ADVANCES IN VERTICAL DX GROUND-SOURCE HEAT PUMPS DESIGN AND CONTROL

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Abstract: Ground-source heat pumps with direct expansion (DX) heat exchangers provide several well-known advantages that may help increase the acceptance of geothermal technology for residential applications. This paper describes a residential ground-source heat pump prototype with vertical DX heat exchanger and improved refrigerating circuits and controls. A simplified design approach for vertical DX ground-coupled heat exchangers is proposed. The paper also investigates some operational issues in cold climates, such as oil return to the compressor and refrigerant management during system start-ups and mode switching. Initial installation errors produced undesirable phenomena, such as excessive soil freezing and the collapsing of vertical ground tubes during long continuous operating cycles in the heating mode. Methods to avoid such incidents are finally suggested.

Key Words: heat pumps, direct expansion ground heat exchangers, energy efficiency

1. INTRODUCTION

The costs of residential ground-source heat pumps (GSHP) with secondary fluid loops, including the cost of borehole drilling and grouting, vertical ground coils, geothermal fluids and steel sleeves, are relatively high. Even if partially compensated by higher evaporating temperatures and the elimination of outdoor fans, the additional electrical energy consumption required for secondary fluid pumping affects the seasonal performance factors (SPF) of such systems compared to other systems, such as air-to-air/water or ground-coupled direct expansion (DX) heat pumps. In the case of the latter, the traditional outdoor heat exchanger (evaporator/condenser) is installed directly underground, horizontally or vertically. This particularity presents a number of advantages, such as eliminating secondary fluids and the electrical energy consumption of circulating pumps. The DX systems may have various configurations and control strategies, and may reduce the borehole depth and costs compared to conventional indirect systems. With adequate ground properties, reversible DX heat pumps may provide air cooling and dehumidification in the summer. The heat flux recovered from, or rejected to the ground, is higher because of higher temperature differences between the soil and the refrigerant saturated temperatures. As a consequence, the SPFs of DX ground-coupled heat pumps are potentially higher compared to those of indirect GSHP systems. In the past, because of refrigerant build-up in vertical DX ground heat exchangers, the compressors were unable to move the refrigerant through the systems properly. Other shortcomings included inadequate oil return to the compressor particularly in the heating mode, inadequate evaporator length and spacing for properly extracting heat from the ground resulting in low capacity and low efficiency of the systems, lack of proper means to store the additional refrigerant required during the cooling operation, but not needed in the heating mode, and insufficient heating capacity during the coldest weather. Today, the DX ground-source systems are in a more advanced development stage and commercially available. The manufacturers claim having solved the majority of critical issues, but some questions and potential problems still remain, such as long-term reliability and seasonal performance especially in cold climates. During the last ten years, technology advances were achieved in Canada. One of these provides optimal heat exchange in the heating mode through three parallel vertical ground loops which can be individually operated to provide staged operation in the cooling mode (Kaye 1993). Two of these loops can be cut off from the system and pump down the refrigerant in the cooling mode.

2. GROUND HEAT EXCHANGER

2.1 Configuration

An original residential DX ground heat pump prototype with vertical ground heat exchanger has been developed for residential applications in cold climate (Minea 2005). The ground heat exchanger consists of two vertical copper U-tubes with ½" ($12.7 \cdot 10^{-3}$ m) nominal diameter (d_0) and vertical length l_0 (Figure 1). The thin-wall, soft copper U-tubes act as an evaporator in the heating mode and as a condenser in the cooling mode. They are coated with thin plastic films to avoid corrosion and installed inside two boreholes located at a 5 m distance one from each other, and at about 5 m from the building. The vertical U-tubes are connected to the building by horizontal runs buried at a 1.3 m depth. The tube size, much smaller than that of the high density polyethylene tubes diameter (1 1/4") used in traditionally secondary loop systems, has been chosen in order to achieve small refrigerant pressure drops and velocities higher than 10 m/s to provide proper oil return to the compressor. It can be noted that, when designed as evaporators, it is possible to size the vertical U-tubes so as to minimize or eliminate pressure losses. Effectively, the refrigerant pressure losses during the convective vaporization inside the down comers may be compensated by the pressure wins caused by gravity. This is why in most Canadian systems the nominal diameter of down comers and risers is 3/8" (9.5 mm) and ½" (12.7 mm) respectively. In this study, the working fluid is the refrigerant HFC-407C, a long-term, zero depletion zeotropic mixture of three pure HFCs. To limit the risk of underground refrigerant leakage, thermal fusions have been permitted on the site only for soldering the bottom U-bends.

2.2 Simplified design approach

In the simplified proposed design procedure, the actual configuration with two vertical U-tubes (Figure 1) is replaced by one U-tube with equivalent nominal diameter $d_{eq,i} = \sqrt{2}d_i$ (0.0179 m) and equivalent vertical length $L_{eq,vert}$ (Figure 2a). The length of the bottom bend of the equivalent U-tube is ignored with respect to heat transfer, simply because it doesn't actually exist. The horizontal runs have a length ($L_{eq,horiz}$) estimated at 15 m so that the total length of the equivalent DX ground heat exchanger (vertical plus horizontal) is (Figure 2b):

$$L^{tot}_{eq} = 2L_{eq,\,vert} + L_{eq,\,horiz} = 2L_{eq,\,vert} + 15\ m \quad (1)$$

Let's consider placing around the vertical tubes 1 m radius homogeneous soil cylinders with an average thermal conductivity of 1.17 W/mK. The surface temperature of these soil cylinders is deemed to be constant at infinite (r_∞ = 1 m distance from the refrigerant tubes) (T_∞). In the heat pump *heating mode*, the soil temperature T_∞ is deemed to be constant at about 8°C i.e. the site average ground temperature. Other design assumptions include : (i) the effect of gravity forces during the refrigerant down and up flow is ignored in both the *heating* (convective vaporization) and *cooling* (convective condensation) modes; (ii) saturated temperatures inside the equivalent vertical tubes are constant during both two-phase heat transfer processes; (iii) single-phase vapor superheating in the heating mode and liquid sub-cooling in the cooling mode are ignored; (iv) the thermal resistance of copper tubes is ignored.

