# SIMPLE DESIGN TOOLS FOR EARTH-AIR HEAT EXCHANGERS

Ralph T. Muehleisen
Decision and Information Sciences
Argonne National Lab, Argonne, IL, USA
rmuehleisen@anl.gov

## ABSTRACT

In the early stages of the design of building systems, the use of simple design tools can help estimate the size and/or impact of system components in evaluating the viability of various technologies. However, such design tools are not readily available to evaluate earth-air heat exchangers (EAHEs), also known as earth-tubes. Furthermore, even though many researchers have developed sophisticated equations to analyze EAHEs, they cannot be easily recast into design equations and must be used by trial-and-error. This paper describes a set of simplified analysis and design equations to support early-stage EAHE design and which are suitable for implementation in a spreadsheet. The equations we have developed allow the designer to quickly determine the length of tubing required for a desired level of heat transfer effectiveness; estimate the pressure drop across the system and required fan power; and estimate the mean monthly temperature of air exiting the tube.

## INTRODUCTION

As the need for energy-efficient building designs increases, the use of passive heating/cooling and renewable resources also increases. One way to reduce the use of energy in the heating and cooling of ventilation air is to preheat the air in the winter and precool the air in the summer using an earth-air heat exchanger (EAHE) also known as an earth-tube. In an EAHE, ventilation air is drawn into the building through a system of tubes located in the soil near or beneath the building. EAHEs are not a new technology; indeed, the concept dates back at least to the 1st century BC in the Middle East (Oleson, 2008), where air was cooled in the qanats that were used to transport water and also used in Roman architecture.

The EAHE concept is quite simple: a tube is buried in the soil as shown in Figure 1. The soil will be at a temperature warmer than the outside air in winter and cooler than the outside air in summer. Ventilation air is drawn into the building through the buried tube, heating it in the winter and cooling it in the summer—and passively reducing the overall cooling and heating load.

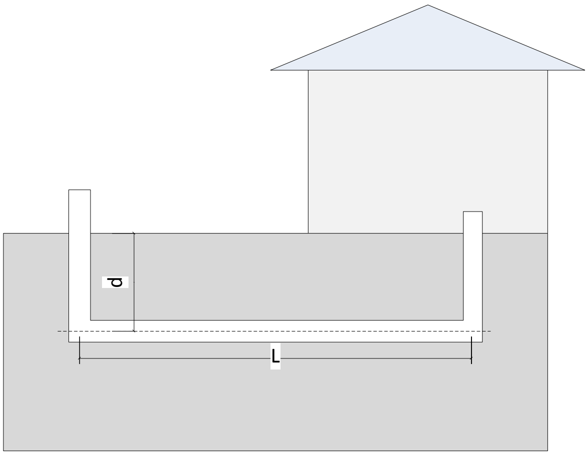


*Figure 1 Earth-air heat exchanger (EAHE) of length L buried at depth d in the soil*

In the past two decades, much research has been conducted to develop analytic and numerical methods for analyzing EAHEs (Hollmuller 2003; Florides and Kalogirou 2007; Tittelein et al. 2009; Lee and Strand 2008; Ghosal et al. 2005; Cucumo et al. 2008). There are now earth-tube analysis modules available for TRNSYS (Hollmuller and Lachal 1998) and EnergyPlus (Lee and Strand 2006). To obtain a complete analysis of the performance of earth-tubes in buildings, the use of such software is recommended; however, to determine the suitability of the technology and for initial design phases, development of the required TRNSYS or EnergyPlus models is both time consuming and unnecessary. Rather, using a set of simplified design equations that help inform the technology potential and determine basic component sizing is more appropriate.

De Paepe and Janssens (2003) and, more recently, Badescu and Isvoranu (2011) have developed EAHE design equations and procedures. De Paepe and Janssens developed equations for estimating the number of heat transfer units per unit length (NTU/L) from which the required length of tube, as well as normalized pressure drops across the tube can be estimated, given the inside temperature of the earth-tube wall. However, De Paepe and Janssens did not develop any equations for estimating the ground temperature and assumed that the inside of the tube wall would be at the same temperature as the surrounding soil. Badescu and Isvoranu's more recent method is more advanced in terms of the basic airflow analysis because it deals with pressure balances in parallel branches of tubes; however, like De Paepe and Janssen, it is quite simplified in terms of the heat transfer analysis.

