (4)).The overall heat transfer coefficient in Ntu_2 is also a function of A when the mass flowrate of geothermal water and R_2 are constants.

On rearranging the equation (2), the heat transfer area of terminal radiator A_r can be obtained:

$$A_r = \left(\frac{G_h C_p}{B_3 - B_1 Q_{id}} \right)^{1+\beta} \cdot \frac{Q_{id}}{\alpha} \tag{19}$$

The purpose is to find the minimum total initial capital cost of the plate heat exchanger and the terminal heat radiators. Suppose the costs of the plate heat exchanger and the radiators are linear functions of their heat transfer areas, respectively. This means larger the size of PHE, the cheaper per unit heat transfer area will be when a_1 and a_2 are positive.

$$COST_p = a_1 + b_1 A \tag{20}$$

$$COST_r = a_2 + b_2 A_r \tag{21}$$

So the optimum problem becomes that of finding the minimum total cost of $COST_t$.

$$COST_t = COST_p + COST_r = a_1 + a_2 + b_1 A + b_2 A_r \tag{22}$$

where $COST_p$ and $COST_r$ are the costs of PHEs and the terminal heat radiators, respectively. b_1 and b_2 are the costs of the PHE and the terminal heat radiators per unit heat transfer area, respectively. The running cost due to pumping the two fluids through the heat exchangers is not considered in the paper.

Since the estimation of heat load always corresponds to a specific outdoor temperature, the heat load Q_{id} and T_a should be given in pairs. The pair values can be given in two ways. One is as taking the peak loading heat (the maximum heat demand) and the lowest outdoor temperature, and the other using the time accumulated average heat load with the average outdoor temperature. However, Q_{id} can not exceed $C_h(T_h\text{-}T_a)$, the theoretical maximum heat that can be drawn from the geothermal water while its disposal temperature reaches the ambient temperature. In general the disposal temperature is around 40°C, in turn, the heat extracted from the geothermal water is $C_h(T_h\text{-}40)$, which should be a little bit more than the heat demand. This does not include the case where geothermal water after the PHE is used for other heating or bathing purposes. In those cases the outlet temperature of geothermal water is about 45°C instead of 40°C. When the average heat demand is accepted in the design of a heating scheme, a peak loading set such as the back-up boiler or the heat pump is required. But in any case, the optimum results should be valid because usually the peak heat demand can be achieved by regulating the inlet temperature of the fluid flow through the boiler or the heat pump. Generally, the time period of these peaking sets being loaded is short.

5. RESULTS AND DISCUSSION

The given calculation example was based on a real geothermal heating project in Tianjin. The wellhead temperature is 92.5°C. The production rate is about 150 m^3/h. Other parameters are shown in Table 1.

5.1 Comparison of Results for Direct and Indirect Heating Systems

With increasing outdoor temperature, the relative optimum thermal efficiency increases as shown in the Figure 1. The minimum required geothermal water flowrates for indirect heating system (optimum operation state) and for direct heating system with the increasing outdoor temperature is shown in the Figure 2. Curves for three heating load factors of 0.8, 1.0 (original design) and 1.2 have been shown in the figure. It can be seen that with increasing the outdoor temperature and decreasing the heating load factor, the relative optimum thermal efficiency increases. The conservative design or the lower heating load factor of an indirect heating project can make it to run at high relative thermal efficiency. However, the capital costs would be high. The conservative design means both the heat transfer areas of the PHEs and the heat radiators should be large. The benefit by increasing the heat transfer area of one side alone is limited. The relative thermal efficiency at different circulating water flowrates is shown in the Figure 3. The dashed line in the Figure 3 is the best operation state of the indirect heating system at different heating load factors. The over heating load factor urges the indirect heating system running at high circulating water flowrate, even with this the relative thermal efficiency is low.

5.2 The Best Operation of Indirect Heating Systems

The designed heat transfer areas of PHEs, A, and of the heat radiators, A_r, are 192 m^2 and 30000 m^2, respectively. As described in the section 2, when the geo-water flowrate is fixed, the best operation problem is to find the best heat capacity ratio R_2 or the circulating water flowrate when ignoring the specific heat capacity difference due to the change of temperature. Let the left and right sides of equation (18) to be denoted as H and F, respectively. In the Figure 4, H and F are plotted against R_2. The value of R_2 at which H and F are equal, i.e. the connect point of H and F is the best operation mode to be determined. Four kinds of situations are considered. One is the original designed situation shown as the point “A”. Point “B” is the situation of the power index, m, increasing, or the heat transfer performance of PHEs is better. Point “C” indicates the fouling heat resistance factor r being doubled. Point “D” means that double plate spacing of the PHEs is accepted at the cold circulating waterside. It can be seen that the higher the heat transfer performance of the PHE (the larger m) is the closer $(R_2)_{opt}$ to one, because the unbalance of the two sides flow results in an apparent sacrifice of decreasing the overall heat transfer coefficient of the PHE (point B). The increase of the fouling heat resistance urges the PHE running at high flowrate of circulating water, i.e. the lower $(R_2)_{opt}$ (point C). The PHE with the unequal cross-sectional area of flow channels has the lowest $(R_2)_{opt}$ in order to balance the film heat transfer coefficients of cold and hot sides (point D). The dashed lines in the Figure 4 are the values of *H* at constant *Ntu.*