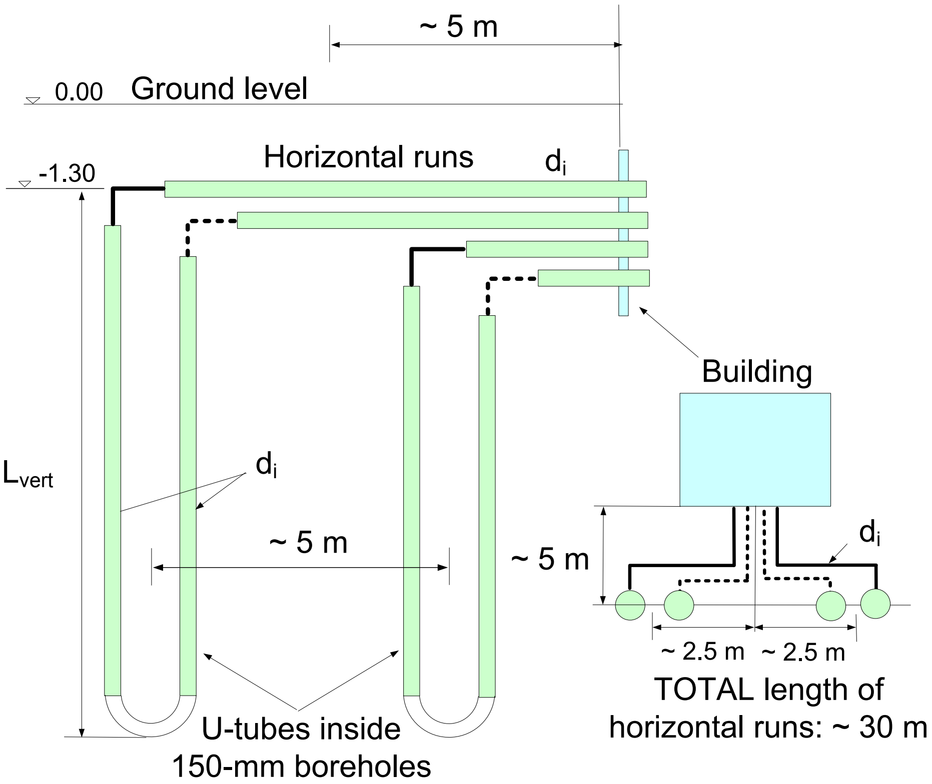

Figure 1: Actual configuration of the vertical DX heat exchanger

Even if the thermal conductivity of these tubes (380 W/mK) is much higher compared to the thermal conductivity of the high density polyethylene tubes used in indirect systems, it has no significant impact on the global soil-to-refrigerant heat transfer coefficient; (v) on the surface of vertical and horizontal refrigerant tubes, the heat flux density through the soil cylinders equals the heat flux density of both convective vaporization (heating mode) and convective condensation (cooling mode) processes. In the *heating mode*, the convective vaporization of the low pressure, low temperature refrigerant takes place inside the equivalent vertical U-tube. Under Fourier's hypothesis, the density of the heat flux through the soil depends only on the cylinder radius (Figure 2c):

$$\dot{q}_{soil,\,heat} = -kgradT = \frac{k_s}{r}\frac{|T_\infty - T_w|}{\ln\frac{r_\infty}{r_{eq,w}}} = \frac{6.16}{r}(W/m^2) \qquad (2)$$

In equation 2, the average temperature of the external surface of the U-tube wall (T_w) is -16°C (257°K) and the soil temperature at infinite (T_∞) is constant at 8°C (281°K). For a 1 m length of soil cylinders, the *theoretical* unitary heat flux (per-unit of length) will be:

$$\dot{Q}_{soil,1} = 2\pi r(\frac{6.16}{r}) = 38.7W/m \qquad (3)$$

Consequently, the heat flux transferred to the refrigerant per-unit of the equivalent U-tube area will be: $\dot{q}_{heat} = 742W/m^2$ (Table 1).