In this paper, we extend the methods of De Paepe and Janssens (2003) and Badescu and Isvoranu (2011) to include calculation of the soil temperature around the tube, as well as the effects of heat transfer through the tube wall, and to use more more recently developed correlations for friction factors and Nusselt numbers for increased accuracy in computation of heat transfer. In particular, we selected correlations which include the effects of surface roughness, because the high surface roughness of concrete pipes will result in increased heat transfer compared to smooth PVC or steel pipes. The basic performance measures of De Paepe and Janssens are then used to develop design equations for the required tube length and normalized pressure drop to achieve the desired level of heat transfer performance.

## ANALYSIS AND DESIGN EQUATIONS

### Development

The development of design equations usually stems from simplified analysis equations. The typical procedure involves determining which variables of the system can be considered to be "known" (i.e., inputs to the system) and which variables are the desired design outputs. Once the design outputs are selected, the analysis equations are manipulated to solve for the design variables in terms of the input variables. Although sometimes a closed-form solution is not available, often the analysis equations can be cast into a form where the design variable solutions are the roots to some characteristic equation. In either case, the design problem is amenable to spreadsheet implementation either through direct calculation of the design outputs or through the use of root finding or minimization routines built into the spreadsheet.

In the case of the EAHE system, we have decided that we will consider the location as known, which means the outside air temperatures throughout the year and the basic soil conditions are known. We assume the designer has selected an air volume flow rate through the EAHE. This choice would usually be dictated by the ventilation requirements of the building. We further assume that the designer has selected a basic tube size for use in the EAHE; this selection sets the inner and outer radius of the tube, as well as its thermal properties. Although tube diameter could have been allowed to be a design variable, in most cases, designers will be selecting from a fairly fixed set of tube sizes, so the tube size is considered a fixed input. In addition, the number of tubes used in the system is also a fixed input. Selecting the best combination of tube diameter and number of tubes may involve a little trial and error on the part of the designer; however, because the number of combinations is limited, we think the choice of inputs is still well suited to preliminary design.

For determining the suitability of an EAHE, a user usually wants to know the length of tube required to provide a desired amount of possible preheating or precooling of the ventilation air. For that scenario, the designer simply wants to know how close the air can get to the tube wall temperature as it travels through. Our tool lets the designer input the desired heat-transfer effectiveness of the EAHE, and the required tube length is computed. At the same time, the normalized pressure drop of the main EAHE tube (pressure drop per unit length not including any bends) is computed. This initial computation does not require knowledge of any local weather conditions, except for an air temperature required to compute the thermodynamic properties of air.

Once the length of the tube has been computed (and is known), the designer can add more complete weather information such that the spreadsheet can predict the following: the monthly average soil temperature, the monthly average temperature of the air leaving the tube, and the average expected passive heating and cooling energy provided to the ventilation air by the EAHE. Further, the tool allows users to add the effects of bends in the tubes to estimate the expected total pressure drop in the system so that users can estimate the required fan size.

### Assumptions

In order to simplify the equations and make the problem more amenable to simple "spreadsheet implementation," several assumptions were made in equation development.

First, we assume that the temperature of the outside of the tube (i.e., the tube wall in contact with the soil) is constant along its length. For a fairly short tube (10–50 m), this assumption is probably fair, but for a longer tube, it will no longer be true. This assumption can be overcome by including an effective thermal resistance between the soil and the outside of the duct.

Second, we assume that the soil ground temperature is uniform around the tube. A sensitivity analysis showed that this assumption holds for tubes where the diameter was up to 15% of the depth for typical depths, although the deviation is both depth and seasonally dependent. When this condition is not the case, the tool will overpredict EAHE performance, and so the designer should probably increase the length of the tube to compensate and should definitely perform an analysis with a more accurate program.

Third, we only consider the yearly ground temperature oscillation. The short-timescale, daily variations are ignored. This, too, is a good assumption since most tubes that are buried more than 1 m deep where the variations become negligible.

Fourth, we assume air turbulent flow throughout the EAHE. With small-diameter pipes, the entrance length is small enough that this approximation is fair. If the tube length is very short, then a full computational fluid dynamics (CFD) analysis may be required so that the designer can accurately compute the heat transfer from soil to air in the tube.

