SIMULATION OF EARTH-TO-AIR HEAT EXCHANGERS IN HYBRID VENTILATION SYSTEMS

Jian Zhang and Fariborz Haghighat

Department of Building, Civil and Environmental Engineering, Concordia University, Montreal, Quebec, Canada, H3G 1M8

ABSTRACT

Earth-to-Air Heat Exchanger (ETAHE) technology has been implemented in many mechanical ventilation systems. A literature review revealed that recent advancements in ETAHE modeling were focused on simulating heat conduction in soil. Air temperatures along the duct length were determined by calculating heat convection on the ETAHE duct surfaces using various empirical correlations. Comparison of the correlations for typical ETAHE in mechanical ventilation systems revealed that large discrepancy exists among them. A CFD simulation of an ETAHE applied to a hybrid ventilation building in Grong, Norway was conducted and the results confirmed field measurements. The convective heat transfer coefficient, obtained from the simulation, showed that the correlations significantly underestimate the heat convection in the ETAHE. Therefore, the existing models are not appropriate to simulate ETAHE in hybrid ventilation systems. Furthermore, a one-dimensional model was developed to predict the hybrid ventilation ETAHE system and provided satisfactory results.

INTRODUCTION

As an earth coupling strategy, ETAHE is designed to indirectly cool or heat buildings by ventilating outdoor air through a buried duct exploiting temperature gradients between outdoor air and the earth resulting in energy savings when conditioning an indoor environment. This principle has been used as early as in ancient Persian architecture (Bahadori 1978).

In conventional mechanical ventilation systems ETAHE is designed using small cross sectional area ducts. Average air velocities in the ducts are typically maintained at several meters per second using fans. A recent trend in the development of mechanical ventilation systems is to minimize pressure drops in the whole systems to reduce fan energy consumptions. Natural ventilation systems seem to be an alternative, however, their driving forces, i.e. wind and/or buoyancy cannot always deliver satisfactory airflow rates. Therefore, hybrid ventilation systems, simultaneously or alternatively employing natural and mechanical driving forces, become an energy efficient option. Recent research conducted by the IEA ECBCS-Annex 35 showed that fan and cooling energy consumptions can be substantially reduced using hybrid ventilation systems (Heiselberg 2002). It is believed that the present greatest potential for achieving energy savings is to rationally integrate hybrid ventilation systems and building elements such as ETAHE. Challenges of the integration lie in developing methods to optimize the interaction of ETAHE and hybrid ventilation systems (Heiselberg 2004). The first hurdle is to seek appropriate methods capable of modeling ETAHE thermal performance.

A literature review revealed that a number of studies had been conducted for modelling ETAHE applied to mechanical ventilation systems using small cross sectional area ducts. However, implementing ETAHE technology with hybrid ventilation systems requires a duct with a much larger cross-sectional area to minimize the pressure drop and decrease the air velocity. This configuration change causes the heat and mass transfer in the duct to become very complicated to model. The objective of this study is to investigate the complex processes in ETAHE systems and to find appropriate modeling methods.

REVIEW OF ETAHE RESEARCHES

From extensive field measurements at various ground depths for several locations in the U.S., Kusuda and Achenkach (1965) found soil temperatures below 6 feet remained fairly constant throughout the year. Assuming ground temperature is a pure harmonic function, undisturbed soil temperature profiles can be mathematically represented as a function of time (Labs 1989 and Mihalakakou et al. 1992). Assuming soil temperatures are not affected by the presence of ETAHE, several simulation models have been developed. Most of them divide the duct into a number of control volumes along its running length. Heat balance for each control volume, namely heat loss/gain by the air is equal to heat gain/lost by the earth, is adopted to predict the outlet air temperatures. Tzaferis et al. (1992) compared experimental data with results from eight ETAHE models and there were some general agreements among them.

The mathematical expression generating undisturbed soil temperature profiles normally requires a few physical parameters as inputs, e.g. average annual soil surface temperature and amplitude of surface temperature variation. These parameters are usually obtained by conducting long-term experiments and they are not available for most locations. Therefore, theoretical analysis for the heat transfer in soil became a useful method to obtain soil temperature distribution.

