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CLOSED-LOOP HEAT EXCHANGER FOR GROUND –COUPLED HEAT PUMPS

G. DRAGOMIR[1] I. BOIAN V. CIOFOAIA

Abstract: *Hydraulic imbalances tend to affect the thermal behavior of borehole heat exchangers used in case of ground coupled heat pumps, GCHP. The paper is focused on the influence of the layout of the borehole field and on the necessity of the balancing valves on the thermal efficiency of a GCHP A central loop and a subcentral loop have been analyzed in order to find out the optimal connection with lowest costs and improved efficiency of the GCHP system .*

Key words: *Heat pump systems, borehole heat exchangers balancing.*

1. Introduction

Optimal operation of Ground Coupled Heat Pumps (GCHP) used for space heating and for air conditioning depends on the good correlation between the heat pump and the heat requirements of the building on one side, but on the heat pump and the ground heat exchanger on the other side, too. Heat pumps offer the same comfort level as boilers based on fossil fuels (solid, liquid and gaseous) and electrical driven air conditioning systems, but being more sustainable and having less green house gasses emissions become more attractive. [2]. Considering the fact that ground is keeping its temperature almost constant over the entire heating season this issue is very advantageous when soil is compared with other source for space heating/cooling.

High capacity GCHP operated in urban agglomeration usually relies on vertical ground heat exchangers (VGHE). These ground loops, circulated by an antifreeze fluid, are U-shaped pipe, or double U and sometimes concentric tubes, as shown in Figure 1. The loop is inserted in a borehole 50 to 100 m deep and having a diameter of 100…150 mm. After the loop insertion the borehole is backfilled with bentonite grout (entire deepness or only the upper to avoid surface water penetration and potential groundwater contamination.

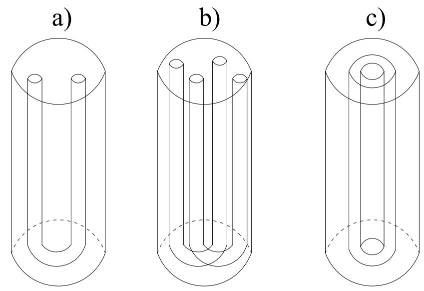

Fig.1. a) *U-tube; b) double U-tube; c) Concentric tubes.*

When designing a GCHP system one of the points of focus is to have an optimal heat exchange with the ground. It is important to know if the flow-balancing is necessary in case of a ground field of

[1] Building Facilities Dept., Civil Engineering Faculty, *Transilvania* University of Braşov.

boreholes consisting of dozens of loop subheaders. The paper deals with the impact of the hydraulic imbalance on the thermal performance of a heat pump due to different behavior of different loops in the ground.

2. Ground heat exchanger layout

The required length for heating can be calculated as

$$L_h = \frac{q_a \cdot R_{ga} + \left(q_{lh} - 3.41 \cdot W_h\right)}{T_g - \frac{T_{ai} + T_{ae}}{2} - T_p} \cdot \beta$$

$$\beta = \left(R_p + PLF_m \cdot R_{gm} + R_{gd} \cdot F_{sc}\right)$$

where

L_{sc} short circuit heat loss factor, m
q_a net annual average heat transfer to the ground, kW
R_{ga} effective thermal resistance of the ground, annual pulse, (mK)/kW
q_{lh} building design heating block load, kW
W_h power input at design heating load, kW
R_p thermal resistance of pipe, (mK)/kW
PLF_m the part load factor during design month
R_{gm} effective thermal resistance of ground, monthly pulse, (mK)/kW
R_{gd} is the effective thermal resistance of ground, daily pulse, (mK)/kW
F_{sc} short circuit heat loss factor
T_g undisturbed ground temperature, ^{o}C
T_{wi} liquid temperature at heat pump inlet, ^{o}C
T_{wo} liquid temperature at heat pump outlet, ^{o}C
T_p temperature penalty for interference of adjacent bores, ^{o}C.

After determining the total length of the ground heat exchanger L_h the number of boreholes and its layout must be specified.

Two patterns of borehole heat exchangers have been analyzed, both of them consisting of 40 boreholes grouped in a field. The heat exchangers are double U-bend tubes, high-density polyethylene (HDPE, thermally fused), De 32 mm, 100 m in length per parallel loop, being inserted in a 115 mm diameter borehole. The first field is a central loop having a single central header, and is shown in Figure 2.