## Air Flow

The EAHE designer must know the desired volume flow rate of air through the tube in order to begin choosing the size and number of ducts to be used. The number and size of ducts are not unique for a given EAHE performance, and so designers must weigh the tradeoffs of heat transfer performance and pumping power required to generate the flow. In our design equations, the designer must input a required total volume flow rate $V$, an inner tube radius $r_i$, and the number of parallel tubes $N_t$. The mean velocity of the air in one of the heat exchange tubes, $v_a$, will then be:

$$v_a = \frac{V}{N_t \pi r_i^2} \tag{1}$$

and the mass flow rate in the one tube, $m_a$, will be:

$$m_a = \rho_a v_a \pi r_i^2 \tag{2}$$

where $\rho_a$ is the density of air in the tube.

## Soil Temperature

For our simplified model, we chose to implement the soil temperature estimation method found in Annex A of BS EN 15241 (British Standards Institution, 2007). Cucumo et al. (2008) developed an advanced analytic equation that provides a more accurate estimate of the ground soil temperature but also requires the user to input the daily average temperature and daily temperature fluctuations. Although such information can be fairly easily estimated from typical meteorlogical year (TMY) data, such accuracy is rarely needed in the initial design stage of an EAHE system and makes use of the spreadsheet implementation more cumbersome.

In a simplified method, the ground temperature can be assumed to oscillate about a mean ground temperature with a yearly cycle of the form

$$T_G(t) = T_{GM} - \Delta T_G \sin\left(\omega t - \phi\right) \tag{3}$$

where $T_{GM}$ is the mean ground temperature, $\Delta T_G$ is the annual temperature amplitude swing, $\omega$ is the frequency of oscillation, and $\phi$ is the phase shift of the thermal wave. According to EN 15241, the mean annual ground temperature, $T_{GM}$, can be written as follows:

$$T_{GM} = gmT_{AM} \tag{4}$$

where $gm$ is a ground material correction factor given in Table A.1 of EN 15241, and $T_{AM}$ is the annual mean air temperature. EN 15241 then gives the temperature oscillation at a depth $d$ as

$$\Delta T_G \sin(\omega t - \phi) = gm\Delta T_A AH \sin\omega(JH - VS + 24\cdot 25) \tag{5}$$

where $\Delta T_A$ is the amplitude of the annual outside air swing, $AH$ is an amplitude depth correction factor, $\omega = 2\pi/8{,}760$ is the hourly frequency of one year in rads/hr, $JH$ is the hour of the year (i.e., the number of hours since midnight on January 1), and $VS$ is a depth-dependent phase shift (in hours) for the thermal wave into the ground. The annual temperature swing, $\Delta T_A$, is half the difference between the maximum and minimum average monthly temperatures (occurring usually July and January in North America). The phase shift $\phi$ in Eqn. (3) is then identified as

$$\phi = \omega\left(VS - 24\cdot 25\right) = \omega\left(VS - 600\right). \tag{6}$$

The amplitude depth correction factor $AH$, in hours, is given in EN 15241 as

$$AH = 1 - 0.1993d + 0.01381d^2 - 0.000335d^3 \tag{7}$$

where $d$ is the depth, in meters, at which the soil temperature calculation is desired. The phase shift $VS$, in hours, is given by

$$VS = 24\left(0.1786 + 10.298d - 1.0156d^2 + \ldots \\ 0.3385d^3 - 0.0195d^4\right). \quad (8)$$

To compute the monthly average heat transfer to/from air in the tube, we will also need the monthly average ground temperature $T_{Gmon}$. $T_{Gmon}$ can be obtained by integrating Eqn. (3) over a month. If the month is $2\delta_m$ hours long, and $\alpha_m$ is the hour of the year for the middle of the month, that integral will result in

$$\begin{aligned} T_{Gmon} &= T_{GM} + \int_{\alpha_m-\delta_m}^{\alpha_m+\delta_m} \Delta T_G \sin(\omega t - \phi) \\ &= T_{GM} + \Delta T_G \frac{\sin(\omega\delta_m)}{\omega\delta_m} \sin(\omega\alpha_m - \phi). \end{aligned} \quad (9)$$

Figure 2 shows a plot of $\Delta T_G \sin(\omega t - \phi)$ for depths of $d = 0$ m (i.e., at the ground surface) to $d = 4$ m for a case where $\Delta T_A = 14$°C. The data show that the amplitude of the temperature swing drops and the phase delay increases with increasing depth $d$. The amplitude oscillation, $\Delta T_G$, becomes negligible at depths greater than about 4 m, and then soil temperature is a constant value of $T_G \approx T_{GM}$.