According to a model developed by Bansal et al. (1983), annual soil temperatures in New Delhi, India, at 4m depth remain relatively constant for various soil properties and surface conditions. However, significant differences existed among the constant values when different soil properties and surface conditions were selected. The constant temperature at 4m depth can be 17°C for wet shaded surface or as high as 52°C for dry glazed surface under the same climate. In other words, soil properties and surface conditions could have great effects on ETAHE thermal performance. Mihalakakou et al. (1996) concluded that earth surface conditions might be a significant controllable factor for the improvement of the ETAHE performance. Mihalakakou et al. (1997) presented a model for predicating daily and annual variations of ground surface temperatures. The model is based on an energy balance equation at the ground surface using a transient heat conduction equation and boundary conditions. The overall analysis was used for predicting the thermal performance of ETAHE.

To improve the simulation accuracy for ETAHE duct surface temperatures, a number of models were developed focusing on solving heat transfer in the ground. Their common algorithm was to simultaneously determine the heat conduction in a soil mass around the ETAHE duct and the heat convection inside the duct. Convective heat flux is used as boundary conditions for the two calculation domains at the soil and duct interface. Several differences exist among the various models, each taking into account specific considerations, such as:

- Effect of possible moisture condensation inside the ETAHE duct (e.g. Goswami and Ileslamlou 1990)
- Parallel multiple duct system configuration (e.g. Sodha et al. 1994)
- Effect of moisture transfer through soil on heat transfer (e.g. Kumar et al. 2003)
- Heat and moisture transfer in the soil along the axial direction (e.g. Benkert et al. 1997 and Hollmuller 2003)
- Techniques for solving the differential equations describing the combined heat and moisture transfer processes (Bojic et al. 1997 and De Paepe 2001)

HEAT CONVECTION IN ETAHE APPLIED TO MECHANICAL VENTILATION SYSTEMS

Although there are some differences among the models based on their respective assumptions, their principles are fundamentally the same. They all use Equation 1 to calculate convective heat flux between the duct surfaces and the air.

$$q'' = h\left(T_{air} - T_{wall}\right) \tag{1}$$

The h, convective heat transfer coefficient (CHTC), is a complex function of air velocity and temperature differences between the air and the duct surfaces. The heating or cooling capacity of ETAHE is determined by integrating the heat flux over the total internal duct surface area. Therefore, CHTC determination is critical to the prediction of the system’s capacity. Since analytical solutions of the air velocity and the air temperature profiles are usually not available, except for some very simple scenarios, most CHTC in ETAHE are obtained experimentally. In the existing ETAHE models, a few empirical correlations have been adopted and they are all applicable for fully developed turbulent flow in smooth circular ducts.

Validity of the fully developed assumption

When a fluid is drawn into a horizontal circular duct with a uniform entry velocity distribution and a constant surface temperature, within the entrance length, the velocity and temperature profiles are developed simultaneously. Beyond the entrance length, the friction and the temperature difference at the surface are sensed by the fluid along the duct center line, resulting in fully developed flow. According to experimental studies for simultaneously developing airflow in circular ducts, Boelter et al. (1941) produced two empirical correlations as shown in Equation 2 and 3. They describe the dimensionless ratio of mean CHTC, measured from the duct inlet to a specific distance, over that of an infinitely long duct. The relation is plotted in Figure 1, for airflow conditions commonly implemented in actual ETAHE in mechanical ventilation systems for four Reynolds numbers.

$$\frac{\bar{h}}{\bar{h}_\infty} = 1 + \frac{1.4}{L/D} \text{ for } \frac{L_c}{D} < \frac{L}{D} < 60 \tag{2}$$

$$\frac{\bar{h}}{\bar{h}_\infty} = 1.11 \frac{\mathrm{Re}_D^{\,1/5}}{\left(L/D\right)^{4/5}} \text{ for } \frac{L}{D} < \frac{L_c}{D} \tag{3}$$

Where $L_c / D = 0.625 \mathrm{Re}^{1/4}$ is the number of diameters required for the friction factor to become constant.

