Performance and Optimum for a Ground-Coupled Liquid Loop Heat Recovery Ventilation System

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PERFORMANCE AND OPTIMUM FOR A GROUND-COUPLED LIQUID LOOP HEAT RECOVERY VENTILATION SYSTEM

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ABSTRACT

Ground-source heat pump systems can be designed to include liquid-loop Heat Recovery Ventilation (HRV), free cooling and recharging of the borehole collector. The HRV-system uses an exhaust-air coil to warm brine that has been preheated by a borehole heat exchanger. The warm brine then heats a supply-air coil and returns to the borehole. The performance of the exhaust-air and supply-air coils has an influence on the HRV system efficiency. This paper presents a mathematical model to investigate how the brine flow rate and the allocation ratio between the exhaust and supply-air coils affect the heat recovery efficiency.

1. INTRODUCTION

Ground-coupled heat pumps are commonly used in Swedish residential heating systems. The most popular heat source is a vertical borehole and there are now more than 230,000 systems in operation. In the boreholes, temperature will slowly drop with operating time and hence *COP* of the heat pump as well as the heat extraction will drop. To mitigate this situation, a deeper borehole is usually the most cost-effective solution for a new installation. However, recharging the borehole may be a better method to prevent a long-term degradation of performance and even improve on the initial results. Recharging can be used for both new and existing installations and will compensate for the extracted energy and resulting long-term temperature drop. Most Swedish houses have mechanical ventilation and exhaust-air is a feasible heat source available all the year. The effect of recharging the borehole to improve the performance of a ground-coupled heat pump system has been examined by experimental [Fahlen, 2002] and theoretical research [Claesson et al., 1985].

A ground-coupled heat recovery ventilation system includes three heat exchanges (see figure 1): two coils are connected with the borehole heat exchanger. The loop brine was warmed by the exhaust coil, and then warm the supply coil and finally being preheated by the borehole heat exchanger. Hence it is complicated to determine the operating conditions for maximum heat recovery efficiency. This paper set up a mathematical model to better understand the characteristics of a ground-coupled HRV system. The model can also assist in calculating the overall heat recovery efficiency and in finding the optimal brine flow rate. Finally, a model will be helpful in the search for the optimal heat transfer allocation ratio between the exhaust and supply coils. An optimal combination implies the maximum heat recovery efficiency.

2. MATHEMATICAL MODEL

In a ground-coupled liquid-loop HRV system the brine extracts heat from the exhaust coil to heat the supply coil. After being cooled by the supply-air, the brine is preheated by the borehole heat exchanger. Using the effectiveness-*Ntu* method, and considering the energy balance, a set of equations can be written to describe heat transfer in the ground-coupled HRV system.

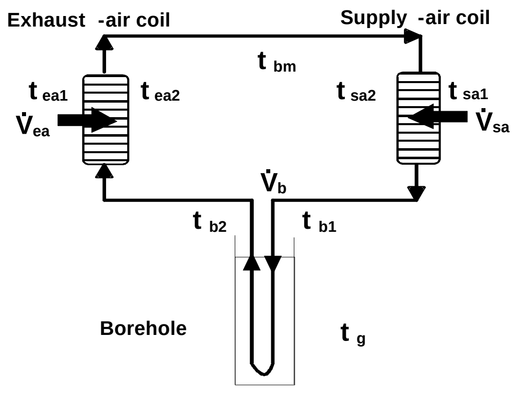

Figure 1: A ground-coupled liquid-loop HRV system

The exhaust-air coil

$$\dot{C}_{ea} = \dot{V}_{ea} \cdot \rho_a \cdot c_{p,a} \quad \text{[W/K]} \quad (1)$$

$$\dot{Q}_{ea} = \dot{C}_{ea} \cdot (t_{ea1} - t_{ea2}) \quad \text{[W]} \quad (2)$$

$$\dot{Q}_{ea} = \dot{C}_{b} \cdot (t_{bm} - t_{b2}) \quad \text{[W]} \quad (3)$$

$$\dot{Q}_{ea} = \varepsilon_{ea} \cdot \dot{Q}_{ea,\max} \quad \text{[W]} \quad (4)$$

$$\dot{Q}_{ea,\max} = \dot{C}_{ea,\min} \cdot (t_{ea1} - t_{b2}) \quad \text{[W]} \quad (5)$$

$$Ntu_{ea} = \frac{(U \cdot A)_{ea}}{\dot{C}_{ea,\min}} \quad \text{[-]} \quad (6)$$

$$\varepsilon_{ea} = f(Ntu_{ea}, \frac{\dot{C}_{ea,\min}}{\dot{C}_{ea,\max}}) \quad \text{[-]} \quad (7)$$