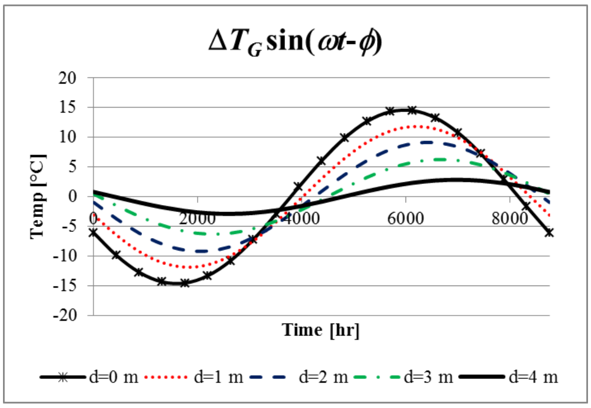


*Figure 2 Plot of the soil temperature swing, $\Delta T_G$, about $T_{GM}$ for depths d = 0 m to d = 4 m and $\Delta T_A$ = 14°C*

### Heat Transfer and Pressure Drop in Tube

Unless otherwise noted, the equations in this section can be referenced back to the ASHRAE *Fundamentals* Handbook (ASHRAE, 2009).

The overall heat transfer coefficient per unit length from the soil to the air in the tube, $U_t$, is given by

$$U_t = \left(\frac{1}{h_c} + \frac{1}{2\pi k_t} \ln\frac{r_o}{r_i}\right)^{-1} \quad (10)$$

where $k_t$ is the thermal conductivity of the tube, $r_o$ and $r_i$ are the outer and inner radii of the tube, and $h_c$ is the convective heat transfer coefficient from the inside of the tube to the air. This equation assumes that the thermal resistance between the soil at temperature $T_G$ and the duct is negligible; however, that resistance can be taken into account by adding the thermal resistance per-unit-length to the other two in Eqn. (10).

To find $h_c$, we use the standard form for forced convection heat transfer coefficient of

$$h_c = \frac{\text{Nu}}{2k_a r_i} \quad (11)$$

where Nu is the Nusselt number and $k_a$ is the thermal conductivity of the air. For this work, we compute Nu using the Gnielinski correlation (Gnielinski, 1976) so

$$\text{Nu} = \begin{cases} \dfrac{f/8(\text{Re}-1000)\text{Pr}}{1+12.7\sqrt{f/8}\left(\text{Pr}^{2/3}-1\right)} & \text{Re} > 2300 \\ 3.66 & \text{Re} < 2300 \end{cases} \quad (12)$$

where Re is the Reynolds number for the air flow in the tube, Pr is the Prandtl number of the air, and $f$ is the friction factor. For this interior tube problem, the Re for air at a velocity $v_a$ with dynamic viscosity $\upsilon_a$ will be

$$\text{Re} = \frac{2v_a r_i}{\upsilon_a}. \quad (13)$$

The friction factor we use here is the newly developed correlation of Ghanbari et. al. (Ghanbari et al., 2011), who give $f$ as

$$f = \frac{0.4033}{\log\left(\left(\dfrac{\varepsilon}{14.42r_i}\right)^{1.042} + \left(\dfrac{2.731}{\text{Re}}\right)^{0.9152}\right)^{2.169}} \quad (14)$$

where $\varepsilon$ is the tube roughness factor. The effect of surface roughness can be significant as $f$, Nu, and $h_c$ can change by a factor of two or more if a design changes from a smooth PVC tube with $\varepsilon$ =1.5 μm to a rough concrete tube with $\varepsilon$ =1 mm.

Once $U_t$ has been computed, the temperature of the air at the end of the tube is given as

$$T_{a,L} = T_G + \left(T_{a,in} - T_G\right)e^{-2\pi r_i L \frac{U_t}{m_a C_{p,a}}} \quad (15)$$

where $C_{p,a}$ is the specific heat of the air (usually evaluated at $T_{a,in}$ or the average of $T_G$ and $T_{a,in}$). The monthly average, $T_{a,Lm}$, can be obtained by using monthly average temperatures in Eqn. (15).