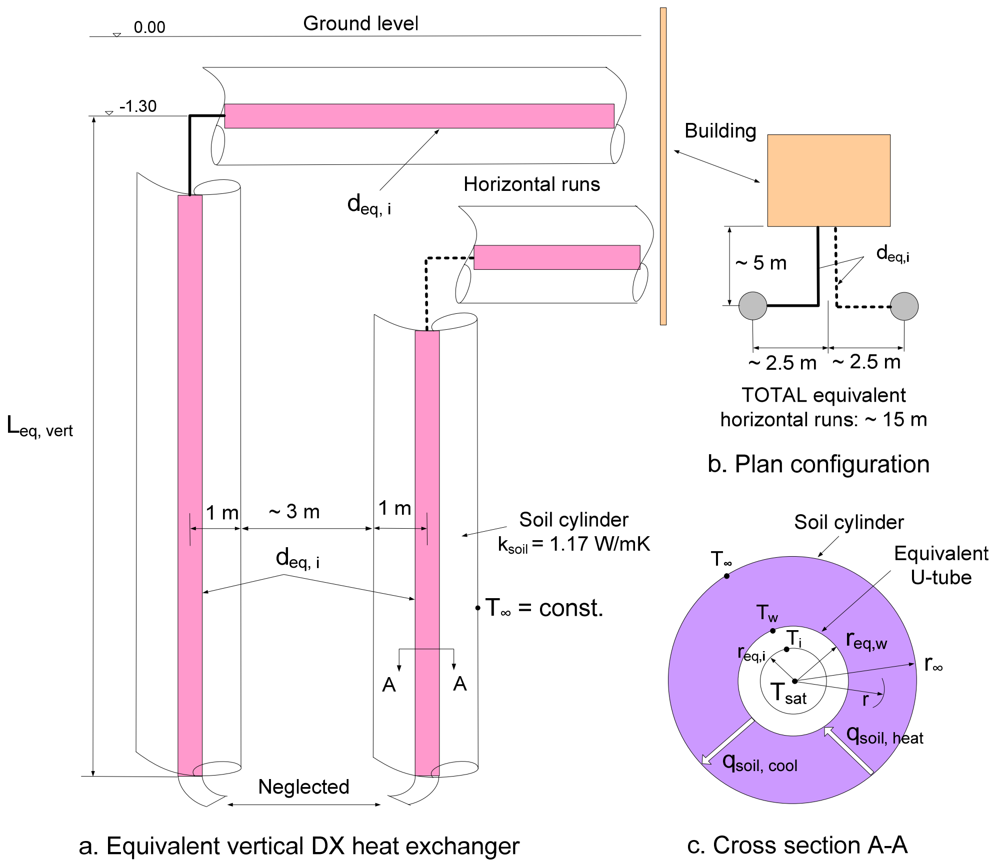

Figure 2: Equivalent vertical DX heat exchanger

The convective vaporization heat transfer coefficient on the refrigerant side is calculated using the following correlation (Ciconkov 2001):

$$h_{cv} = C\frac{G^{0.1}\,\dot{q}_{heat}^{\,0.7}}{d^{0.5}_{eq,i}} = 928 W/m^2K \qquad (4)$$

where G (123 kg/m^{2}s) is the refrigerant mass velocity through the equivalent vertical U-tube defined as “flow rate/flow area” and C, a constant depending on the refrigerant thermodynamic properties at convective vaporization temperature of -17°C:

$$C = f(k_l, h_{lv}, \rho_l, \rho_v, g, T_{cv},\ldots\ldots) = 0.786 \qquad (5)$$

The total length of the equivalent vertical and horizontal ground refrigerant tubes required in the heating mode may be calculated using the following equation:

$$L^{tot}_{eq,heat} = \frac{\dot{Q}_{extr,heat}}{h_{cv}\pi d_{eq,i}(T_i - T_{sat})} = 175m \qquad (6)$$

where $\dot{Q}_{extr,heat}$ is the heat pump designed heat transfer rate of extraction (7 300 W). The *actual* heat flux absorbed by unitary tube length will be 41.7 W/m i.e. about 40 to 45% higher compared to that of indirect systems with secondary fluid closed loops. If 15 m represent the length of the *horizontal* runs, the total required length of the *vertical* equivalent U-tube for the heating mode is about 160 m.

Table 1: Design parameters for the heat pump heating mode

Parameter	Unit	Symbol	Value
Bubble point temperature	°C	-	-23
Dew point temperature	°C	-	-17
Temperature of the external wall of the equivalent U-tube	°C	T_w	-16
Soil temperature at infinite	°C	T_∞	8
Heat flux per unit length of equivalent U-tube	W/m	$\dot{Q}_{heat,1}$	41.7
Heat flux per unit area of equivalent U-tube	W/m^2	$\dot{q}_{heat}$	742
Refrigerant mass velocity through the equivalent U-tube	kg/m^{2}s	G	123
Overall heat transfer coefficient – convective vaporization	W/m^{2}K	h_{cv}	928
Total required length of ground DX heat exchanger	m	$L^{tot}_{eq,\ heat}$	175
Length of equivalent horizontal runs	m	-	15
Required length of equivalent vertical U-tube	m	-	160

In the *cooling mode*, convective condensation takes place inside the equivalent vertical U-tube at an average condensing temperature (T_{cd}) of 34°C (Table 2). The convective condensation heat transfer coefficient is given by the following equation:

$$h_{cc} = Nu \frac{k_l}{d_{eq,i}} = 3332 W/m^2 K \qquad (7)$$

where *Nu* is the Nusselt number (Incropera 2002):

$$Nu = (0.026).\mathrm{Pr}_l^{0.33}\left[\mathrm{Re}_l\left(\frac{\rho_l}{\rho_v}\right)^{0.5} + \mathrm{Re}_l\right]^{0.8} = 702 \qquad (8)$$

Re, the Reynolds number:

$$\mathrm{Re}_l = \frac{\omega_l . d_{eq,i}}{\nu_l} = 16188 > 5000 \qquad (9)$$

and *Pr*, the liquid Prandtl number:

$$\mathrm{Pr}_l = \frac{c_l . \eta_l}{k_l} = 2.56 \qquad (10)$$

The heat flux density and the unitary heat flux for1 m of equivalent U-tube are respectively $\dot{q}_{cc} = h_{cc}.\Delta T_{cc} = 833 W/m^2$ (where ΔT_{cc} is the difference between the constant condensing temperature and the interior wall temperature) and 56.2 W/m. The total required length of the refrigerant ground tubes in the *cooling mode* may be calculated using the following equation:

$$L^{tot}{}_{eq,cool} = \frac{\dot{Q}_{rej,cool}}{h_{cc}\pi d_{eq,i}(T_{cd} - T_i)} = 142 m \qquad (11)$$

where $\dot{Q}_{rej,cool}$ is the heat pump heat transfer rate of rejection (8 000 W). If the length of the horizontal runs is 15 m, the total required length of the vertical equivalent U-tube for the cooling mode will be of 127 m (Table 2).