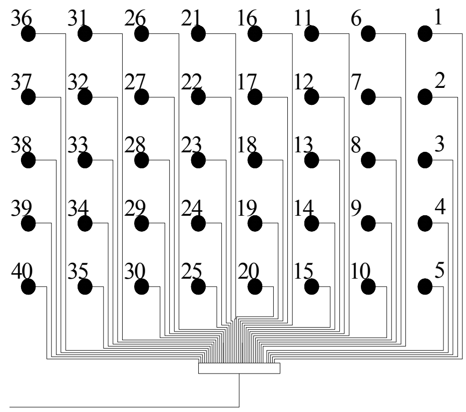

Fig.2. *The borefield layout organized as a central loop (all the ground heat exchangers are connected to a single header).*

The second field, organized as subcentral GCHP loop, has four sub-fields with the afferent four subheaders, as shown in Figure 3. In both cases the separation distance between adjacent boreholes is dictated by two reasons: the required land area and the necessary bore length.

The antifreeze fluid is circulated trough the loop by a pump with a flow of 17.20 l/s for which it delivers a head 8.50 m in case of the central loop and 10.8 m for the subcentral version. The average flow rate for each borehole heat exchanger is 0.43 l/s.

3. Mathematical formulation

The flow through the loop nearest the building will differ from the flow through the most remote loop as a result of hydraulic imbalances caused by unequal head losses.

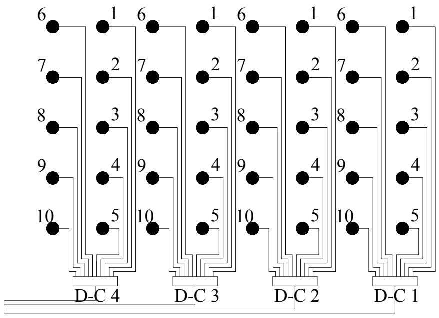

Fig.3. *Subcentral GCHP loop: Four subheaders devide the field of borehole heat exchangers.*

Such differences in flow rates will impact on heat transfer between the ground and the fluid circulating from/to the heat pump.

The first step is to find the correlation of flow versus head loss for the two loops: the nearest and the most remote one.

The flow rate through the pipe, $\dot{V}$ was calculated as

$$\dot{V} = \frac{\sqrt{\dfrac{2 \cdot D \cdot \Delta p}{\lambda \cdot L \cdot \rho}} \cdot \pi \cdot D^2}{4 \cdot L} \tag{1}$$

Where:

Δp - Pressure drop [Pa]

L- Borehole or pipe length [m]

ρ -density [kg/m^3]

The pressure drop of loops is determined then for the longest U-tube and header path, and for the shortest path, respectively.

For a good heat transfer the liquid velocity must be sufficient to avoid laminar flow especially at full-load conditions. But in case of part load operation a variable flow is necessary and a laminar flow could occur, which is tolerable. In such a case (Re <2300) the friction factor λ is given by

$$\lambda = \frac{\mathrm{Re}}{64}. \tag{2}$$

For current turbulent flow, Re>2300, the friction factor λ was calculated as

$$\lambda = \frac{1}{\left(0{,}79 \cdot \ln \mathrm{Re} - 1{,}64\right)^2}. \tag{3}$$

The Reynolds number *Re* was calculated as expressed in equation (4), considering the velocity *w* and the kinematic viscosity of the fluid ν flowing through the pipe with an equivalent inner diameter *D*

$$\mathrm{Re} = \frac{w \cdot D}{\nu}. \tag{4}$$

The heat transfer rate $\dot{Q}$ was evaluated as

$$\dot{Q} = \dot{m} \cdot c_p \cdot \varepsilon \cdot \left(T_{ground} - T_{water}^{ent\ gr\ coil}\right) \tag{5}$$

where $\dot{m}$ is the mass flow o the fluid and c_p its specific heat capacity. T_{ground} is the temperature of the ground (undisturbed) and $T_{water}^{ent\ gr\ coil}$ represents the temperature of water entering the ground coil. The heat exchanger effectiveness *ε* was considered as in case of an evaporator (the deep ground temperature is practically constant along the heating season)

$$\varepsilon = 1 - \exp(NTU). \tag{7}$$

The number of heat transfer units *NTU* was calculated using the net thermal resistance of the bore R_b, the total length of the pipe inserted in it (double of the borehole deepness) L [ ]

$$NTU = \frac{L}{R_b \cdot \dot{m} \cdot c_p}. \tag{8}$$

The net bore resistance R_b is the sum of the pipe R_p and the backfill R_{bf} resistances

$$R_b = R_p + R_{bf} \tag{9}$$

The thermal resistance of the pipe R_p, including the inside film coefficient can be estimated from *Table 1* [1]

Pipe thermal resistance, R_p Table 1

Re	R_p, mK/W	
	Φ25	Φ32
Turbulent flow	0,0433	0,0537
2300...4000	0,0479	0,0583
1000...2300	0,0577	0,0681