The supply-air coil

$$\dot{C}_{sa} = \dot{V}_{sa} \cdot \rho_a \cdot c_{p,a} \quad \text{[W/K]} \quad (8)$$

$$\dot{Q}_{sa} = \dot{C}_{sa} \cdot (t_{sa2} - t_{sa1}) \quad \text{[W]} \quad (9)$$

$$\dot{Q}_{sa} = \dot{C}_{b} \cdot (t_{bm} - t_{b1}) \quad \text{[W]} \quad (10)$$

$$\dot{Q}_{sa} = \varepsilon_{sa} \cdot \dot{Q}_{sa,\max} \quad \text{[W]} \quad (11)$$

$$\dot{Q}_{sa,\max} = \dot{C}_{sa,\min} \cdot (t_{bm} - t_{sa1}) \quad \text{[W]} \quad (12)$$

$$Ntu_{sa} = \frac{(U \cdot A)_{sa}}{\dot{C}_{sa,\min}} \quad \text{[-]} \quad (13)$$

$$\varepsilon_{sa} = f(Ntu_{sa}, \frac{\dot{C}_{sa,\min}}{\dot{C}_{sa,\max}}) \quad \text{[-]} \quad (14)$$

The borehole heat exchanger

$$\dot{C}_{b} = \dot{V}_{b} \cdot \rho_b \cdot c_{p,b} \quad \text{[W/K]} \quad (15)$$

$$\dot{Q}_g = \dot{C}_b \cdot (t_{b2} - t_{b1}) \qquad \text{[W]} \qquad (16)$$

$$\dot{Q}_g = \varepsilon_g \cdot \dot{C}_b \cdot (t_{b1} - t_g) \qquad \text{[W]} \qquad (17)$$

$$Ntu_g = \frac{(U \cdot A)_g}{\dot{C}_b} \qquad \text{[-]} \qquad (18)$$

$$\varepsilon_g = 1 - \exp(-Ntu_g) \qquad \text{[-]} \qquad (19)$$

$$\varepsilon_g = \frac{t_{b2} - t_{b1}}{t_g - t_{b1}} \qquad \text{[-]} \qquad (20)$$

Equation (21) defines the heat recovery efficiency of a HRV system, which is applicable also to a ground-coupled system:

$$\eta = \frac{t_{sa2} - t_{sa1}}{t_{ea1} - t_{sa1}} \qquad \text{[-]} \qquad (21)$$

Obviously, the types and sizes of the three heat exchangers, their heat transfer capabilities $(U \cdot A)_{ea}$, $(U \cdot A)_{sa}$ and $(U \cdot A)_g$, the air and brine flow rates $\dot{V}_{ea}$, $\dot{V}_{sa}$, and $\dot{V}_b$, as well as the operational temperatures t_g, t_{ea1} and t_{sa1}, will affect the system heat recovery efficiency.

In order to achieve the maximum heat recovery efficiency, there are two optimising tasks. The first is to decide on the best allocation of the individual heat transfer capacities $(U \cdot A)_{ea}$, $(U \cdot A)_{sa}$ and $(U \cdot A)_g$ within a fixed total capacity (i.e. a fixed first cost). The second is to determine the optimal brine flow rate $\dot{V}_b$ in the loop. To simplify the investigation of the efficiency of a HRV system with three heat exchangers, we define the following three dimensionless parameters:

$$m = \frac{(U \cdot A)_{ea}}{U \cdot A} \qquad n = \frac{(U \cdot A)_g}{U \cdot A} \qquad C_r = \frac{\dot{C}_b}{\dot{C}_{ea}} \qquad \text{[-]} \qquad (22)$$

where $(U \cdot A)$ is the total overall heat transfer coefficient of the exhaust coil and the supply coil, i.e. $U \cdot A = (U \cdot A)_{ea} + (U \cdot A)_{sa}$.