The instantaneous rate of heat transfer from the ground to the air in each tube, $Q_t$, is then given as

$$Q_t = 2\pi r_i L\left(T_G - \tfrac{1}{2}\left(T_{a,in} + T_{a,L}\right)\right). \quad (16)$$

The average monthly heat transfer, $Q_{t,mon}$, can be obtained by using monthly average temperatures in Eqn. (16).

From Eqn. (15), we can define the nondimensional heat transfer unit, NTU, as

$$\mathrm{NTU} = \frac{2\pi r_i L U_t}{m_a C_{p,a}} \tag{17}$$

and the EAHE efficiency, $\eta$, as

$$\eta = \frac{T_{a,L} - T_{a,in}}{T_G - T_{a,in}} = 1 - e^{-\mathrm{NTU}}. \tag{18}$$

Finally, the pressure drop, $\Delta p$, at the end of the tube will be

$$\Delta p = \rho_a f \frac{v_a^2}{4r_i} L. \tag{19}$$

Because tubes will also have bends, the pressure drop from the bends should be included in estimates of the total pressure drop. The effect of bends can be significant if the required main tube length is small. The pressure drop across a bend can be estimated as

$$\Delta p_b = \rho_a C_{loss} \frac{v_a^2}{2} \tag{20}$$

where $C_{loss}$ is the loss coefficient for the bend. Because the tube will most likely be circular in cross section and the bends will most likely be 90° right angles, an equation for $C_{loss}$ as a function of tube diameter can be obtained from fitting the measured data presented in the ASHRAE Handbook (ASHRAE 2009) to a quadratic equation. The results we obtained are

$$C_{loss} = 0.09057 - 0.001.439d + 0.001294d^2 \tag{21}$$

where $d$ is the tube diameter in meters. The air fan power, AFP, required to move the volume of air $V$ across the total pressure drop $\Delta p$ is then calculated as

$$AFP = V\Delta p\,. \tag{22}$$

Because both NTU and $\Delta p$ are proportional to the length of the tube, we can use NTU/$L$ and $\Delta p/L$ as the main performance measures that will help us determine the required length of tube for design purposes.

### Design Equations

From the analysis equations shown above, development of design equations is fairly straightforward. We assume the designer has selected a required volume flow rate, tube diameter, depth in soil, and number of parallel ducts. The designer must also input a temperature for calculation of the fluid properties (we find that $T_{AM}$ is a good value to use). Finally, the designer selects a heat exchanger efficiency goal, and the required duct length and pressure drop across the duct can then be computed. Use of heat exchanger efficiency as a design goal means that the ground temperature need not be computed until the designer wants to estimate the actual rate of heat transfer from the tube to the air.

Eqn. (18) is easily inverted so that the required NTU is given by

$$NTU = -\ln(1-\varepsilon) \tag{23}$$

and the required length of tube $L_t$ is obtained from Eqn. (17) as

$$L_t = \mathrm{NTU}\frac{m_a C_{p,a}}{2\pi r_i U_t}. \tag{24}$$

The pressure drop in the pipe is then computed from Eqns. (19) and (20), and the air fan power is computed from Eqn. (22), which can be used to help in fan selection.

## SPREADSHEET IMPLEMENTATION

We have implemented our design and analysis equations in a Microsoft Excel spreadsheet. We have avoided using Visual Basic or Macros in the sheet for increased portability to other spreadsheet programs, and, as a result, the spreadsheet does not utilize *Goal Seek* or *Solver* for optimization. In our spreadsheet implementation, we have two main sheets: a heat exchanger (HX) design sheet and an HX analysis sheet.

### HX Design Sheet

The HX design sheet limits the inputs to the desired HX effectiveness, the tube's inner diameter (in inches), tube wall thickness (in inches), air flow volume (in cubic feet per minute [cfm]), the number of parallel tubes, the tube material (PVC, concrete, or steel), and the mean annual air temp in ºC. The spreadsheet uses mostly IP inputs reflecting the predominant use of IP units for design in the United States, although inputs are converted to SI and internal calculations are performed in SI. The mean temperature is input in Celsius because those SI data are easily found. The selection of PVC, concrete, or steel as the tube material sets the values of $k_t$ and roughness $\varepsilon$ from a table of values taken from the ASHRAE Handbook (ASHRAE 2009), which are as listed below in Table 1. Because it is a spreadsheet, users can easily add the properties of additional materials, as desired.