Finally, the thermal resistance of the borehole backfill R_{bf} was calculated, considering the U-tubes spaced between the outer wall and touching, as

$$R_{bf} = \frac{1}{\lambda_{bf} \cdot S_b}. \tag{10}$$

Where λ_{bf} thermal conductivity of the borehole [W/m K], S_b results from the borehole diameter D_b and the pipe diameter D_p

$$S_b = 17{,}44 \cdot \left(\frac{D_b}{D_p} \right)^{0{,}6052} \tag{11}$$

4. Results and discussion

Due to different length of the loops the flow through the nearest and the most remote heat exchanger are not equal. Table 2 presents an evaluation of flow rates for three different separation distances between the boreholes

Flow and heat transfer rates calculated for a central loop borefield Table 2

Borehole No.	Distance between borehole heat exchangers											
	6 m				10 m				15 m			
	L	Δp	$\dot{V}$	Q	L	Δp	$\dot{V}$	Q	L	Δp	$\dot{V}$	Q
	m	bar	l/s	kW	m	bar	l/s	kW	m	bar	l/s	kW
1	3	4	5	6	3	4	5	6	3	4	5	6
1	308	0.64	0.393	4.98	372	7.38	0.38	4.89	447	8.32	0.37	4.77
2	296	0.64	0.402	5.05	352	7.38	0.39	4.99	417	8.32	0.38	4.90
3	284	0.64	0.411	5.13	332	7.38	0.41	5.10	387	8.32	0.4	5.04
4	272	0.64	0.421	5.21	312	7.38	0.42	5.22	357	8.32	0.42	5.20
5	260	0.64	0.432	5.30	292	7.38	0.44	5.34	327	8.32	0.44	5.36
6	296	0.64	0.402	5.05	352	7.38	0.39	4.99	417	8.32	0.38	4.90
7	284	0.64	0.411	5.13	332	7.38	0.41	5.10	387	8.32	0.4	5.04
8	272	0.64	0.421	5.21	312	7.38	0.42	5.22	357	8.32	0.42	5.20
9	260	0.64	0.432	5.30	292	7.38	0.44	5.34	327	8.32	0.44	5.36
10	248	0.64	0.443	5.39	272	7.38	0.45	5.48	297	8.32	0.46	5.54
11	284	0.64	0.411	5.13	332	7.38	0.41	5.10	387	8.32	0.4	5.04
12	272	0.64	0.421	5.21	312	7.38	0.42	5.22	357	8.32	0.42	5.20
13	260	0.64	0.432	5.30	292	7.38	0.44	5.34	327	8.32	0.44	5.36
14	248	0.64	0.443	5.39	272	7.38	0.45	5.48	297	8.32	0.46	5.54
15	236	0.64	0.455	5.48	252	7.38	0.47	5.62	267	8.32	0.49	5.74
16	272	0.64	0.421	5.21	312	7.38	0.42	5.22	357	8.32	0.42	5.20
17	260	0.64	0.432	5.30	292	7.38	0.44	5.34	327	8.32	0.44	5.36
18	248	0.64	0.443	5.39	272	7.38	0.45	5.48	297	8.32	0.46	5.54
19	236	0.64	0.455	5.48	260	7.38	0.47	5.56	275	8.32	0.48	5.69
20	224	0.64	0.468	5.58	240	7.38	0.49	5.71	245	8.32	0.52	5.90

(6m, 10m and 15m). A span of ±6 % over the average flow results in case of 6 m separation distance between the boreholes and of ±20 % for a 15 m distance. Only 20 boreholes have been considered because of symmetry.

For the subcentral loop configuration a similar calculation was done and results for the four subheaders and for the shortest (248m) and the longest path (308m), versus the average one (272m) are presented in Figure 4. This layout presents less than ±7% flow rate differences between the longest and the shortest path of the same header and less than ±15% when considering the extreme values for a distance between boreholes of 6m (Header No.1 vs. Header No.4). For the15m separation distance the flow rate imbalance is about ±10% for the the same header and less than ±29% between the extreme values, i.e. Header No.1 @ Borehole No.1 vs. Header No.4 @ Borehole No.10.

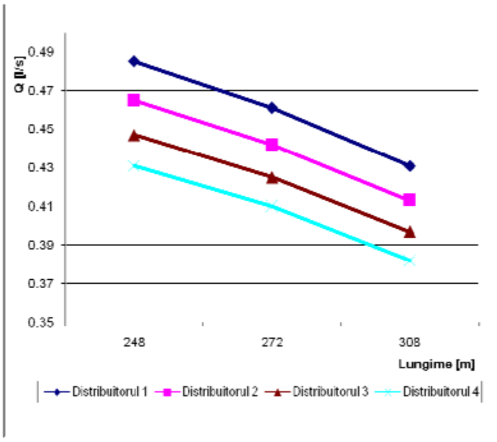

Fig.4. *The flow rate as a function the length of the path for the subcentral loop configuration.*

As a result of hydraulic imbalance i.e. different flow rates specific for distinct heat exchangers a thermal imbalance will occur.