Table 2: Design parameters for the heat pump cooling mode

Parameter	Unit	Symbol	Value
Bubble point temperature	°C	-	31.2
Dew point temperature	°C	-	36.5
Average condensing temperature	°C	T_{cd}	34
Overall heat transfer coefficient - convective condensation	W/m^2K	h_{cc}	3 332
Heat flux per unit area of equivalent U-tube	W/m^2	$\dot{q}_{cool}$	833
Heat flux per unit length of equivalent U-tube	W/m	$\dot{q}_{c,l=1}$	56.2
Total required length of ground DX heat exchanger	m	$L^{tot}_{eq,\ cool}$	142
Length of equivalent horizontal runs	m	-	15
Required length of equivalent vertical U-tube	m	-	127

3. HEAT PUMP

3.1 Refrigerant circuits

Based on the simplified design approach, a ground-source heat pump prototype with vertical DX heat exchanger and 8 kW nominal cooling capacity was developed and tested (Picture 1). Two 37.5 m deep boreholes of 150 mm diameter each were drilled and grouted with silica sand. The total length of the two vertical boreholes (75 m) is at least 40% lower compared to that of Canadian indirect systems with secondary fluid loops. The ground structure contained clayey soil on the first 12.5 m followed by heavy damp soil and dense rock. Steel sleeves were installed on the first 12.5 m of both boreholes to protect the ground heat exchanger tubes. A distinctive feature of the site was the presence of groundwater under relatively high pressure, especially in spring and fall. To prevent the groundwater from flowing up and flooding the land, the steel sleeves were capped at their upper end. This, as well as the borehole grouting with silica sand, finally proved to be installation errors, even though they were technically justifiable at the beginning. The refrigerant circuits (Figure 3) contain a number of original features aiming at improving the heat pump operation and control. Solenoid valves 1 and 2 and check-valves 3 and 4 allow the refrigerant to flow down (in the heating mode) and up (in the cooling mode) through the vertical U-tubes. The refrigerant flow direction changes according to the thermostat heating or cooling demands by switching reversing valve 5. Thermostatic expansion valves 6 and 8, based on the refrigerant actual pressures and temperatures at the evaporator outlets in heating and cooling modes, are used as refrigerant control devices. Check valves 7 and 9, installed in parallel, as well as bi-directional liquid receiver 10 let the refrigerant flow in both directions according to the operating mode. In the *heating mode*, the refrigerant enters the vertical U-tubes in a saturated two-phase state at low pressure. Convective vaporization takes place with the refrigerant first flowing down and then flowing up through the vertical U-tubes, and forced convective vapor superheating occurs at the end of the vertical tubes. The low pressure superheated vapor leaving the ground vertical U-tubes (evaporator) flows through solenoid valve 11 (open) and unidirectional pressure regulating valve 12, check-valve 13, reversible valve 5 and the suction accumulator, and enters the suction line of the compressor C. The sub-cooled liquid from the condenser (fan coil) flows through the check valve 7, bidirectional liquid receiver 10, solenoid valve 14 (open) and expansion valve 8 prior entering the ground vertical U-tubes at low pressure (valves 1 and 2 are open). In *cooling mode*, the refrigerant flow is reversed. The high pressure superheated refrigerant vapor from the compressor C flows through the check-valve 15 and enters the vertical U-tubes where it first flows down and then up. Superheated vapor cooling by forced convection followed by convective condensation takes place, and the heat is transferred to the ground. The high pressure sub-cooled liquid flows through the check-valves 3, 4 and 9, the bidirectional liquid receiver 10, the expansion valve 6 and, finally, evaporates at low pressure inside the fan coil (evaporator).

Picture 1: View of the heat pump prototype with vertical ground-coupled DX heat exchanger

3.2 Short cycles in heating mode

Experiences in *heating mode* with relatively short on/off cycles allowed the soil to periodically recover its thermal capacity as a heat source. During such relatively short cycles the heat pump compressor operated with average input powers of 1.6 kW and compression ratios of around 5.6. Table 3 presents other average thermodynamic parameters, such as suction and discharge pressures, and DEW line evaporating/condensing temperatures. The heat pump supplied hot air at around 30°C, and no undesired phenomenon, such as oil or liquid trapping inside the vertical tubes of the ground DX heat exchanger, was observed.