*Table 1: Tube material properties used in analysis*

| Material | $k_t$ [W/(mK)] | $\varepsilon$ [m] |
|---|---|---|
| Concrete | 1.0 | 1.0E-3 |
| PVC | 0.19 | 1.5E-6 |
| Carbon Steel | 54 | 1.5E-6 |

The main outputs of the sheet are the required length of tubing; pressure drop across the tubing Δ*p* (neglecting pressure drop of bends); the overall heat transfer coefficient from soil to air in the tube $U_t$; the NTU; and $J = \Delta p/\mathrm{NTU}$, which is a figure of merit comparing the pressure drop (which is related to required fan power) and the heat transfer, with lower *J* resulting in a higher-performing design. A screen capture of the HX design sheet inputs and outputs is shown in Figure 3.

| Duct Info Input | |
|---|---|
| HX effect | 50% |
| $D_{tube}$ [in] | 12 |
| $\Delta R_{tube}$ [in] | 0.375 |
| $V_{air}$ [cfm] | 10600 |
| $N_{tubes}$ | 4 |
| Tube Material | PVC |
| TAM [°C] | 10 |
| *TAM is used here only for fluid property calculation | |

| Heat Transfer Output | |
|---|---|
| $L_{tube}$ [m] | 84.4 |
| $L_{tube}$ [ft] | 276.9 |
| ΔP (Pa) | 716.5 |
| ΔP (in wg) | 2.88 |
| $U_t$ [W/m²·K] | 13.4 |
| NTU/L | 0.01 |
| ΔP/L (Pa) | 8.5 |
| NTU desired | 0.69 |
| J=ΔP/NTU (Pa) | 1033.64 |

| Duct Input Conversion | |
|---|---|
| $D_{in,tube}$ [m] | 0.305 |
| $D_{out,tube}$ [m] | 0.324 |
| $V_{tube}$ [m³/s] | 1.25 |

| Calculation Outputs | |
|---|---|
| $v_{tube}$ [m/s] | 17.12 |
| $h_c$ [W/m2K] | 42.38 |
| Re | 3.69E+05 |

*Figure 3 Screen capture of the HX design sheet*

**HX Analysis Sheet**

The HX analysis sheet augments the inputs of the HX design sheet by adding the length of the tube input, the number of bends in the tube, monthly mean air temperatures, and a choice of soil type (moist soil, dry sand, moist sand, moist clay, and wet clay) as inputs. The choice of soil is used to set the soil correction factor, *gm*, to the appropriate value given in EN 15241.

The HX analysis sheet augments the outputs of the HX design sheet with estimates of the pressure drop including bends, the air fan power required to move the air, the average monthly ground temperatures, the average monthly air temperatures out of the HX tube, and the monthly average rate of heat transfer to/from the air. A screen capture of part of the HX analysis sheet is shown in Figure 4.

The HX analysis sheet also has another input for single-day analysis. This functionality would be applicable to finding the heating or cooling potential of the EAHE. For that use, a designer inputs a particular day of the year (which is used to compute the ground temperature) and an air tube inlet temperature (i.e., the ambient air temperature), and the spreadsheet computes the instantaneous air temperature at the tube output and the instantaneous rate of heat transfer to/from the air in the duct. A screen capture of that part of the spreadsheet is shown in Figure 5.