For the central loop differences of ±6% exists between the heat rates (Borehole No.1 vs. Borehole No. 20) in case of a separation distance of 6 m. The 15m separation is leading to higher heat rates differences of less than 12%, as shown in Figure 5.

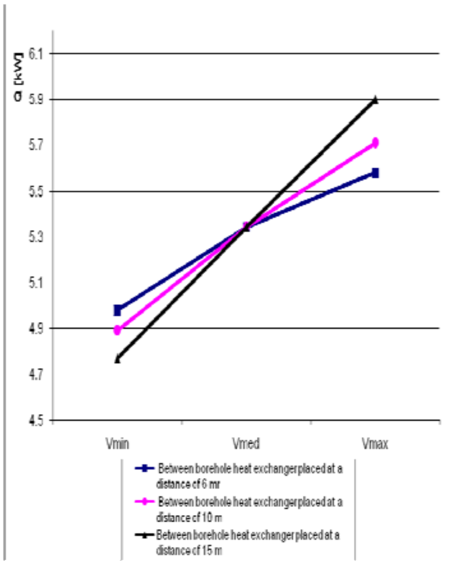

Fig.5. *Heat rate imbalance for a central loop configuration and for 6m, 10m and 15m separation distance of boreholes.*

V_{min} is the lowest fluid flow rate being specific for the most remote heat exchanger. V_{med} is the average flow rate and V_{max} is the maximal one existing for the nearest heat exchanger.

The subcentral loop presents lower thermal imbalance of the heat exchangers i.e. less than ±4% for every header and for 6 m separation distance between boreholes. For higher separation distances, as 15m, thermal imbalances became higher too, i.e. 7%, as shown in Figure 6.

5. Conclusions

A comparison between a central and a subcentral loop has been carried out in order to estimate the advantage of every configuration. When the size of the ground loop exceeds 35 to 70 kW consideration should be given to arranging the ground loop into subgroups.

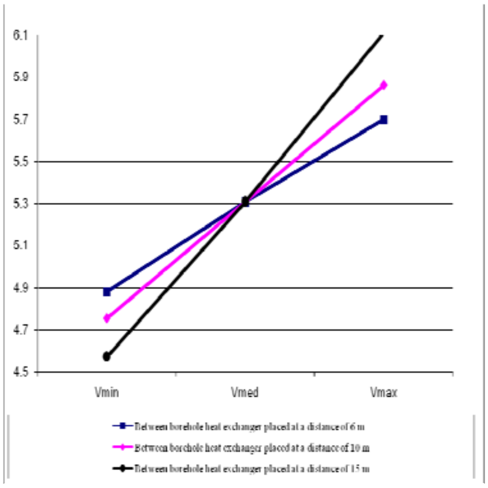

Fig.6. *Heat rate imbalance for a subcentral loop configuration and for 6m, 10m and 15m separation distance of boreholes*.

The headers would feed a close header placed in the center of a subgroup of U-tubes.

Water loop designers are usually very concerned about flow balancing. However, the nature of heat transfer in the ground loops can tolerate minor imbalances without affecting overall performance. A 15% imbalance among vertical ground loops is acceptable [3]. This is possible because:

- large differences in the internal heat transfer coefficients have minimal effect on overall heat transfer coefficients
- the effectiveness of ground heat exchangers is poor, so a 15% change in approach temperature has little effect on effectiveness and heat transfer.

Therefore, exactly equal parallel loops, reveres-return headers, or flow-balancing valves are not usually required in the ground loop. Sufficient flow balancing in the ground loop can be accomplished by adjusting the pipe length and/or header diameters. The subheader isolation valves can be placed in an interior equipment room or in a below-grade vault near the centre of the ground loop. This will allow purging the installation, which requires a 0.6 m/s velocity. Depending on contractor, the subheaders can be located either in the building or in the below-grade

References

1. *Kavanaugh, S., P., Rafferty K. Ground –Source Heat Pumps.ASHRAE,1997.*
2. *Miaomiao H., - Numerical Modelling of Geothermal Borehole Heat Exchanger Systems Institute of Energy and Sustainable Development De Montfort University, Leicester, UK,2012*
3. **** ASHRAE Handbook. HVAC Applications. SI Edition, 2011.*