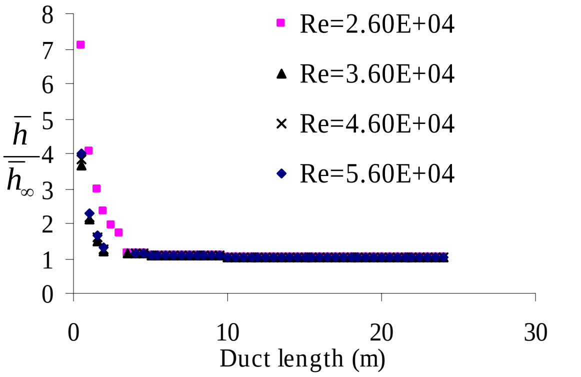

Figure 1 Entrance effect on convective heat transfer coefficient in circular ducts

Figure 1 shows that the turbulent airflows in typical ETAHE become fully developed with smaller CHTCs after approximately two meters from the inlet. All the ETAHE models neglect the entrance effect since it occurs in the vertical inlet section. Therefore, the CHTC correlations obtained from fully developed turbulent flow experiments are valid.

Comparison of correlations

Since all the empirical correlations used to predict the convective heat transfer in the ETAHE assumed fully developed turbulent airflow, ideally, they are expected to yield similar values for same operating condition. To exam this, a typical design of ETAHE ducts in mechanical ventilation systems is used to calculate the CHTC. The commonly used correlations are listed in Table 1.

Table 1 List of various CHTC correlations

#	CORRELATIONS	REFERRED BY
1	$h = \frac{3.6(v\rho)^{0.8}}{(2D)^{0.2}}$	Goswami and Ileslamlou 1990
2	$h = \frac{K}{D} 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.4}$	Singh 1994
3	$h = \frac{K}{D} 0.023\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{0.4}$	Sodha 1994a
4	$h = \frac{K}{D} 0.0214 \left(\mathrm{Re}^{0.8} - 100\right) \mathrm{Pr}^{0.4}$	Benkert et al. 1997
5	$h_{cooling} = \frac{K}{D} 0.011 \mathrm{Re}^{0.96} \mathrm{Pr}^{0.3}$	Bojic et al. 1997
6	$h = \frac{K}{D} 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.33}$	Hollmuller 2003
7	$h = \frac{K}{D} \frac{(\mathrm{Re} - 1000) \mathrm{Pr}\, \xi / 8}{1 + 12.7\sqrt{\xi/8}\left(\mathrm{Pr}^{2/3} - 1\right)}$	De Paepe 2001

Note $\xi = \left[1.82 \log(\mathrm{Re}) - 1.64\right]^{-2}$

The duct configuration and air properties selected for comparing the correlations are listed in Table 2.

Table 2 Air properties and design parameters in a typical ETAHE system

PARAMETERS	VALUE
Air conductivity	0.0242 W/m K
Air density	1.225 kg/m^3
Air viscosity	1.79E-05 kg/m s
Duct diameter	0.4 m
Duct length	30 m

Figure 2 shows the comparison of the CHTC produced by different correlations as a function of Reynolds number.

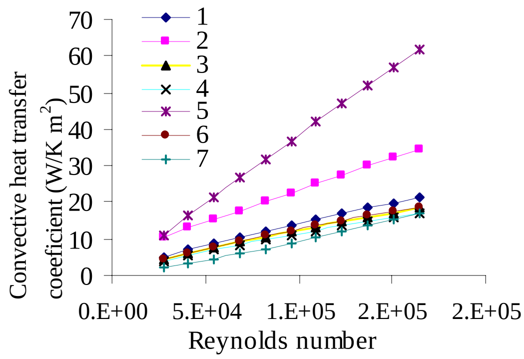

Figure 2 Comparison of CHTC correlations

As shown in Figure 2, five of the seven correlations yield similar values. Usually, slight variation of accuracy within approximate 20 percent is expected for the empirical correlations (Burmeister 1993). However, significantly greater values are obtained by the other two correlations used by Singh (1994) and Bojic et al. (1997). This might be attributed to different experimental conditions, which were adopted to derive the correlations, for example, the surface roughness of the experimental ducts. The large discrepancies indicate that appropriate correlation has to be selected if one uses any of the existing models to simulate the performance of the ETAHE.