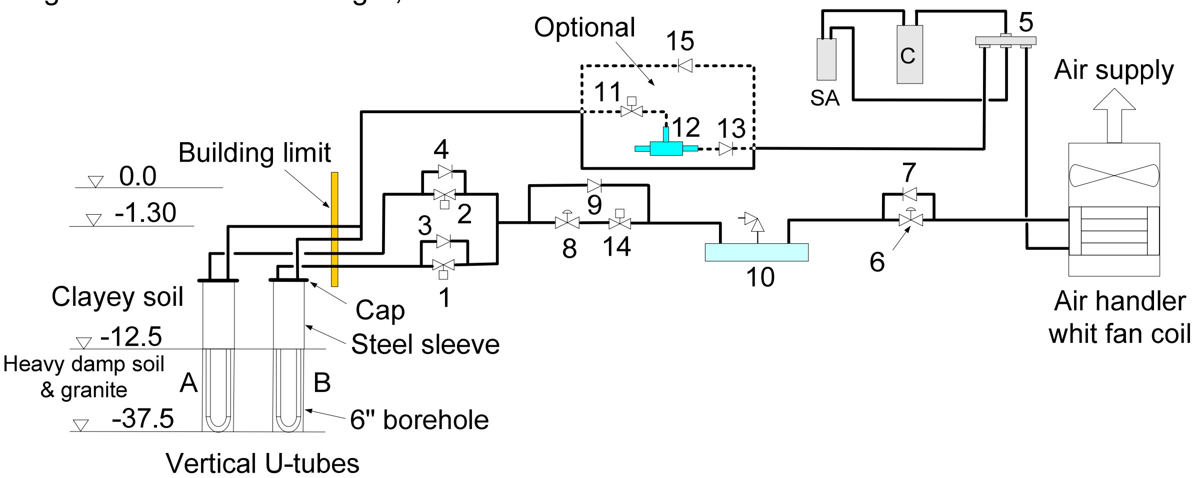

Figure 3: Schematic of the ground-source heat pump with vertical DX heat exchanger. A, B: vertical ground U-tubes; C: compressor; SA: suction accumulator

Table 3: Average thermodynamic parameters of short operating cycles - *heating mode*

Parameter	Unit	Short cycles		
-	-	#1	#2	#3
Compressor input power	kW	1.59	1.60	1.51
Suction pressure	kPa,r	253	255	232
Discharge pressure	kPa,r	1389	1441	1379
Compression ratio	-	5.6	5.6	5.9
Suction temperature	°C	- 5.9	- 6	-6.4
Discharge temperature	°C	59	60	61
Evaporating temperature (dew line)	°C	-17	-17	-18
Condensing temperature (dew line)	°C	35	36.4	35
Pressure drop inside ground U-tubes	kPa	131	124	138
Entering air temperature (fan coil)	°C	19	20	20
Leaving air temperature (fan coil)	°C	35	35	32
Refrigerant flow rate	kg/s	0.0309	0.0305	0.0306
Heat pump average coefficient of performance	-	4.8	4.8	4.6

3.3 Long cycles in heating mode

The heat pump prototype was tested during very long operating cycles in *heating mode* (up to 200 hours or more than 8 days), because such situations may occur in cold climates. It can be seen that, during the first 56 hours of continuous heating demand, the system ran normally with more or less long on/off cycles (see details a, b, c and d in Figures 4, 5, 6 and 7). At the end of this period of time, the compressor power input progressively dropped by up to 31% (Figure 4). Simultaneously, the suction pressure dropped from 234 kPa,r to less than 52 kPa,r, and the discharge pressure diminished by 27% (Figure 5). At the same time, the compressor suction temperature increased from – 6°C to +19°C, while the discharge temperature increased up to 145°C, providing excessive discharge superheating (Figure 6).

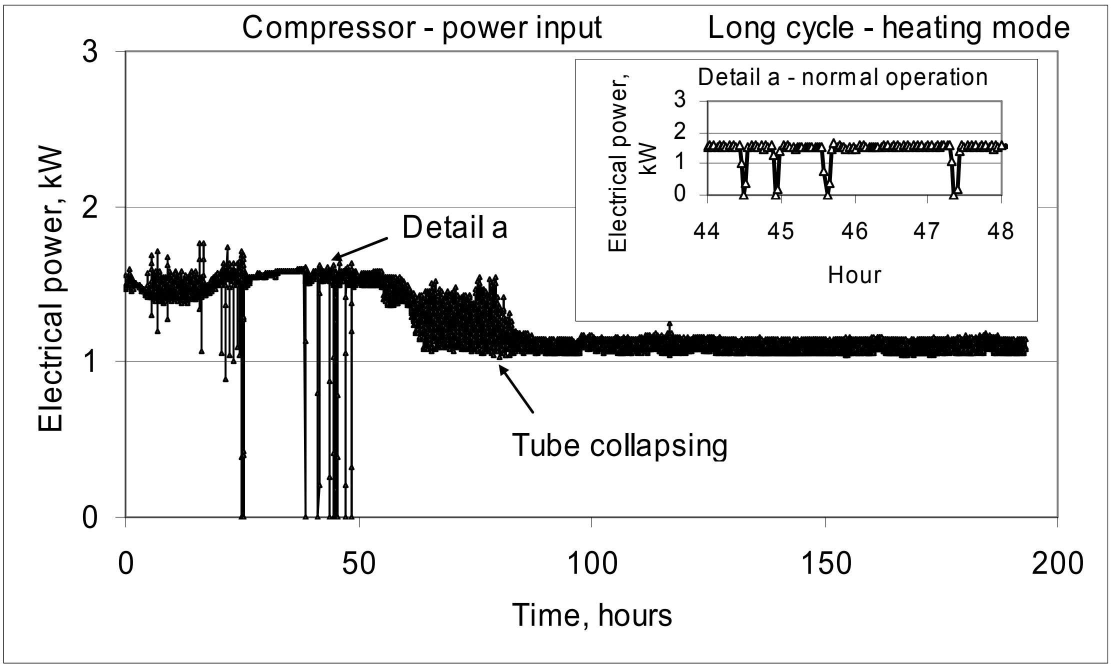

Figure 4: Long cycle in heating mode: compressor operating profile

As a result, the temperature of the indoor air leaving the fan coil dropped from 35°C to less than 22°C indicating a significant loss in the heat pump heating capacity and efficiency. Finally, the pressure drop inside the ground U-tubes (evaporator) increased by nearly 300%, i.e. from about 150 kPa to more than 440 kPa (Figure 7).