Hence, the heat recovery efficiency η of the ground-coupled HRV system is related to the parameters m, n, C_r, t_{ea1}, t_{sa1}, and t_g. Unfortunately, it is difficult to express the efficiency η explicitly as a function of these parameters. Therefore, a numerical analysis was made using the programming platform EES [EES, 2005].

3. NUMERICAL ANALYSIS

For a traditional liquid-coupled HRV system with two-coil, balanced heat capacity flow rates $\dot{C}_{ea} = \dot{C}_{sa} = \dot{C}_b$ provide the maximum thermal efficiency [Igor et al., 2003] with equal heat transfer capabilities of the exhaust and supply coils. For this ground-coupled HRV system with three heat exchangers, what relationship between three heat exchangers can we get the maximum thermal efficiency? For investigating the characteristics, we chose an existing installation [Fahlen, 2002] with an exhaust-air coil for recharging of a borehole collector as a case study. In this, we investigated the effects of varying the parameters that affect the thermal efficiency of the described three-heat exchanger HRV system.

3.1 Conditions of the case study

A mathematical model based on the previous discussion was used to analyse the heat recovery efficiency of a ground-coupled HRV system in a single-family house [Fahlen, 2002]. The house has an exhaust-air-to-brine coil which is a single-pass, cross-counter-flow heat exchanger with both fluids unmixed. And the house is also equipped with an identical supply-air coil. The brine-side of the coils is connected to a ground heat exchanger in a vertical borehole with L = 60 m, D_{bh} = 115 mm, and D_o = 33.4 mm (a U-tube made of polyethylene). The brine is a 30 % ethanol-water mixture. In the calculations the following data, largely based on measurements, are used: thermal conductivity of the bedrock λ_g = 3.5 $W\cdot m^{-1}\cdot K^{-1}$, undisturbed ground temperature t_g = 6.5 °C, exhaust-air temperature t_{ea1} =20 °C, and air flow rate $\dot{V}_{ea} = \dot{V}_{sa}$ = 165 m^3/h.

3.2. Optimising brine flow rate

Using the data of 3.1 as input to the model, as well as assuming outdoor air temperature t_{sa1} = -10 °C, the heat recovery efficiency of the ground-coupled HRV system was calculated. Ground borehole heat exchange transfer capability $(U{\cdot}A)_g$ is time dependent, so the efficiency for different brine flow rates was calculated after one hour of operation. Figure 2a shows that there is an optimal brine flow rate giving a maximum heat recovery efficiency of the system. The heat recovery efficiency η decreases rapidly when the brine flow rate goes below the optimum, but it is much less sensitive for flow rates exceeding the optimal value. The optimal flow rate is designated $C_{r,opt}$. In the current example, $C_{r,opt}$ = 1.5 (see figure 2a). From calculation result, also getting $m = (U{\cdot}A)_{ea}/(U{\cdot}A)$ = 0.5, and $n = (U{\cdot}A)_g/(U{\cdot}A)$ = 0.68.

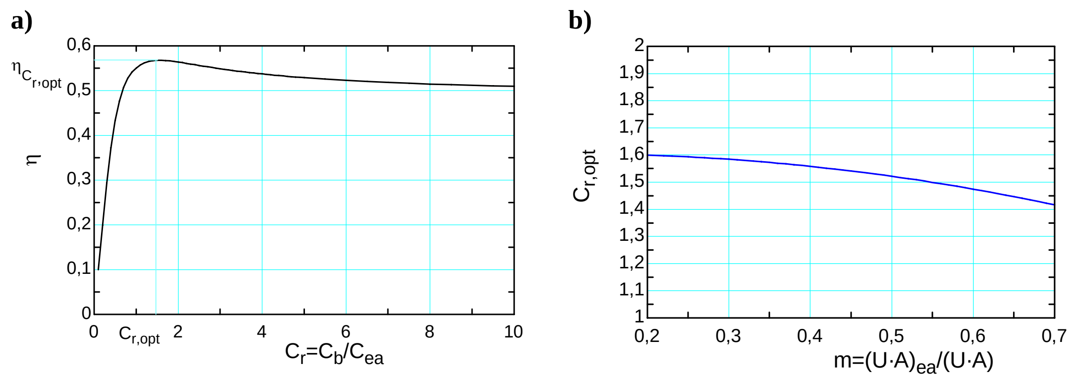

Figure 2: The optimal brine flow rate (a) for m = 0.5 and n = 0.68, and the effect (b) of the allocation ratio m of $(U{\cdot}A)$ on optimal brine flow rate for n = 0.68, t_{sa1} = -10 °C.