HEAT CONVECTION IN ETAHE APPLIED TO HYBRID VENTILATION SYSTEMS

ETAHE simulation results are sensitive to the applied CHTC. A field study by Wachenfeldt (2003) on a large size hybrid ventilation ETAHE installed in a school building in Grong, Norway showed that the average measured CHTC was considerably larger than that produced from empirical correlations listed in Table 1 with a graphical comparison in Figure 3.

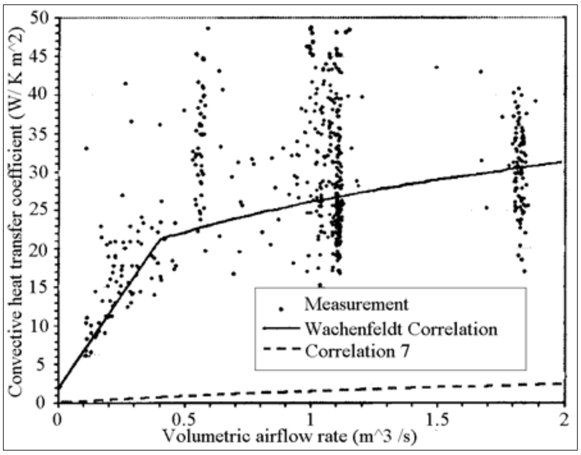

Figure 3 Comparison of measured CHTC from the hybrid ventilation ETAHE system versus correlations (Wachenfeldt 2003)

Reverse airflow direction in the top half duct and temperature stratification through the entire duct height were identified in the middle section of the duct. These two phenomena can be explained by the slow airflow velocity and the large temperature differences between the top and bottom surfaces of the ETAHE duct. The average air velocity was approximately 0.5m/s with the temperature difference of about 10°C. Thus, the vertical buoyancy force had the same order of magnitude as the horizontal force produced by the inlet and outlet pressure difference.

To perform an energy simulation on the building, Wachenfeldt (2003) curve fitted the measurement data of the heat transfer coefficients versus duct airflow rates. He proposed a correlation as shown in Figure 3 which is specific to the installed system. A general correlation or method describing heat convection in such a large size duct is in demand to accurately study ETAHE performance in hybrid ventilation systems.

CFD SIMULATION APPLIED TO ETAHE

Computational Fluid Dynamic (CFD), well known as a powerful method to study heat and mass transfer, has been used in building ventilation studies for many years (Li and Heiselberg 2003). It provides numerical solutions of partial differential equations governing airflow and heat transfer in a discretised form. Many studies had been conducted using CFD method to investigate heat convection on room surfaces (e.g. Awbi 1998 and Beausoleil-Morrison 2002). To exam the complicated airflow and heat transfer processes in an ETAHE system, commercial CFD software, AirPak v2.1, was used in this study.

Verification of CFD simulation for ETAHE

To exam the validity of CFD based ETAHE modeling, a simulation on an actual ETAHE in a mechanical ventilation system was conducted. The system is located in Mathura, India. The duct is 80m long with a cross sectional area of 0.53m2 and is buried 4m below the soil surface. Average air velocity at the duct outlet was maintained at 4.9m/s. The 24-hour inlet air temperature on June 15, 1983 is plotted in Figure 4. According to the model developed by Bansal et al. (1983), undisturbed soil temperature at a depth of 4m is approximated to be 21°C for a dry shaded soil surface condition and it is assumed to be the duct surface temperature.

The standard $k-\varepsilon$ turbulent model was used to simulate the airflow and heat transfer. The model was defined as a transient problem with every time step of half hour. The validity of the CFD simulation is shown in Figure 4 by comparing the simulated air temperature at the distances of 40m and 80m from the duct inlet with experimental results presented by Sodha et al (1985).

