Design of earth-air heat exchangers

A. Janssens
Department of Architecture and Urbanism, Ghent University,Belgium
M. De Paepe
Department of Flow, Heat and Combustion Mechanics, Ghent University, Belgium

Earth-air heat exchangers, also called ground tube heat exchangers, are an interesting technique to reduce energy consumption in a building. They can cool or heat the ventilation air, using cold or heat accumulated in the soil. Several papers have been published in which a design method is described. Most of them are based on a discretisation of the one dimensional heat transfer problem in the tube. Three dimensional complex models, solving conduction and moisture transport in the soil are also found. These methods are of high complexity and often not ready for use by designers. In this paper a 1 dimensional analytical method is used to analyse the influence of the design parameters of the heat exchanger on the thermo-hydraulic performance. A relation is derived for the specific pressure drop, linking thermal effectiveness with pressure drop of the air inside the tube. The relation is used to formulate a design method which can be used to determine the characteristic dimensions of the earth-air heat exchanger in such a way that optimal thermal effectiveness is reached with acceptable pressure loss. The choice of the characteristic dimensions, becomes thus independent of the soil and climatological conditions. This allows designers to choose the earth-air heat exchanger configuration with the best performance.

1 INTRODUCTION

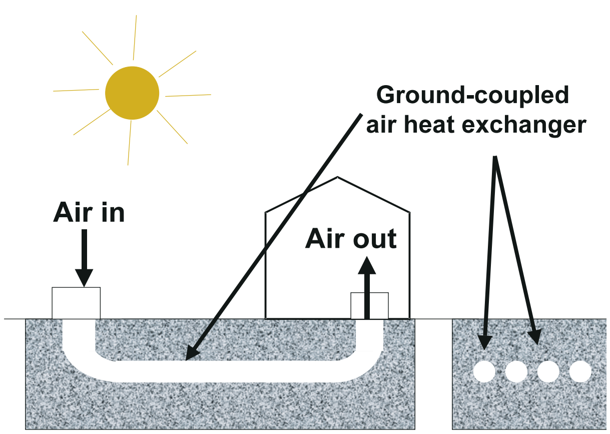

Figure 1: Ground-coupled heat exchanger

The Kyoto-protocol stimulates the world to reduce CO_2 production. This resulted in two important measures. First of all efforts are focused on producing electricity with higher efficiency. Old power plants are more rapidly phased out and replaced by new, more efficient plants. Secondly the attention was drawn on energy use. More efficient use of energy in production processes, buildings, etc., not only reduces the consumption of electricity, but also lowers the spending of primary energy sources.

Buildings, residential or offices, mainly use energy to obtain comfort for it's inhabitants. This comfort is visual, ergonomic, but mainly thermal. In order to reduce energy consumption of buildings, several passive techniques are nowadays introduced in HVAC-installations. In most cases solar energy is directly or indirectly used to supply heat or electrical energy. Sometimes solar gains inside the building are avoided to keep down the size of the air-conditioning unit. Other techniques are based on recovering heat or even 'cold'. In most cases several passive measures are combined.

An interesting and promising technology are earth-air heat exchangers, sometimes called ground tubes, or ground-coupled air heat exchangers. Tubes are put into the ground, through which air is drawn (see Figure 1). Because of the high thermal inertia of the soil, the temperature fluctuations at the ground surface exposed to the exterior climate, are damped deeper in the ground. Further a time lag occurs between the temperature fluctuations in the ground and at the surface. Therefore at a sufficient depth the ground temperature is lower than the outside air temperature in summer and higher in winter. When fresh ventilation air is drawn through the earth-air heat exchangers the air is thus cooled in summer and heated in winter. In combination with other passive systems and good thermal design of the building, the earth-air heat exchanger can be used to avoid air-conditioning units in buildings, which results in a major reduction in electricity consumption of a building.

In several European countries this technique is implemented in private houses and office buildings. Recent examples are found in Germany (Schuler 1999) and Switzerland (Zimmerman 1998). Architects and building installation designers are often interested in installing earth-air heat exchangers, but due to lack of knowledge or design criteria, the introduction of the heat exchanger in the building design is omitted. In literature several design strategies and calculation methods are discussed. Most of them are based on experiments or on 1, 2 or 3 dimensional calculation models. They are proven to be accurate, but mostly they are highly complex. Recent experience shows that these methods do not find their way from an academic level to the every day practice. Therefore this paper presents an analytical analysis of the earth-air heat exchanger based on the definition of the heat exchanger effectiveness. With this the important design parameters are studied : tube diameter, tube length and number of tubes. As a result the relative influence of these parameters can be analysed. This results in a design-map on which a proper selection of the design parameters can be made.