The benefit from the optimal operation of a geothermal district heating system is not apparent as shown in the Table 2. There is not much difference for the heat supply Q_{id} by the system in a quite wide range of heat capacity ratio R_2. But it does not mean that the result is meaningless. On the contrary, since the heat load Q_{id} is always changing with the outdoor temperature, the flowrate of geothermal water should be regulated according to the rule proposed. Otherwise, the circulation pump is doing a part of useless work, which may not attract the user's attention. It is theoretically proved that the best flowrate ratio is about 0.8, which is commonly proposed in practice (Harrison et al 1990). The decreasing of R_2 is helpful to overcome the misdistribution problem of the fluid flow in different branches of a heating network, and only a slight decrease of the heat load Q_{id} has been obtained during this process. However, pumping power would be a problem if R_2 were too low. To increase R_2 is unfavorable for both the efficiency of heat utilization and the hydraulic stability of a geothermal heating system.

5.3 The Best Design of PHEs and Heat Radiators

First of all, the heating load is fixed as Q_{id} equals 6.0 MW. The conventional design method is to design PHEs and the heat radiators separately at given disposing temperature of geo-water. Since the supply and return temperatures cannot be given before the PHEs have been designed, the conventional design method is not suitable for this case.

Substituting Equation (19) into Equation (22), a new equation for the total cost is obtained, which has *A* as the only independent variable. The minimum value of $COST_t$ can be determined, therefore, by finding the solution of the differential equation with respect to *A* directly. The optimum value of A_{opt} is obtained by a numerical method. The equation is given by:

$$\frac{dP_2}{dA} = \frac{P_2^2\left(C_2 - \dfrac{Q_{id}}{P_2}\right)^{2+\beta}}{C_1^{1+\beta}(1+\beta)Q_{id}} \cdot \frac{b_1}{b_2} \qquad (13)$$

Figure 5 shows the best design values of *A* and A_r against the change of cost ratio per unit heat transfer area of the heat radiator to PHEs.

6. CONCLUSIONS

The best design and operation method of the indirect geothermal district heating systems and their performance comparison with the direct heating systems were proposed in the paper. The effects of some thermal parameters on the design and operation of low temperature geothermal district heating systems have been shown in formula and Figures, which is helpful for the engineers and technicians to understand the general behaviors of a low temperature geothermal heating system and its design.

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Table 1. Parameter set in the example

PHE arrangement	: 2 pass -2 pass countercurrent
PHE Heat performance	: $Nu = 0.35\,\mathrm{Re}^{0.61}\,\mathrm{Pr}^{0.3\,or\,0.4}$
Heat radiator performance	: $k = \alpha \Delta T^{\beta}$, α=2.05, β=0.35
Heat loss capacity of the building, C_vV	: 222000 W/°C
Flowrate of geo-water	: 150 m^3/hour
Well head temperature, T_h	: 92.5 °C
Outdoor temperature, T_a	: -9 °C
Heat demand (T_r = 18°C)	: 6.0 MW
Cost of PHE per m^2	: 250 USD (Titanium)
Cost of radiator per m^2	: 7.5 USD (Cast iron)

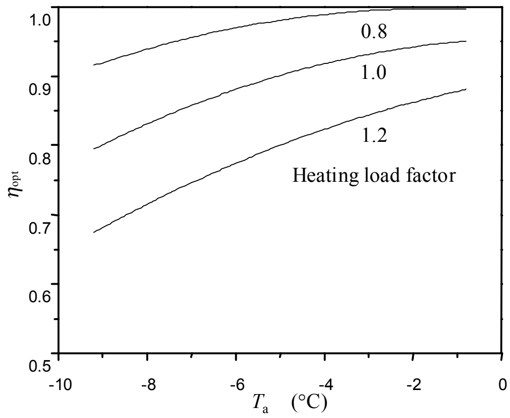

Figure 1. Relative optimum thermal efficiency against the outdoor temperature at different heating load factors