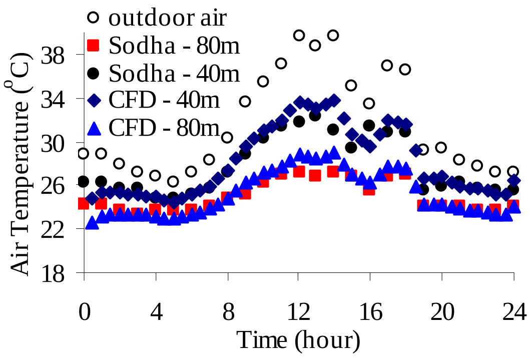

Figure 4 Comparison of CFD simulation results with field measurements for air temperature

Good agreement can be found from the temperature comparison with maximum errors of 8% and 6% at 40m-length and 80m-length, respectively. The simulated daily cooling capacity of the ETAHE is 506 kWh from the CFD analysis, and the results obtained by Sodha et al. (1985) and Kumar et al. (2003) for the same case are 512 kWh and 456 kWh respectively. The results indicate that the CFD method is capable of modeling the performance of ETAHE systems.

Application to hybrid ventilation systems

To investigate the ETAHE performance in hybrid ventilation systems, a CFD simulation was conducted on the duct installed in the Grong school building. As seen in Figure 5, the duct has a length of 15m, a height of 2.0m and a width of 1.5m. A 1m×1m outlet opening is located on the top side of the duct end section.

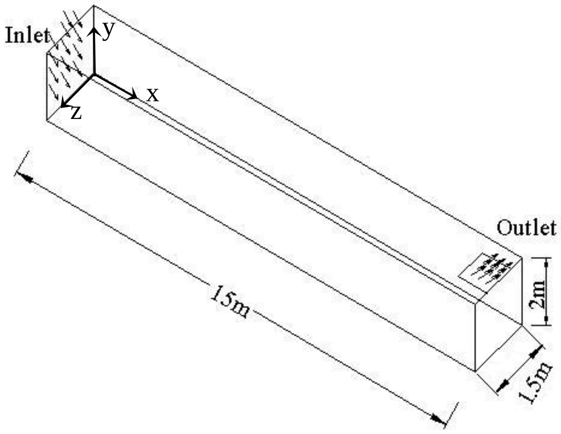

Figure 5 Schematic of the ETAHE in the hybrid ventilation system

Unlike small cross sectional area circular ducts in mechanical ventilation systems, in which all variables are usually considered to be symmetrical about the lengthwise central axis, temperature differences usually occur between the top and bottom duct surfaces in hybrid ventilation ETAHE. The surface temperatures are affected by three combined heat transfer processes, i.e. conduction in the soil, convection between the air and the duct, and radiation between different duct surfaces. Since detailed boundary condition data about the system is not available, the field observation data reported by Wachenfeldt (2003) and Heiselberg (2004) was used. Based on these references, the top, bottom, and wall surface temperatures are respectively assumed to be 10°C, 18°C, and 14°C and outdoor air temperature to be -10°C. The vertical wall at the end of the duct is assumed to be adiabatic and the airflow rate to be $0.9 m^3/s$ with an angle of 45° to the horizontal direction.

Since the air velocity and the expected buoyancy forces in the duct are similar to those in indoor environments, the Zero-Equation turbulent model (Chen and Xu 1998) was used in the CFD simulation. A first grid size of 0.1m suggested by Zhai and Chen (2004) for indoor forced convection airflows was adopted. The simulated air velocity and temperature distributions at the middle lengthwise plane of the duct (z=0.75m) are plotted in Figure 6 and Figure 7, respectively. The results confirm the field observation, i.e. obvious temperature stratification and a reverse airflow on the top region of the duct (Heiselberg 2004).

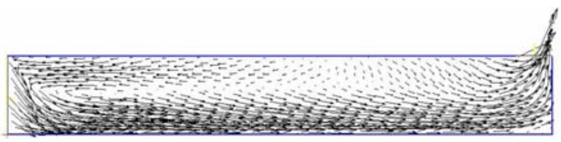

Figure 6 Air velocity distribution in the ETAHE duct at z=0.75m

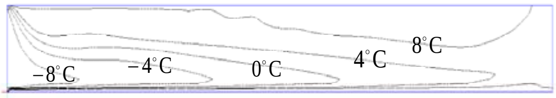

Figure 7 Temperature distribution in the ETAHE at z=0.75m

To further exam heat convection on the duct surfaces, the duct is assumed consisting of n equal length sections. A Local Averaged Convective Heat Transfer Coefficient (LACHTC) for the i'th section and the j'th surface is defined as:

$$h_{i,j} = \frac{1}{A_{i,j}\left(T_{wall,i,j} - \bar{T}_{air,i}\right)} \sum_{k=1}^{k=m} q''_{i,j,k} \, A_{i,j,k} \qquad (4)$$