Maintaining the overall $U{\cdot}A$ and the other input data of the case study constant, the allocation ratio between the exhaust coil and the supply coil can be changed. This implies changing the dimensionless parameter m to study the effects on system performance. Figure 2b illustrates the resulting relation between $C_{r,opt}$ and the dimensionless parameter m. The diagram shows that the optimal flow rate $C_{r,opt}$ depends to some extent on the allocation ratio m. The optimal brine flow rate decreases with increasing exhaust-air coil heat transfer surface. The effect on $C_{r,opt}$, however, is weak.

3.3. Optimising allocation ratio between the exhaust coil and the supply coil

The relative size of $(U{\cdot}A)_{ea}$ and $(U{\cdot}A)_{sa}$ will affect the performance of the HRV system. The question is how to allocate the heat transfer capacity to each of the two coils to achieve maximum efficiency. Assuming that the total overall heat transfer coefficient $(U{\cdot}A)$ is fixed, figure 3a shows that there is a maximum heat recovery efficiency of the system when the allocation ratio m is changed. For each allocation ratio m, the system operates at the optimal brine flow rate. Figure 3a indicates that the optimal allocation ratio is m_{opt} = 0.34. This means that $(U{\cdot}A)_{ea}$ is roughly one third of the total overall heat transfer capacity $(U{\cdot}A)$, i.e. when $(U{\cdot}A)_{ea} = 1/3{\cdot}(U{\cdot}A)$ or $(U{\cdot}A)_{ea} = 1/2{\cdot}(U{\cdot}A)_{sa}$, the system provides a maximum heat recovery efficiency.

Figure 3b shows the effect of the ratio n of $(U{\cdot}A)_g$ and $(U{\cdot}A)$ on m_{opt}. In the range n = 0.1 to n = 1, m_{opt} only changes from 0.36 to 0.3. The average value of m_{opt} is 0.33 and thus m_{opt} = 1/3. Therefore, in a ground-coupled HRV system, the exhaust-air coil size should be half of that supply-air coil size which provides the maximum heat recovery efficiency. A reason for allocating more capacity to the supply coil is that this coil handles the total capacity of the ground and exhaust-air coils.

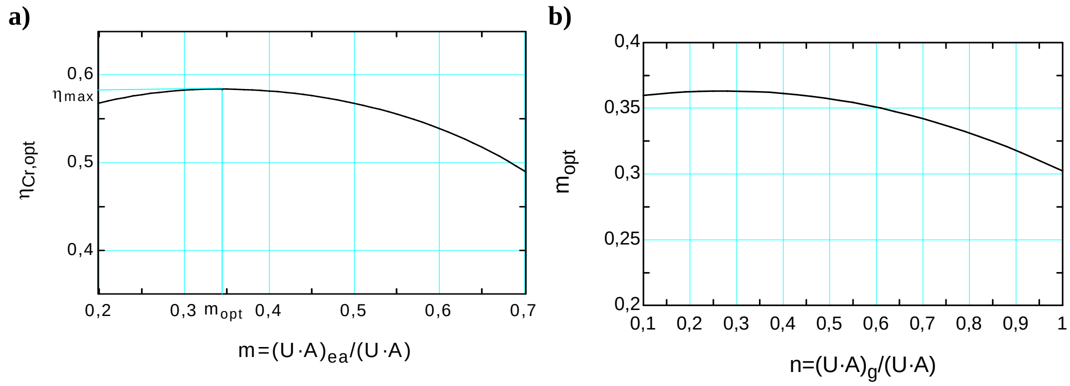

Figure 3: The effect (a) of the allocation ratio m of $U{\cdot}A$ on heat recovery efficiency for n =0.68 and the effect (b) of the ratio n of $(U{\cdot}A)_g$ and $U{\cdot}A$ on the optimal allocation ratio m_{opt}. Outdoor temperature t_{sa1} = -10 °C.