| Tube Info Input | |
|---|---|
| $D_{tube}$ [in] | 12 |
| $\Delta R_{tube}$ [in] | 0.375 |
| $L_{tube}$ [ft] | 277 |
| $V_{air}$ [cfm] | 10600 |
| $N_{tubes}$ | 4 |
| $N_{bends}$ | 2 |
| Tube Depth [ft] | 6 |
| Tube Material | PVC |
| Tmean [°C] | 10 |
| Soil type | Wet Clay |

| HX Analysis Output | |
|---|---|
| $h_c$ [W/m²·K] | 43.38 |
| $U_t$ [W/m²·K] | 13.54 |
| NTU | 0.70 |
| HX Effect. | 0.50 |
| $v_{tube}$ [m/s] | 1.71E+01 |
| ΔP (Pa) | 753.3 |
| ΔP (in wg) | 3.03 |
| AFP [W] | 3769 |
| AFP [HP] | 5.0 |

| Monthly Temperature Input | Jan | Feb | Mar | Apr |
|---|---|---|---|---|
| Monthly $T_{mean}$ | -4.4 | -2 | 3.1 | 9.3 |

| Monthly Averages | | | | |
|---|---|---|---|---|
| $T_{Gmon}$ [°C] | 6.83 | 3.01 | 1.11 | 1.61 |
| $T_{a,Lmon}$ [°C] | 1.25 | 0.52 | 2.10 | 5.44 |
| $\Delta T_{Amon}$ [°C] | 5.65 | 2.52 | -1.00 | -3.86 |
| $Q_{tmon}$ [kW] | 36.88 | 16.45 | -6.53 | -25.23 |
| $Q_{tmon}$ [kBtu/h] | 125.83 | 56.14 | -22.29 | -86.10 |

*Figure 4 Screen capture of part of the HX analysis sheet. Shown are tube information inputs, main HX results, monthly temperature inputs for January through April, and monthly average outputs*

| Single Day | |
|---|---|
| Date | 8/2/2012 |
| $T_{a,in}$ [°C] | 30 |
| Hour | 5148.00 |
| $T_G$ [°C] | 16.56 |
| $T_{a,L}$ [°C] | 23.25 |
| ΔT [°C] | -6.8 |
| $Q_t$ [kW] | -44.12 |
| $Q_t$ [kBtu/h] | -150.52 |

*Figure 5 Screen capture of the instantaneous performance calculator in the HX analysis sheet*

**Example Calculation**

As an example of how to use the calculator, consider the sizing of an EAHE for a building located near Chicago, Illinois, with a required ventilation of 5,000 L/s (10,600 cfm). Local tubing costs suggest that PVC tubing that is 30.5 cm in diameter (12 in.) is a good selection choice. That tubing normally has a wall thickness of about 9.5 mm (0.375 in.). The outdoor yearly average air temperature in Chicago is about 10°C.

Using the HX design calculator with a choice of one tube, the predicted required tube length of a single tube is 265 m (868 ft). If we instead use four tubes, the predicted required length is 84.4 m (277 ft) for each tube, as can be seen in Figure 3.

Putting the 84.4 m (277 ft) length into the HX analysis calculator and assuming two bends per tube, the HX analyzer predicts a pressure drop of 753 Pa (3.03 in wg), which would require an air fan power of 3.77 kW (5.0 HP) as shown in Figure 4. The sheet also predicts the monthly average air temperatures and average heat transfers (also shown in Figure 4).

Putting the information for a warm summer day into the instantaneous heat transfer calculator in Figure 5 shows that the tube can extract 44 kW of heat from the air, dropping the air temperature from 30°C to 23°C, which would greatly reduce the cooling load of the ventilation air.

**Comparison to Other Simplified Methods**

As stated earlier, both De Paepe and Janssens (2003) and Badescu and Isvoranu (2011) neglected the thermal resistance of the duct wall and used less accurate correlations for the friction factor and Nusselt number. To see the effects of our updated equations on design predictions, we can compare our predictions using our equations to predictions using others for the same PVC 30.5 cm tube discussed earlier as well as for steel tube of the same dimensions and a 30.5 cm concrete tube with a 50 mm wall thickness. The results listed in Table 2 show that for the PVC duct, the methods of Badescu and De Paepe, both of which neglect the thermal resistance of the tube wall, predict a much higher $U$ and therefore a much shorter required $L$ (a factor of 4 for Badescu and Isvoranu and 5 for De Paepe and Janssens) than the current method.