2 LITERATURE REVIEW

In the literature several calculation models for ground coupled heat exchangers are found. Tzaferis et al. studied eight models (Tzaferis et al.1992). The authors classified the algorithms in two groups:

1. The algorithms that first calculate the convective heat transfer from the circulating air to the pipe and then calculate the conductive heat transfer from the pipe to the ground inside the ground mass. The necessary input data are :
   - the geometrical characteristics of the system
   - the thermal characteristics of the ground
   - the thermal characteristics of the pipe
   - the undisturbed ground temperature during the operation of the system
2. Those algorithms that only calculate the convective heat transfer from the circulating air to the pipe. In this case the necessary input data are :
   - the geometrical characteristics of the system
   - the thermal characteristics of the ground
   - the temperature of the pipe surface

Six of the models use a steady-state one-dimensional description of the pipe. A relation between inlet and outlet temperature of the tube is derived. For all these models no influence of thermal capacity of the earth can be taken into account. Secondly the influence of different pipes on each other cannot be studied. In one algorithm the ground is divided into co-axial cylindrical elements. The thermal resistance of the ground is considered to be time-dependent. The pipe is divided in segments. In each segment the exit temperature is determined. In an other algorithm the steady-state heat balance is solved between a point in the ground and the tube. The authors conclude that the different models give comparable results. This is mainly caused by the fact that the models offer different solution methods and dicretisations of the same equations. The compliance with measurements done by Tzaferis et al. is quite good. This shows that a steady-state one-dimensional model may characterize the behaviour of the earth-air heat exchangers. This approach will be followed in this paper.

Mihalakakou et al. (Mihalakakou et al. 1994), Bojic et al. (Bojic et al. 1997), Ghautier et al. (Gauthier et al. 1997) and Hollmuller et al. (Hollmuller & Lachal 2001) have reported on more complete and dynamic models for earth-air heat exchangers. The models differ in the way the geometry is described (2D, 3D, polar coordinates) and in the way the effects of moisture transport in the ground and in the air are accounted for. However these four calculation methods are quite complex. They are mainly used to show that the heat exchanger is a promising and effective technology. Therefore the applicability for design is limited to people who are able to use the calculation codes. In most cases the earth-air heat exchangers are just one component in a whole buildings system. Designers do not have much freedom of choice to determine the size and lay-out of the heat exchanger. They are limited by space-constraints and economic boundary conditions. They need a simplified way to predict the general performance of the heat exchanger. Their main concern is to be able to select a reasonable size of the tube diameter, tube length and number of tubes. The most important question is whether adding another tube or another meter to the tubes, will result in an economic performance amelioration. These influences can be derived by the method discussed in this paper.

3 DESIGN PARAMETER ANALYSIS

3.1 *Design parameters of the earth-air heat exchanger*

3.1.1 *Specified parameters*

The earth-air heat exchanger should be sized in order to meet certain design requirements. For instance, during cold weather the ventilation air at the outlet of the earth-air heat exchanger should be heated above the freezing point to prevent icing of other heat recovery components in the ventilation system. Alternatively the air leaving the earth-air heat exchanger should deliver total or part of the building cooling load during a design summer day. These design requirements are achieved by heating or cooling the ventilation air in the ground tube from the external air temperature towards the ground temperature in the vicinity of the tube. Hence by the nature of the design problem the following parameters of the sizing problem are specified:

- $\dot{m}_{air}$: the air mass flow rate
- $T_{air,in}$: the inlet air temperature
- $T_{air,out}$: the desired outlet air temperature after the heat exchanger
- T_{ground} : the ground temperature

The air mass flow rate and the outlet air temperature are set by the design requirement. The inlet air temperature and the ground temperature follow from the design climate conditions for the problem.