This coefficient represents average heat convection intensity on individual duct surface of the n'th duct section. The purpose of defining LACHTC is to find a suitable CHTC as inputs for a proposed one-dimensional ETAHE model, which is presented in the next section. The CFD simulation results are used to obtain the LACHTCs for the simulated duct and they are presented in Figure 8.

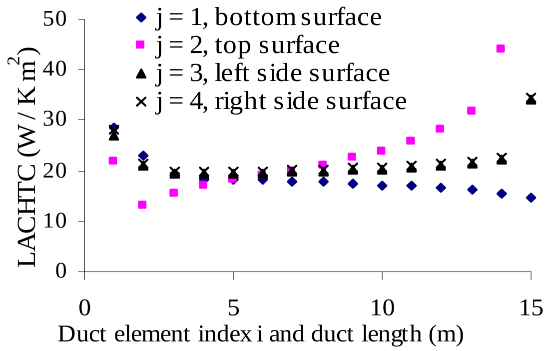

Figure 8 Local Average Convective Heat Transfer Coefficients

The defined LACHTC has similar physical meanings to the CHTC measured by Wachenfeldt (2003) shown in Figure 3. For the same system, the CFD simulation produced an overall average LACHTC of 21 W/m^2K. Corresponding to the same airflow rate of 0.9 m^3/s, the curve fitted correlation produced a CHTC of 25 W/m^2K. This comparison verified the CFD simulation again.

ONE-DIMENSIONAL MODEL OF HYBRID VENTILATION ETAHE

As illustrated in Figure 6 and Figure 7, airflow and heat transfer studied in the large size duct are three dimensional. CFD simulation may be the only method available if one seeks detailed air velocity and temperature distributions. However, if only average air temperatures at the duct outlet are of interest, airflow may be assumed to one dimensional. When the convective heat transfer coefficients and temperatures of each duct surface are known, the outlet air temperature can be calculated by solving heat balance equations for the duct as per the existing ETAHE models. In developing a one-dimensional ETAHE model, five assumptions were made as follows:

- Air inside the duct is incompressible with constant thermal properties.
- Airflow inside the duct is one-dimensional and steady state.
- The duct consists of 'n' elements with uniform air temperatures in each.
- The surface temperature of each duct wall is known and not affected by the airflow.
- No latent heat transfer occurs on the duct surfaces.

A heat balance equation for each element can be developed as follows

$$\dot{m}c_p\left(\overline{T}_{air,i}-\overline{T}_{air,i-1}\right)=\sum_{j=1}^{j=4}A_{i,j}h_{i,j}\left(T_{wall,i,j}-\frac{\overline{T}_{air,i}+\overline{T}_{air,i-1}}{2}\right) \quad (5)$$

When an outdoor air temperature is given, Equation 5 can be used to calculate the exit air temperature of the first duct element. Then, this air temperature is considered as an inlet air temperature for the next duct element. In this way, air temperatures along the duct running length can be calculated. To validate this simplified 1-D model, the ETAHE duct in Grong, Norway was simulated again. The results are plotted in Figure 9 together with the results from the CFD simulation.

According to the comparison shown in Figure 9, the two results in the second half of the duct are very close to each other. However, relatively large differences occur in the first half section. This may due to the reverse airflow phenomenon shown in Figure 6. In the first half of the duct, noticeable reverse airflow with higher temperature flows back into the first few duct elements. The actual energy balance in these elements should take into account the energy brought by the reverse airflow. Since the proposed 1-D model neglects this energy, the air temperature was underpredicted in the first half of the duct. As a simplified model describing the ETAHE performance in hybrid ventilation systems, the proposed model produced acceptable modeling results to some extent comparing with the CFD simulation.