*Table 2: Comparison of predictions for PVC Pipe*

| Quantity | Current | Badescu | De Paepe |
|---|---|---|---|
| $f$ | 0.0141 | 0.0139 | 0.0139 |
| Nu | 520.5 | 516.4 | 812.4 |
| $U_t$ (W/m$^2$·K) | 13.4 | 42.0 | 66.1 |
| $L$ (m) | 84.4 | 27 | 17.2 |

The concrete tube has a thicker wall but one with higher thermal conductivity and increased surface roughness, which results in a slightly higher overalls $f$, Nu, and $U$, resulting in a decreased $L$ using the current method as seen in Table 3. However, the required length is still much longer than Badescu and Isvoranu or De Paepe and Janssens which, because surface roughness and wall thermal resistance are not included, yield the same tubing length as when thinner walled PVC was used.

*Table 3: Comparison of predictions for Concrete Pipe*

| Quantity | Current | Badescu | De Paepe |
|---|---|---|---|
| $f$ | 0.0274 | 0.0139 | 0.0139 |
| Nu | 1056.8 | 516.4 | 812.4 |
| $U_t$ (W/m$^2$·K) | 17.6 | 42.0 | 66.1 |
| $L$ (m) | 64.5 | 27 | 17.2 |

If the PVC tube was instead made of steel with such a high thermal conductivity that the thermal resistance of the tube wall is almost negligible, the only difference in length prediction results will be because of the differing correlations used in calculation of $f$ and Nu. As shown in Table 4, the predicted required length drops to 27 meters, matching the Badescu prediction.

*Table 4: Comparison of predictions for Steel Pipe*

| Quantity | Current | Badescu | De Paepe |
|---|---|---|---|
| $f$ | 0.0141 | 0.0139 | 0.0139 |
| Nu | 520.5 | 516.4 | 812.4 |
| $U_t$ (W/m$^2$·K) | 42.1 | 42.0 | 66.1 |
| $L$ (m) | 27.0 | 27.0 | 17.2 |

## CONCLUSIONS

A set of simplified design equations which are suitable for spreadsheet implementation have been developed for earth-air heat exchangers (EAHEs). The equations capture the effect of heat transfer through the sides of the tube and use recent correlations for the friction factor and Nusselt number. The equations were implemented in an Excel spreadsheet, and example screenshots of the implementation were shown.

The design sheet predicts the required length of tube for desired heat exchange efficiency. The analysis sheet predicts the monthly average and instantaneous heat transfer performance, the system pressure drop, and the required air fan power once a tube length has been selected. The importance of including heat transfer through the sides of the tube and heat transfer correlations that include surface roughness was shown by comparison to simplified methods that neglect those factors.

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## NOMENCLATURE

$AFP$ = air fan power
$AH$ = ground temperature amplitude depth correction
$C_{p,a}$ = specific heat of air
$C_{loss}$ = loss coefficient for bends in tube
$d$ = depth of tube
$f$ = fluid flow friction factor
$gm$ = ground material correction factor
$H_m$ = number of hours in the month
$h_c$ = heat transfer coefficient from tube wall to air
$JH$ = hour of the year
$k_a$,$k_t$ = thermal conductivity of air and tube
$L_t$ = length of tube
$m_a$ = mass flow rate of air in a single tube
$N_t$ = number of tubes used in system
NTU = number of heat transfer units
Nu = Nusselt number for air in tube
$\Delta p$, $\Delta p_b$ = pressure drop in tube and bends
Pr = Prandtl number for air in tube
$Q_t$, $Q_{t,mon}$ = instantaneous and monthly average heat transfer
Re = Reynolds number for air in tube
$r_i$,$r_o$ = inner and outer tube radius
$T_G$ = ground temperature at depth $d$
$T_{GM}$ = mean annual ground temperature at depth $d$
$T_{AM}$ = mean annual aboveground air temperature
$\Delta T_A$ = annual aboveground air temperature amplitude swing
$T_{a,in}$, $T_{a,L}$ = temperature of air entering and leaving tube
$TG_{mon}$, $T_{a,Lmon}$ = monthly averages of $T_G$ and $T_{a,L}$
$U_t$ = ground-to-air overall heat transfer coefficient
$V$ = total air volume flow rate
$v_a$ = velocity of air in tube
$VS$ = ground temperature phase shift
$\alpha_m$ = hour of the middle of the month
$\delta_m$ = half the number of hours in the month
$\eta$ = efficiency of tube heat exchange
$\phi$ = total ground temperature phase shift
$\rho_a$ = density of air in tube
$\upsilon_a$ = dynamic viscosity of air in tube
$\omega$ = hourly frequency of one year

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