The ground temperature is defined by the external climate and by the soil composition, its thermal properties and water content. The ground temperature fluctuates in time, but the amplitude of the fluctuation diminishes with increasing depth of the tubes, and deeper in the ground the temperature converges to a practically constant value throughout the year. Optimal depths are in the range of two to four meter (IEA Annex 28 1999).

3.1.2 *Dimensions of the heat exchanger*

The geometric sizing parameters of an earth-air heat exchanger are :

- D : the diameter of the tube
- L : the length of the tube
- n : the number of tubes in parallel in the heat exchanger

Once the number of tubes is known, the problem is reduced to determining the size of the tube. The thermo-hydraulic problem is then limited to 1 tube, with an air flow rate given by :

$$\dot{m}_{air,tube} = \frac{\dot{m}_{air,tot}}{n}. \tag{1}$$

For the designer these parameters have to be determined in such a way that the boundary conditions and the heat exchanger performance are met. This means that the location, the available space, the building design and economics induce restrictions to the choice of tube length and number of tubes. It is important to be able to evaluate the influence of the parameters on the performance. Different combination of these parameters will lead to the same thermal performance. So a second design criterion has to be introduced. Pressure loss of the flow through the tube is of main importance for the quantification of the fans in the ventilation system.

3.2 *Heat exchanger effectiveness and NTU*

In the earth-air heat exchanger air is the only heat transporting fluid. The heat released or absorbed by the air is flowing through the tube walls to the surrounding soil. If the contact of the tube wall with the earth is considered to be perfect and the conductivity of the soil is taken to be very high compared to the surface resistance, then the wall temperature at the inside of the tube can be assumed to be constant.

The total heat transferred to the air when flowing through a buried pipe can be written as :

$$\dot{Q} = \dot{m}_{air} c_{p,air} \left(T_{air,out} - T_{air,in}\right) \tag{2}$$

Due to convection between the wall and the air, the transferred heat can be also be written as :

$$\dot{Q} = hA\Delta T_{lm} \tag{3}$$

The logarithmic average temperature difference is given by $(T_{ground} = T_{wall})$:

$$\begin{aligned} \Delta T_{lm} &= \frac{\left(T_{air,in} - T_{wall}\right)\left(T_{air,out} - T_{wall}\right)}{\ln\left(\frac{T_{air,in}-T_{wall}}{T_{air,out}-T_{wall}}\right)} \\ &= \frac{T_{air,in} - T_{air,out}}{\ln\left(\frac{T_{air,in}-T_{wall}}{T_{air,out}-T_{wall}}\right)} \end{aligned} \tag{4}$$

Eliminating $\dot{Q}$ from (2) and (3) gives the exponential relation for the outlet temperature of the tube as function of the wall temperature and inlet temperature :

$$T_{air,out} = T_{wall} + \left(T_{air,in} - T_{wall}\right) e^{-\frac{hA}{\dot{m}_{air} c_{p,air}}} \tag{5}$$

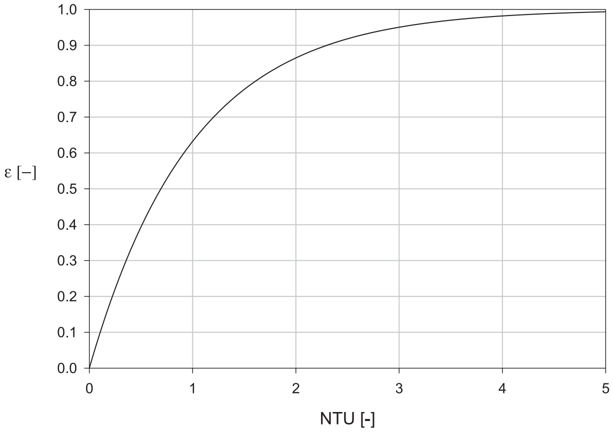

Figure 2: Effectivness as function of NTU

If a tube of infinite length $(A = \infty)$ is used the air will be heated or cooled to the wall temperature. The effectiveness of earth-air heat exchanger can thus be defined as :

$$\epsilon = \frac{T_{air,out} - T_{air,in}}{T_{wall} - T_{air,in}} \tag{6}$$

Using (5) the effectiveness becomes

$$\epsilon = 1 - e^{-\frac{hA}{\dot{m}_{air} c_{p,air}}}. \tag{7}$$

The non-dimensional group is called the Number of Transfer Units (NTU).

$$\text{NTU} = \frac{hA}{\dot{m}_{air} c_{p,air}} \tag{8}$$

which gives

$$\epsilon = 1 - e^{-\text{NTU}}. \tag{9}$$

The effectiveness of the heat exchanger is determined by the dimensionless group NTU. Figure 2 shows the change of the effectiveness as function of NTU. Increasing the NTU increases the effectiveness, though the curve rapidly flattens. After NTU > 3 the relative gain is small. There are several ways to construct an earth-air heat exchanger to obtain a given NTU and thus a desired effectiveness. The influence of the design parameters on NTU will now be studied.