3.4. Effect of outdoor temperature on the HRV system performance

The outdoor-air temperature determines the thermal load of the exhaust and supply coils and hence affects the heat recovery efficiency of the ground-coupled HRV system. Figure 4a shows that the efficiency increases when the outdoor temperature decreases. The diagram provides information at four different ratios n of $(U{\cdot}A)_g$ and $U{\cdot}A$. The ground coil transfer capacity $(U{\cdot}A)_g$ was fixed at the value of the case-study and the total capacity of the air coils was changed. The ratio n = 1 means $U{\cdot}A = (U{\cdot}A)_g$, and n = 0.1 means $U{\cdot}A = 10{\cdot}(U{\cdot}A)_g$. As shown by figure 4a, the larger $U{\cdot}A$ is and the lower the outdoor temperature is, the more energy is transferred to the supply air in a ground-coupled HRV system. At low outdoor temperatures, there will be an increased benefit form the ground-coil as the brine temperature after the supply coil drops to a level that is lower than that of the ground.

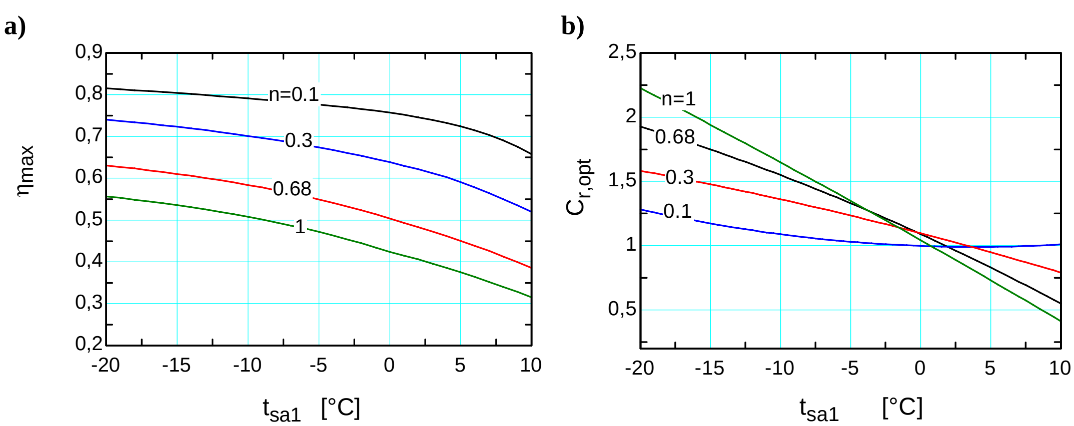

Figure 4: The effect of outdoor temperature on heat recovery efficiency (a) for m = 1/3 and on optimal brine flow rate (b) for m = 1/3.

Figure 4b shows that the optimal brine flow rate will increase with a decreasing outdoor temperature. The diagram also illustrates that for different ratios n, the outdoor temperature influence on the optimal brine flow rate will be different. The larger $U{\cdot}A$ is, the less is the optimal brine flow rate affected by the outdoor temperature. For example, figure 4b indicates that for n = 0.1, i.e. $U{\cdot}A = 10{\cdot}(U{\cdot}A)_g$, the optimal brine flow rate may be kept approximately constant during the whole heating season. Hence a constant speed brine pump can be used and still provide a maximum efficiency.

3.5. Effect of the heat transfer capacity of the air and ground coils on heat recovery efficiency

Improving each heat exchanger performance must improve the total HRV system performance. The question is which one is more efficient to improve the system performance. Figure 6a shows, when keeping the same $(U{\cdot}A)_g$ as in the case-study the heat recovery efficiency is improved by changing the $U{\cdot}A$. With an increased $U{\cdot}A$, see curve 1 in figure 5a, efficiency increases, and the maximum efficiency is very sharp. Deviating from the optimal flow rate, the efficiency decreases rapidly. On the other hand, with a decrease of $U{\cdot}A$, see curve 3, the efficiency decreases and the efficiency curve is flat which implies that the maximum heat recovery efficiency is not very sharp.