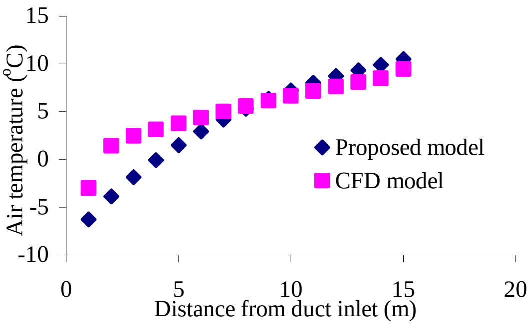

Figure 9 Comparison of proposed 1-D and CFD simulation results for air temperature

In practice, the parameters and the boundary conditions needed for the proposed model, i.e. the wall temperatures and the Local Average Convective Heat Transfer Coefficient, are usually unavailable. In this study, the LACHTC was derived from the CFD simulation. It is time consuming to conduct a CFD simulation every time before using the proposed model. However, if an empirical correlation could be produced from extensive CFD studies for typical ETAHE designs, the problem of lacking of LACHTC would be solved. In addition, if the heat transfer through the soil surrounding a large size ETAHE duct could be solved, the well prescribed wall temperatures would become available as well. These two aspects will be the future tasks

CONCLUSIONS

Literature surveys of ETAHE studies indicate the empirical heat convection correlations may produce greatly different heat transfer coefficients for identical design conditions. To accurately simulate ETAHE in mechanical ventilation systems appropriate empirical correlations have to be selected.

The CFD simulation of a hybrid ventilation ETAHE system produced similar results to field measurements. Modelling airflow and heat transfer processes in the large duct is complicated requiring CFD tools to investigate detailed air velocity and temperature distributions. Since airflow in hybrid ventilation ETAHE systems is far from fully developed, existing models are not sufficient to simulate the ETAHE performance.

A one-dimensional hybrid ventilation ETAHE model was developed to simulate outlet air temperature. The model produced similar results when compared to the CFD simulation. A general correlation or a method describing heat convection is needed to simulate hybrid ventilation ETAHE system using the proposed model.

NOMENCLATURE

A wall surface area, m^2

c_p specific heat of air, J/kg K

D circular cross sectional duct diameter, m

h convective heat transfer coefficient, W/m^2 K

$\bar{h}$ mean convective heat transfer coefficient of a duct from its entrance to a certain distance, W/m^2 K

$\bar{h}_\infty$ mean convective heat transfer coefficient of a infinitely long duct, W/m^2 K

i duct element index

j duct surface index, 1, 2, 3, and 4 representing bottom, top, left, and right surfaces, respectively

k surface grid index

K thermal conductivity of air, W/m K

L duct length, m

L_c distance constant from the duct inlet, m

m total number of discritized surface grids in an duct element in CFD domain

$\dot{m}$ mass flow rate of air, kg/s

Pr Prandtl number

q'' convective heat flux, W/m^2

Re Reynolds number

T_{air} air temperature, K

T_{wall} duct surface temperature, K

$\bar{T}_{air}$ average air temperature, K

v air velocity, m/s

ρ air density, kg/m^3

REFERENCES

Awbi, H.B. 1998. Calculation of Convective Heat Transfer Coefficients of Room Surfaces for Natural Convection. Energy and Buildings, 28(2), 219-227.

Bahadori, M.C., 1978. Passive Cooling Systems in Iranian Architecture, Scientific American, 238(2), 144-154.

Bansal, N.K., Sodha, M.S., and Bharadwaj, S.S. 1983. Performance of Earth Air Tunnels, International Journal of Energy Research, 7(4), 333-345.

Beausoleil-Morrison, I. 2002. The Adaptive Conflation of Computational Fluid Dynamics with Whole-Building Thermal Simulation, Energy and Buildings, 34(9), 857-871.

Benkert, S., Heidt, F.D., and Scholer, D. 1997. Calculation Tool for Earth Heat Exchangers GAEA, Proceeding of Building Simulation, Vol. 2, Fifth International IBPSA Conference, Prague, 9-16.

Boelter, L.M.K., Martinelli, R.C. and Jonassen, F. 1941. Trans. ASME, 63, 447-455.