3.3 *Influence on the heat transfer*

The NTU consists of three parameters which can vary :

- h : the convection coefficient of the air inside the tube
- A : the heat transfer surface of the tube
- $\dot{m}$: the air mass flow rate

The heat transfer area is a function of both D and L :

$$A = \pi D L \tag{10}$$

The convection coefficient inside the tube is defined by :

$$h = \frac{Nu\lambda}{D} \tag{11}$$

The Nusselt-number for flow inside a tube is given by (VDI 1994):

$$\begin{aligned} Nu &= 3.66 \quad \text{if } Re < 2300 \\ Nu &= \frac{\frac{\xi}{8}\,(Re - 1000)\,Pr}{1 + 12.7\sqrt{\frac{\xi}{8}}\left(Pr^{\frac{2}{3}} - 1\right)} \end{aligned}$$

$$\text{with } \ \xi = (1.82 \log Re - 1.64)^{-2}$$

$$\text{if } 2300 \le Re < 5\ 10^6 \text{ and } 0.5 < Pr < 10^6$$

The first equation applies to fully developed laminar flow, the second equation applies to turbulent flow in tubes with smooth internal surfaces.

The Reynolds number is related to average air velocity and diameter :

$$Re = \frac{v_{air} D}{\nu_{air}} \tag{12}$$

The mass flow rate is given by :

$$\dot{m}_{air} = \rho_{air} \frac{\pi D^2}{4} v_{air} \tag{13}$$

The length L is an independent parameter influencing the NTU. There is a linear variation of NTU with length.

Changing the diameter D or the mass flow rate $\dot{m}$ both change the air velocity inside the tube. This results in a changing Reynolds-number Re. D and $\dot{m}$ have thus no independent influence on the NTU. As NTU varies linearly with L, the parameter $\frac{\text{NTU}}{L}$ is only depending on D and $\dot{m}$.

This gives :

$$\begin{aligned} \frac{\text{NTU}}{L} &= \frac{hA}{L\dot{m}_{air} c_{p,air}} = \frac{Nu\,\lambda_{air}}{D} \frac{\pi D L}{\rho_{air} \dot{V} c_{p,air} L} \\ &= Nu \pi \frac{\lambda_{air}}{c_{p,air} \rho_{air}} \frac{1}{\dot{V}} \end{aligned} \tag{14}$$

For laminar flows Nu is constant, so $\frac{\text{NTU}}{L}$ is independent of the diameter D. In Figure 3 the contour

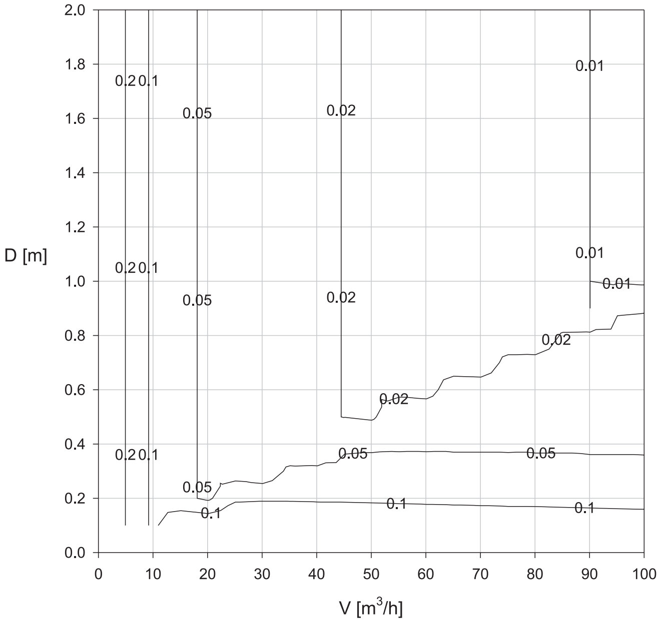

Figure 3: Contour plot of $\frac{\mathrm{NTU}}{L}$ as function of diameter and volume flow rate