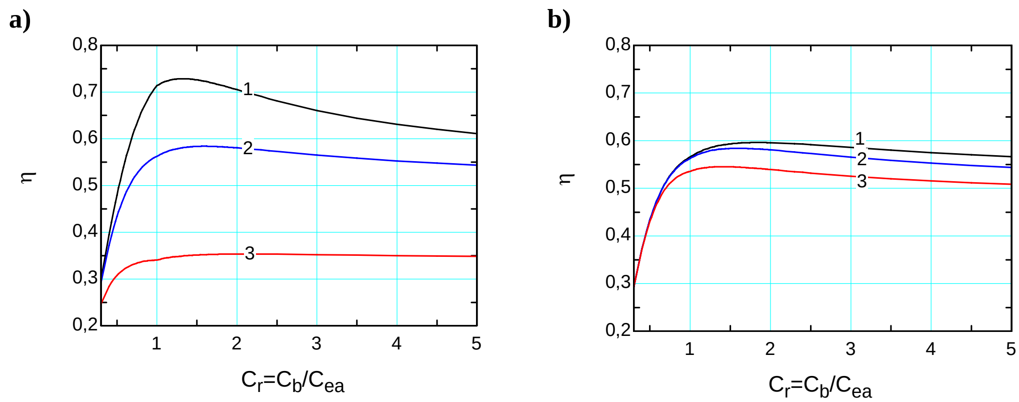

Figure 5: The effect of $U{\cdot}A$ (a) with $(U{\cdot}A)_g = (U{\cdot}A)_{g,case\text{-}study}$ and $(U{\cdot}A)_g$ (b) with $U{\cdot}A = (U{\cdot}A)_{case\text{-}study}$ on the heat recovery efficiency and optimal flow rate for m = 1/3 and t_{sa1} = -10 °C. Designations in the diagrams:
a) 1: $U{\cdot}A = 3{\cdot}(U{\cdot}A)_{case\text{-}study}$, 2: data from case-study, 3: $U{\cdot}A = 1/3{\cdot}(U{\cdot}A)_{case\text{-}study}$;
b) 1: $(U{\cdot}A)_g = 3{\cdot}(U{\cdot}A)_{g,case\text{-}study}$, 2: data from case-study, 3: $(U{\cdot}A)_g = 1/3{\cdot}(U{\cdot}A)_{g,case\text{-}study}$

Figure 5b shows that when keeping the same $U{\cdot}A$ as in the case-study, changing $(U{\cdot}A)_g$ improves the heat recovery efficiency. However, compared with the effect of $U{\cdot}A$ in figure 6a, the effect of $(U{\cdot}A)_g$ is weak. Curve 1 in figure 5b indicates that even increasing $(U{\cdot}A)_g$ to three times the value of the case-study, the efficiency only improves a little. This indicates that the effect of $U{\cdot}A$ on the heat recovery efficiency is larger than the effect of $(U{\cdot}A)_g$ with the prerequisites of the current study. If the ground coil is excluded, the maximum efficiency goes down from 58.4 % to 48.6 % with equal heat transfer capabilities of the exhaust and supply coils $(U{\cdot}A)_{ea} = (U{\cdot}A)_{sa}$ and balanced heat capacity flow rates C_r = 1. Compared to figure 3a, which shows that the maximum efficiency is 58.4 %, the ground-coupled HRV system could make the heat recovery efficiency increase 20 % when keeping the same $U{\cdot}A$ as in the case study. This is without the extra benefit of not needing a defrost sequence.

3.6. Brine inlet temperature to the exhaust coil

To maintain a high heat transfer rate, condensed water vapour must not freeze on the exhaust air coil. This can be avoided if the brine temperature entering the exhaust coil is higher than zero (0 °C). In a ground-coupled HRV system, the brine temperature entering the exhaust coil is pre-heated by the borehole heat exchanger. Figure 6 shows that the brine temperature decreases with the outdoor temperature. However, it never goes below zero (0 °C) in the range of outdoor temperatures of figure 6 and hence there will be no frosting and defrosting will not be needed for the ground-coupled HRV system. Experience from conventional recuperative HRV systems has shown that in the best of circumstances, the efficiency drops by at least 20 % when defrosting becomes necessary. The ground-coupled system becomes correspondingly more favourable during that part of the year when heat recovery and preheating of supply air is needed the most.