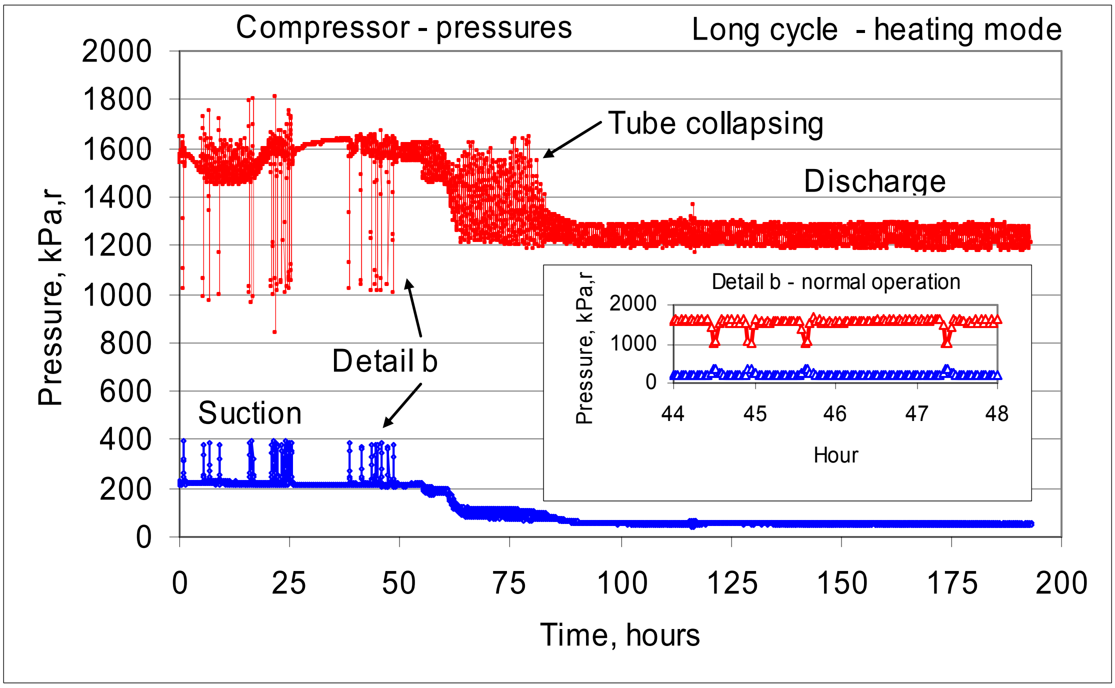

Figure 5: Long cycle in heating mode: suction and discharge pressures

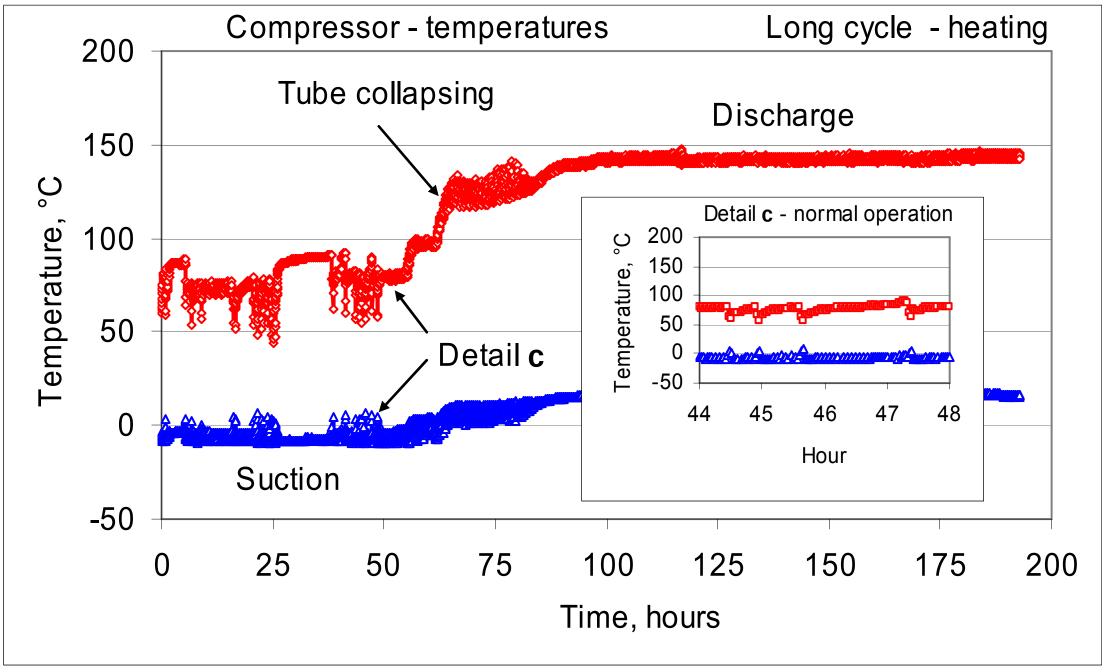

Figure 6: Long cycle in heating mode: suction and discharge temperatures

The relatively sudden change in the heat pump operating behavior after about 56 hours of continuous operation in the heating mode was first explained by the collapsing of the vertical U-tubes due to the ground heat exchanger installation procedure. As previously noted, because of a relatively high groundwater pressure, the upper ends of the steel sleeves were capped to avoid the groundwater from pushing out the grouting material and/or flooding the site. During continuous operating cycles in the heating mode, the evaporating pressures were around 240 kPa_r with DEW line temperatures of -17°C. At such low temperatures, the water and the grout inside the sleeves freeze when the heat pump runs for long periods of time. It was supposed that, because the upper ends of both sleeves had been capped, the relatively high pressure produced by the ice expansion crushed the copper tubes. After collapsing, with evaporating pressures of about 52 kPa,r, the evaporating temperatures dropped by as much as -50°C, and the collapsing phenomenon worsened.