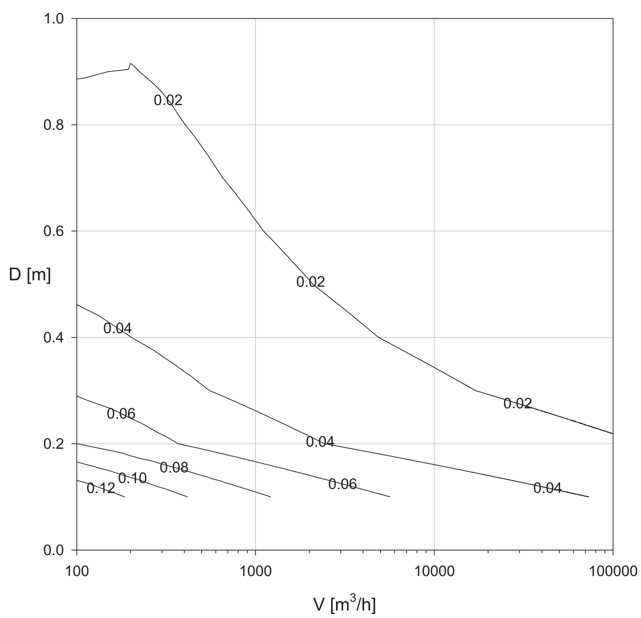

Figure 4: Contour plot of $\frac{\mathrm{NTU}}{L}$ as function of diameter and volume flow rate

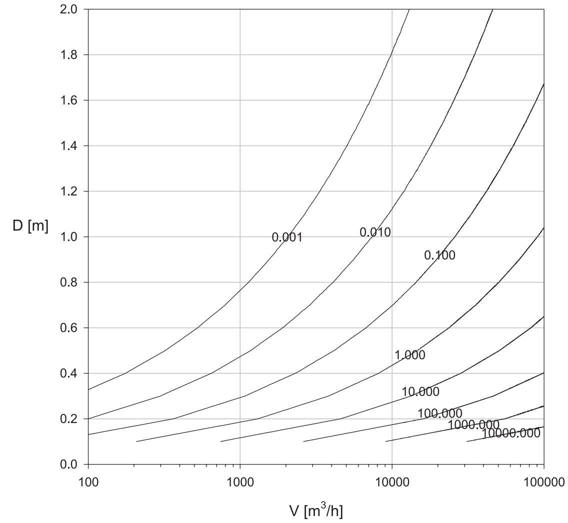

Figure 5: Pressure drop $\frac{\Delta p}{L}$ as function of diameter and air volume flow rate

lines for $\frac{\mathrm{NTU}}{L}$ are given for small flow rates and large tube diameters. When the flow is laminar the contour lines become vertical lines.

In Figure 4 the contour plots for $\frac{\mathrm{NTU}}{L}$ with changing range of D and volume flow rate $\dot{V}$ for the turbulent flow case are shown.

In general, lowering D raises the effectiveness, higher flow rates reduce the effectiveness. So it is better to have several tubes of small diameter over which the flow rate is divided.

Long tubes with a small diameter are profitable for the heat transfer. They however raise the pressure drop in the tubes, resulting in high fan energy.

3.4 *Influence on pressure drop*

The pressure drop in a smooth tube is given by :

$$\Delta p = \xi \frac{L}{D} \rho \frac{v_{air}^2}{2} \tag{15}$$

with (VDI 1994)

$$\xi = \frac{64}{Re} \quad \text{if } Re < 2300$$

$$\xi = (1.82 \log Re - 1.64)^{-2} \quad \text{if } Re \geq 2300$$

The tube length L is again an independent parameter influencing pressure drop. The influence is linear.

Diameter and flow rate have a combined influence. In Figure 5 the contour plots of pressure drop per unit of length $\frac{\Delta p}{L}$ for varying diameter and flow rate are shown. Having a small flow rate per tube and a large diameter gives the least pressure loss. This would mean using many tubes, with a large diameter. This is in conflict with the thermal demand of a small diameter. In both cases a large number of tubes is beneficial.