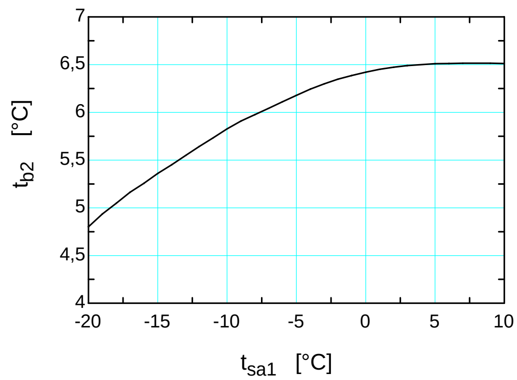

Figure 6: The brine inlet temperature at the exhaust coil for n = 0.68 and m = 1/3.

4. CONCLUSIONS

A ground-coupled liquid loop HRV system, with three heat exchangers, has an optimal allocation ratio between the exhaust coil and the supply coil and the optimal brine flow rate that provides the maximum heat recovery efficiency. The optimal allocation ratio will depend on the specific conditions of the building. In the current case-study, it was determined that the heat transfer capability of the exhaust air coil should be half that of the supply air coil, i.e. $(U{\cdot}A)_{ea}=1/2{\cdot}(U{\cdot}A)_{sa}$. The optimal brine flow rate varies with the outdoor temperature and the ratio of $(U{\cdot}A)_g$ and $U{\cdot}A$. A larger optimal brine flow rate will be needed for a lower outdoor temperature. The larger $U{\cdot}A$ is, the less the outdoor temperature affects the optimal brine flow rate. When $U{\cdot}A$ is larger than ten times $(U{\cdot}A)_g$, it is possible to operate with a constant optimal brine flow rate for the entire heating season.

The performance of the exhaust-air and supply-air coils has a strong influence on the possibility to improve the efficiency of a ground-coupled HRV system. The larger the value of $U{\cdot}A$ is and the lower the outdoor temperature is, the more energy can be transferred to the supply air. Furthermore, because of the borehole heat exchanger, the brine temperature entering the exhaust coil never goes below zero (0 °C) in optimal operation. Hence, contrary to the case of conventional liquid-loop HRV systems, defrosting will not be needed for the ground-coupled HRV system. This can make the ground-coupled system over 20 % more efficient than a conventional recuperative system at low outdoor temperatures. Also, the efficiency increases when the outdoor temperature decreases. This is a desirable characteristic in most buildings since normally the efficiency has to be turned down anyway at rising outdoor temperatures due excess heat from internal loads.

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NOMENCLATURE

$\dot{C}$ heat capacity flow rate $(\dot{C} = \dot{V} \cdot \rho \cdot c_p)$ [W·K^{-1}]
C_r capacity rate ratio ($C_r = \dot{C}_b / \dot{C}_{ea}$) [-]
c specific heat capacity [J· kg^{-1}·K^{-1}]
m overall heat transfer coefficient ratio ($m = (U{\cdot}A)_{ea}/(U{\cdot}A)$)
n overall heat transfer coefficient ratio ($n = (U{\cdot}A)_g/(U{\cdot}A)$)
Ntu the number of heat transfer units [-]
$\dot{Q}$ heat transfer rate [W]

t Celsius temperature [°C]
$U{\cdot}A$ heat exchanger transfer capability [$W{\cdot}K^{-1}$]
$\dot{V}$ volumetric flow rate [$m^3{\cdot}s^{-1}$]

Greek symbols
ε heat exchanger effectiveness [-]
η ground-coupled heat recovery efficiency [-]
λ thermal conductivity, [$W{\cdot}m^{-1}{\cdot}K^{-1}$]
ρ density [$kg{\cdot}m^3$]

Subscripts
b brine
bh borehole
bm middle parameter between exhaust coil and supply coil
ea exhaust-air or exhaust-air coil
g ground
sa supply-air or supply-air coil

Abbreviations
EES Engineering Equation Solver
HRV Heat Recovery Ventilator

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