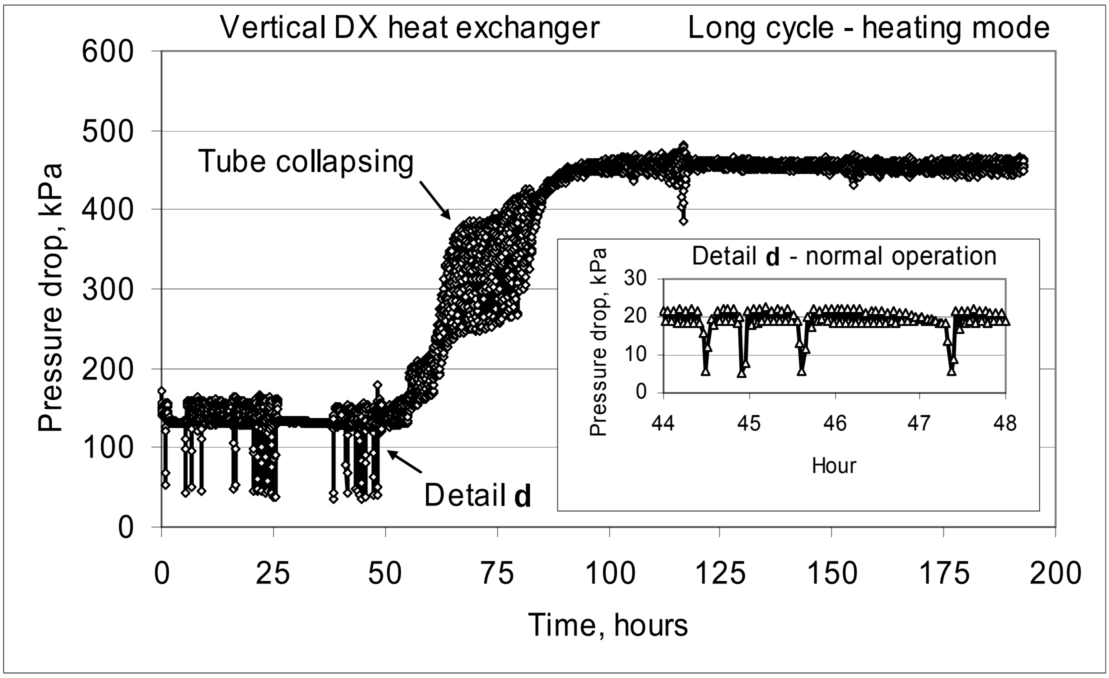

Figure 7 – Long cycle in heating mode: pressure drop inside the ground heat exchanger (evaporator)

Another explanation would be the silica grout weight and the static pressure of the groundwater surrounding the copper U-tubes inside the boreholes. The water static pressure at the bottom of the U-tubes and the equivalent pressure of the silica grout column, combined with the progressive decrease of the refrigerant pressure, probably contributed to the collapsing of the thin-wall, soft copper pipes.

3.3.1 Avoiding tube collapsing

To prevent the ground heat exchanger from collapsing, it is recommended – if permitted - not to grout the boreholes and, also, not to tightly cap the steel sleeves. These measures may prevent ice expansion, which probably was responsible for the refrigerant copper tubes collapsing when very low evaporating pressures occur, and eliminate the risks of copper tube crashing caused by the water and silica grout high static pressure inside the boreholes. Furthermore, these measures help reduce the ground heat exchanger cost and simplify its installation. As additional measure, an (optional) unidirectional adjustable pressure regulating valve (as shown in Figure 3) may be used to keep the refrigerant evaporating temperature above, for example, – 10°C, regardless of the refrigerant type, in order to avoid the risk of excessive soil freezing during long running cycles in heating mode. In this case, the eventual drop in the heat pump heating capacity must be compensated by supplying additional back-up heating.

3.4 Oil return improvement

To improve the oil return to the compressor in heating mode, a simple control strategy may consist in periodical and alternatively pumping-down of the refrigerant liquid from both vertical U-tubes. At certain intervals (for example, every 24 or 48 hours), the solenoid valve 1 is closed and the refrigerant flow through heat exchanger B is stopped. This valve may be kept closed during a short, fixed period of time (seconds), or until the compressor suction pressure reaches a minimum preset pressure. The velocity of the refrigerant and of the oil eventually trapped at the bottom of U-tube B significantly increase, and they are efficiently pumped to the compressor. At the end of these sequences, the solenoid valve 1 opens and the system continues to operate in the heating mode with both U-tubes open. After a period of, for example, five minutes, the solenoid valve 2 is closed during another short period of time (seconds), or until a minimum preset compressor suction pressure is reached. The velocity of the refrigerant and of the oil eventually trapped at the bottom of U-tube A increase and they are rapidly pumped to the compressor. At the end of these operations, the solenoid valve 2 opens and the system continues to normally operate in the heating mode. Another method for improving oil return may be provided by periodically switching the heat pump, for example, every 24 or 48 hours, from the heating to the cooling mode, as suggested in Section 3.5. When using two-speed or two parallel compressors to match the heating/cooling capacity with demand, the refrigerant velocity decreases when the compressor speed slows down, or when only one compressor is running. In these cases, to ensure proper oil recovery from the bottom of the vertical U-tubes at lower heating capacities, the compressor may periodically operate, at certain intervals and for short periods of time, at maximum speed.