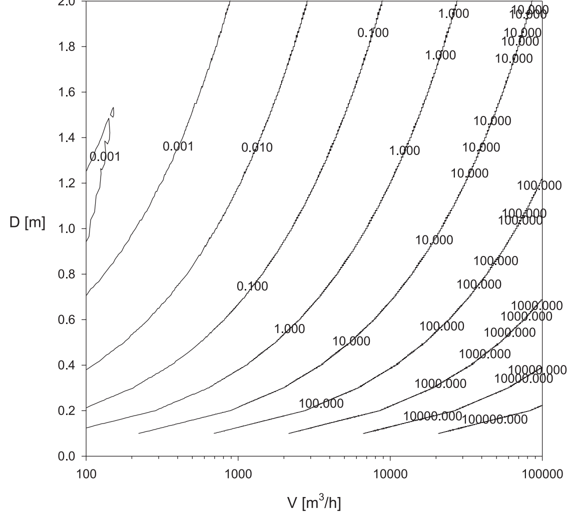

Figure 6: Specific pressure drop J as function of diameter and air volume flow rate

The tube length and diameter combination has to be optimized.

4 HEAT EXCHANGER DESIGN CALCULATIONS

4.1 *Specific pressure drop*

As both NTU and pressure drop are linear with the tube length, the parameters $\frac{\text{NTU}}{L}$ and $\frac{\Delta p}{L}$ are only dependent of diameter and volume air flow rate. These two parameters are respectively a measure for the thermal and hydraulic performance per unit of length.

Total NTU and total pressure drop of the heat exchanger are given by multiplying these unit parameters with the total length of the tubes :

$$\text{NTU}_{tot} = \left(\frac{\text{NTU}}{L}\right).L \quad (16)$$

$$\Delta p_{tot} = \left(\frac{\Delta p}{L}\right).L \quad (17)$$

The specific pressure drop J can be defined as :

$$J = \frac{\Delta p}{\text{NTU}} \quad (18)$$

J is a measure for the pressure drop necessary to realize one unit of NTU.

In Figure 6 J is given as function of $\dot{V}$ and D, as calculated with equations (18), (15) (13) and (8). J grows with growing $\dot{V}$ and smaller D.

The NTU per unit of length can be expressed as function of J and pressure drop per unit of length as :

$$\frac{\text{NTU}}{L} = \frac{1}{J}\frac{\Delta p}{L} \quad (19)$$

4.2 *NTU-J design method*

Given the desired temperature programme the effectiveness ϵ can be calculated with equation (6). With the inverse of equation (9) the desired NTU can be determined with equation

$$\text{NTU} = -\ln(1-\epsilon). \quad (20)$$

This NTU is the minimal desired NTU the heat exchanger has to reach : NTU_{min}.

Given the maximal allowable pressure drop Δp_{MAX}, the maximal allowable J of the heat exchanger can be determined as :

$$J_{MAX} = \frac{\Delta p_{MAX}}{\text{NTU}_{min}} \quad (21)$$

This J_{MAX} defines a zone in figure 6 which is not allowed for $\dot{V}$ and D. The choice of D and $\dot{V}$ per tube is thus limited.

Combining this with figures 3 and 4 , the choice for the most effective heat exchanger is the one with the smallest tube diameter and the smallest flow rate per tube. This allows for the choice of D and the maximum number of parallel tubes.

After the number of tubes in parallel is chosen and the tube diameter is accordingly determined the NTU per unit of length can be determined by (14). Given the desired NTU, this gives the length.

4.3 *Graphical design method*

The $\text{NTU} - J$ design method is used to derive a graphical sizing tool which allows to define the tube length and diameter from the air flow rate per tube and the desired ground tube performance J. The relationship between the specific pressure drop J and the air volume flow rate follows from equations (14) , (15) and (18) :

$$J = 0.258\frac{c_{p,air}\rho_{air}^2\xi}{\lambda_{air}NuD^5}\dot{V}^3 \quad (22)$$

This relation is represented in the lower half of Figure 6 on a log-log scale, for different values of the tube diameter.