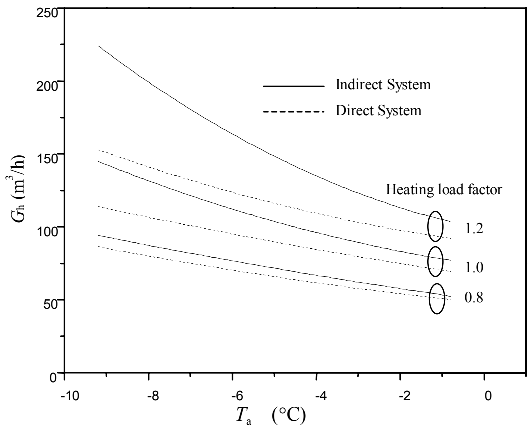

Figure 2. The minimum required geo-water flowrates for indirect heating system (optimum operation state) vs. the outdoor temperature

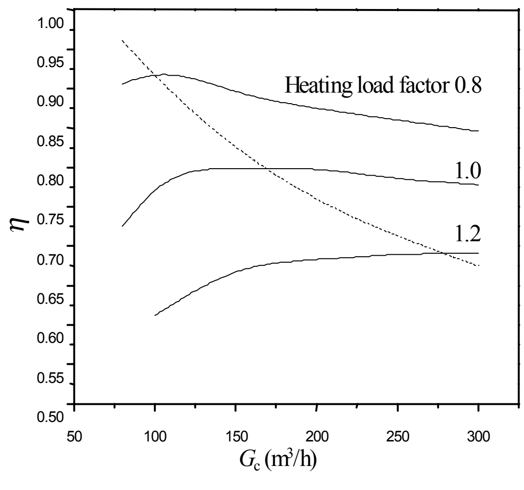

Figure 3. The relative thermal efficiency at different circulating water flowrates

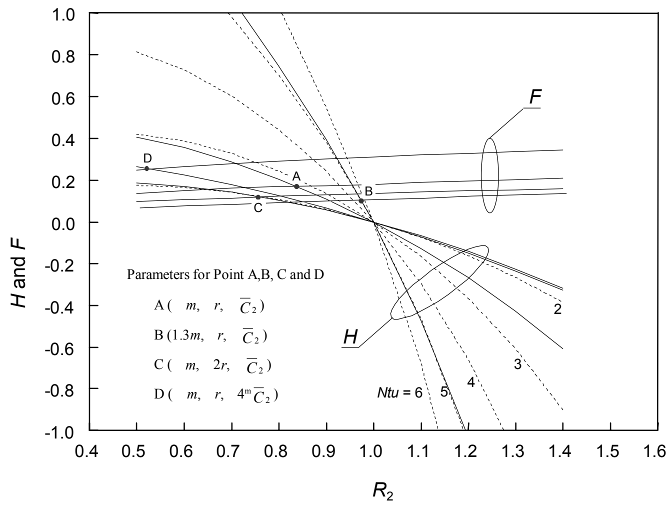

Figure 4. The left side H and right side F of Equation (18) against the heat capacity ratio R_2

Table 2. Calculation results from equation (18)

R_2	Ntu_2	H	F	Q_{id} (MW)
0.5	2.9562	0.4060	0.1379	7.0401
0.6	2.8824	0.3551	0.1502	7.0807
0.7	2.8166	0.2878	0.1611	7.1074
0.8	2.7571	0.2058	0.1709	7.1201
0.835*	2.7375	0.1739	0.1741	7.1277
0.9	2.7027	0.1100	0.1799	7.1188
1.0	2.6526	0.0000	0.1882	7.1040
1.1	2.6061	-0.1249	0.1958	7.0765

* The optimum point "A" in Figure 4

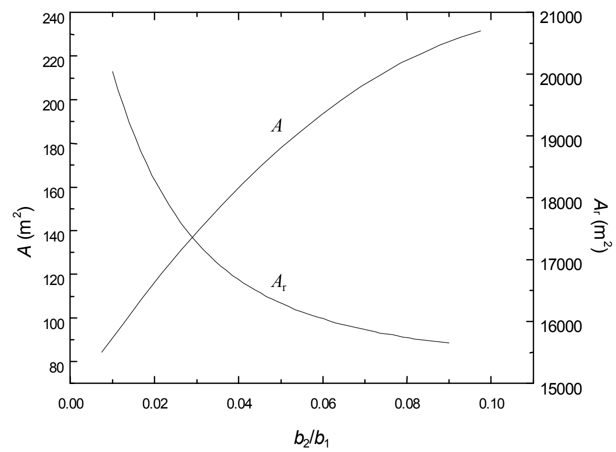

Figure 5. Optimum A and A_r with the changing of cost ratio of radiators to PHEs per unit heat transfer area b_2/b_1 (at R_2=0.8)