3.5 Starting-up and switching modes

When the heat pump is not running, the bulk of the refrigerant within the system naturally migrates to the coldest zones, particularly, to the cold ground heat exchanger areas. In this situation, starting-up the heat pump in the cooling mode or switching the system from the heating to the cooling mode are critical issues because the temperature of the liquid inside the ground U-tubes is very low. The compressor isn't able to build a sufficient pressure differential to push out the liquid from the ground heat exchanger to the expansion valve. In fact, the refrigerant column, which has a density of about 1 320 kg/m^3 at average evaporating temperatures of -17°C, and a height of 37.5 m, provides a static pressure of about 485 kPa$_r$ at the bottom of the vertical U-tubes. If the compressor has to start immediately in cooling mode, its discharge pressure at such low temperatures will be much lower than the liquid static pressure. Also, when the heat pump shuts down in the cooling mode for long periods of time (stand-by mode), the refrigerant is trapped inside the vertical tubes at temperatures of maximum 8°C and builds relatively high static pressures. In both cases, the compressor cannot push the liquid up from the vertical U-tubes because the discharge pressures are lower than the static pressure of the vertical liquid column. To efficiently start-up the system or switch from the heating to the cooling mode, it is proposed to use bi-directional liquid receiver 10.This liquid receiver lets the liquid flow in both directions and is designed to store almost entire refrigerant quantity of the system. To facilitate switching the heat pump *directly* from the heating mode to the cooling mode after having operated for long periods of time in heating mode, or after shut-down periods before cooling demands, the following control strategy is proposed. When the thermostat calls for cooling, the system control first allows the heat pump to start up or to continue running in heating mode during a certain period of time. Solenoid valves 1, 2 and 14 are open and the reversible valve 5 is in his *heating mode* position. The heat pump operates in heating mode, even though the thermostat calls for cooling. After stabilization of the system (5 to 10 minutes), the solenoid valve 14 closes to stop the refrigerant liquid flowing towards ground vertical U-tubes A and B. The refrigerant is thus pumped-down and stored inside the liquid receiver 10 that has a sufficient capacity to store up to 90% of the system's refrigerant charge. If necessary, the rest of the refrigerant charge is stored inside the condenser (fan coil). Both vertical U-tubes are now almost empty of sub-cooled liquid. After a short period of time, or until a minimum preset suction pressure is detected, the compressor shuts down and the control system switches reversible valve 5 in its cooling position. Simultaneously, the solenoid valve 14 opens and the compressor normally starts-up in cooling mode.

4. CONCLUSIONS

The proposed simplified design approach shows that the length of the vertical boreholes and the associated costs of direct expansion residential ground heat exchangers may be reduced by at least 40% compared to those of indirect secondary loop systems. The maximum depth of the vertical boreholes of such ground DX heat exchangers should not exceed 40 m, while the actual unitary heat flux (per-unit U-tube length) between the refrigerant and the surrounding soil is of about 42 W/m in heating mode and of 56 W/m in cooling mode. These performances exceed by 40 to 45% those of indirect secondary fluid loop GSHP systems. The heat pump prototype operated normally during 56 hours of continuous heating demand. However, when this period of time was exceeded, the ground refrigerant soft copper U-tubes collapsed, probably due to the soil freezing around the capped sleeves and/or to the grout weight and groundwater static pressure at the lower sides of the vertical U-tubes inside the boreholes. To prevent such undesired incidents it is recommended – when possible - not to install vertical steel sleeves and/or not to tightly cap them at the upper ends. Finally, new proposed refrigerant circuits, including a bi-directional liquid receiver and simple control sequences provide additional means to improve oil return and to make it easier starting-up and switching the heat pump from the heating and/or stand-by modes to the cooling mode.

5. NOMENCLATURE

c	Specific heat (kJ/kgK)
d	Diameter (m)
η	Dynamic viscosity (Pa.s)
g	Gravitational acceleration (m/s^2)
G	Mass velocity (kg/m^{2}s)
h	Heat transfer coefficient (W/m^{2}K)
k	Thermal conductivity (W/mK)
ν	Cinematic viscosity (m^2/s)
ω	Velocity (m/s)
$\dot{Q}$	Heat transfer rate (W)
$\dot{q}$	Heat flux density (W/m^2)
r	Radius (m)
ρ	Mass density (kg/m^3)
T	Temperature (°C)

Subscripts

cc	Convective condensation
cd	Condensing
cool	Cooling mode
cv	Convective vaporization
eq	Equivalent
extr	Extraction
heat	Heating mode
horiz	Horizontal
i	Interior
l	Liquid
∞	Infinity
rej	Rejection
R	Refrigerant
s	Soil
sat	Saturation
tot	Total
v	Vapor
vert	Vertical
w	Wall

6. REFERENCES

Kaye, G. A. 1993. “Staged cooling direct expansion geothermal heat pump”, US patent

Ciconkov, R. 2001. “Refrigeration solved examples”, Faculty of mechanical Engineering, Skopje, Macedonia.

Incropera, F.P., DeWitt, D.P. 2002. “Fundamentals of Heat and Mass Transfer”, 5th edition, John Wiley & Sons.

Minea, V. 2005. “Development of a vertical direct expansion ground-coupled heat pump”, *Research report*, Hydro-Québec Research Institute, Canada.