The relationship between the tube length and the specific pressure drop J follows from equations (15), (19), (20) and (22):

$$L = \frac{J\ \text{NTU}}{\frac{\Delta p}{L}} = -\ln(1-\epsilon)\left[\frac{c_{p,air}^2\rho_{air}D^5}{8\lambda_{air}^2\xi Nu^2}\right]^{\frac{1}{3}} J^{\frac{1}{3}} \quad (23)$$

This expression may be solved when a value for the effectiveness ϵ of the earth-air heat exchanger is chosen. The desired effectiveness follows from the design requirements and climate conditions (equation (7)), but often an effectiveness of 80% is considered to be an optimum for an earth-air heat exchanger (IEA Annex 28 1999). A higher effectiveness is only achievable at the cost of an important increase of the tube length or of the number of tubes. Equation (23) is represented in the upper half of Figure 7, for different values of the tube diameter and a heat exchanger effectiveness of 80%.

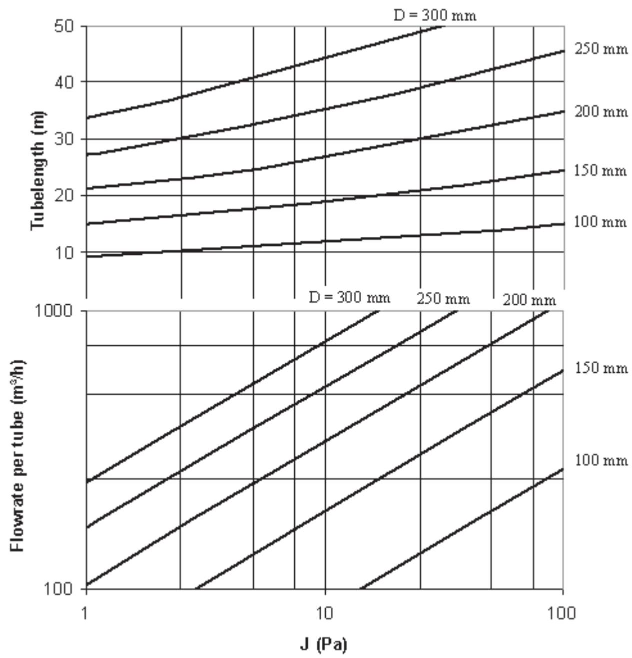

Figure 7: Design graph for the sizing of earth-air heat exchangers with a heat exchanger effectiveness of 80 %

The use of the design graph is now illustrated with an example. The boundary conditions for the sizing of an earth-air heat exchanger for a small office building are the following:

- The design air flow rate is 750 m^3/h.
- The geometry of the building site limits the length of the earth-air heat exchanger to a maximum of 25 m.
- The pressure drop across the heat exchanger should not exceed 100 Pa.

The pressure loss requirement defines the maximum allowable specific pressure drop J. Assuming a heat exchanger effectiveness of 80%, equation (21) gives: $J < 62\ Pa/m$. This value, together with the requirement for the maximum individual tube length, delimits a zone of allowable L and J in the upper half of Figure 7. The maximum tube length depends on the lay-out of the tubes on the available strip of land (Figure 8). The maximum tube length in the example is 25 m for a bundle of parallel tubes, but may be a multiple of 25 m for a serpentine shape. To prevent interference between the individual tubes, the distance between them should be at least 1 m (IEA Annex 28 1999).

From the zone of allowable L and J in Figure 7 a suitable tube diameter may be chosen, closest to the value of maximum specific pressure drop. Following the lines of equal specific pressure drop J to the line for the chosen diameter in the lower half of the graph, the air volume flow rate per tube is defined. The ratio between the design air flow rate and the flow rate per tube thus defines the number of tubes of a specific diameter and length to be installed. Table 1 shows all possible combinations which are a solution to the design problem of the example.

5 CONCLUSION

The design of an earth-air heat exchanger is a separate problem of the building design. Once the ventilation demands are known, the thermo-hydraulic design of the heat exchanger only depends on the constructional constraints and economics.

Three dimensions have to be determined : tube length, tube diameter and number of parallel tubes. Thermal performance and pressure drop both grow with length. Smaller tube diameters give better thermal performance, but also larger pressure drop. More tubes in parallel both lower pressure drop and rise thermal performance.

To be able to reduce the influencing parameters the specific pressure drop is introduced. The specific pressure drop is a measure for the pressure drop needed to realize a given thermal performance. This way a maximal specific pressure drop can be calculated.