Bojic, M., Trifunovic, N., Papadakis, G., and Kyritsis, S. 1997. Numerical Simulation, Technical and Economic Evaluation of Air-to-Earth Heat Exchanger Coupled to a Building, Energy, 22(12), 1151-1158.

Burmeister, L.C. 1993. Convective Heat Transfer, John Wiley and Sons, INC. Bath, United Kingdom.

Chen, Q., and Xu, W. 1998. A Zero-Equation Turbulence Model for Indoor Airflow Simulation, Energy and Buildings, 28(2), 137-144.

De Paepe, M. 2001. Earth-Air Heat Exchangers in the Belgian Climate: Analysis of the Potential with a 3d Modelling Technique. Market Opportunities for Advanced Ventilation Technology, 22nd annual AIVC conference, Bath, United Kingdom.

Goswami, D.Y., and Ileslamlou, S. 1990. Performance Analysis of a Closed-Loop Climate Control System Using Underground Air Tunnel, Journal of Solar Energy Engineering, Transactions of the ASME, 112(May), 76-81.

Heiselberg, P. (ed.) 2002. Principle of Hybrid Ventilation, Aarhus University Center, Denmark.

Heiselberg, P. 2004. Building Integrated Ventilation Systems - Modelling and Design Challenges, CIB 2004 World Building Congress, Toronto, Canada.

Hollmuller, P. 2003. Analytical Characterisation of Amplitude-Dampening and Phase-Shifting in Air/Soil Heat-Exchangers, International Journal of Heat and Mass Transfer, 46(22), 4303-4317.

Kumar, R., Ramesh, S., and Kaushik, S.C. 2003. Performance Evaluation and Energy Conservation Potential of Earth-Air-Tunnel System Coupled with Non-Air-Conditioned Building, Building and Environment, 38(6), 807-813.

Kusuda, T., and Achenkach, P.R. 1965. Earth Temperature and Thermal Diffusivity at Selected Stations in the United States, ASHRAE Transactions, 71.

Labs, K. 1989. Passive Cooling, The MIT press, Cambridge Massachusetts, London, England.

Li, Y., and Heiselberg, P. 2003. Analysis Methods for Natural and Hybrid Ventilation - a Critical Literature Review and Recent Developments. International Journal of Ventilation, Vol. 1 (Special Edition for Hybrid Ventilation), 3-20.

Mihalakakou, G., Lewis, J.O., and Santamouris, M. 1996. The Influence of Different Ground Covers on the Heating Potential of Earth-to-Air Heat Exchangers, Renewable Energy, 7(1), 33-46.

Mihalakakou, G., Santamouris, M., and Asimakopoulos, D. 1992. Modelling the Earth Temperature Using Multiyear Measurements, Energy and Buildings, 19(1), 1-9.

Mihalakakou, G., Santamouris, M., Lewis, J.O., and Asimakopoulos, D.N. 1997. On the Application of the Energy Balance Equation to Predict Ground Temperature Profiles, Solar Energy, 60(3-4), 181-190.

Sodha, M.S., Mahajan, U., and Sawhney, R.L. 1994. Thermal Performance of a Parallel Earth Air-Pipes System, International Journal of Energy Research, 18(4), 437-447.

Sodha, M.S., Sharma, A.K., Singh, S.P., Bansal, N.K., and Kumar, A. 1985. Evaluation of an Earth-Air Tunnel System for Cooling/Heating of a Hospital Complex, Building and Environment, 20(2), 115-122.

Singh, S.P. 1994. Optimization of Earth-Air Tunnel System for Space Cooling, Energy Conversion and Management, 35(8), 721-725.

Tzaferis, A., Liparakis, D., Santamouris, M., and 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, 18(1), 35-43.

Wachenfeldt, B.J. 2003. Natural Ventilation in Buildings Detailed Prediction of Energy Performance, PhD Thesis, Norwegian University of Science and Technology.

Zhai, Z., and Chen, Q. (2004). Numerical Determination and Treatment of Convective Heat Transfer Coefficient in the Coupled Building Energy and CFD Simulation. Building and Environment, 39(8), 1001-1009.