The specific pressure drop is used to derive a straight forward design method.

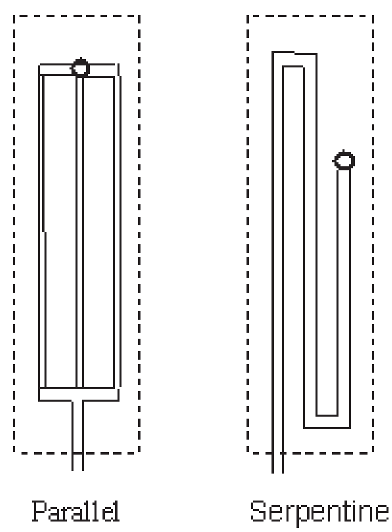

Figure 8: Lay-out of groundtubes om a strip of land

$D\ (mm)$	Number	$L\ (m)$	$v\ (m/s)$	$\Delta p\ (Pa)$	lay-out
100	4	14	6.6	77	Parallel
150	2	22	5.9	61	Parallel
200	3	25	2.2	8	Parallel
250	1	38	4.2	32	Serpentine

Table 1: Heat exchanger configurations for design air flow rate 750 m^3/h, ϵ = 80 %, $\Delta p < 100\ Pa$

NOMENCLATURE

A	:	heat transfer area [m^2]
c_p	:	thermal capacity [J/kgK]
D	:	tube diameter [m]
h	:	convection coefficient [W/m^2K]
L	:	tube length [m]
$\dot{m}$	:	mass flow rate [kg/s]
n	:	number of tubes [-]
NTU	:	Number of Transfer Units [-]
Nu	:	Nusselt number [-]
Pr	:	Prandtl number [-]
Δp	:	pressure drop [Pa]
$\dot{Q}$	:	transferred heat [W]
Re	:	Reynolds number [-]
T	:	temperature [K]
ΔT_{lm}	:	logarithmic temperature difference [K]
v	:	velocity [m/s]
$\dot{V}$	:	volume flow rate [m^3/s]

Greek

ϵ	:	heat exchanger effectiveness [-]
λ	:	thermal conductivity [W/mK]
ν	:	kinematic viscosity [m^2/s]
ρ	:	density [kg/m^3]

Subscribt

air	:	of the air
in	:	the inlet of the heat exchanger
out	:	the outlet of the heat exchanger
$ground$	:	of the ground
$wall$	:	of the tube wall

REFERENCES

Schuler, M., 1999. Design for Daylighting and Energy in Ingolstadt, Advanced Building Newsletter 23/24, pp. 18-23.

Zimmermann, M., 1998. The Schwerzenbacherhof Office and Industrial Building Swerzenbach, Switzerland, Ground Cooling (Air), IEA Low Energy Cooling Demonstration 15/1-15/8.

Tzaferis, A., Liparakis, D.,Santamouris, M., Argiriou, A., 1992. Analysis of the accuracy and sensitivity of eight models to predict the performance of earth-to-air heat exchangers, Energy and Buildings, Vol. 18 pp 35-43.

Mihalakakou, G., Santamouris, M., Asimakopoulos, D. 1994. Modelling the thermal performance of earth-to-air heat exchangers, Solar Energy, Vol. 53, No. 3 pp. 301-305.

Bojic, M., Trifunovic, N., Papadakis, G., Kyritsis, S., 1997. Numerical simulation, technical and economic evaluation of Air-to-earth heat exchanger coupled to a building, Energy, Vol. 22 No 12, pp 1151-1158.

Gauthier, C., Lacroix, M., Bernier, H., 1997. Numerical simulation of soil heat exchanger-storage systems for greenhouses, Solar energy, Vol 60 No 6 pp 333-346.

Hollmuller, P. & Lachal, B., 2001. Cooling and preheating with buried pipe systems : monitoring, simulation and economic aspects, Energy and Buildings, Vol 33 No 5, pp. 509-518.

IEA Annex 28 1999. Early design guidance for low energy cooling technologies. International Energy Agency, Energy Conservation in Buildings and Community Systems Programma, Annex 28 Low Energy Cooling, Subtask 2 Report 2.

VDI, Wärmeatlas, 1994, Springer Verlag.