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Earth-air heat exchanger and its utilization in cold storages

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Abstract

Cold Storage is a special type of room, the temperature is kept very low with the help of machinery and precision equipment. Marine products are also produced in large quantities due to the large coastal areas. Therefore, it is a good job for entrepreneurs to invest. Uses Cold space is used to save fruit and vegetables. If kept in a cool place, they will not rot after many months. Sometimes, during the production of certain vegetable or fruit crops, the demand for those items decreases, which in turn reduces the consumption of the residual item and is stored in a cool place. Cold storage works primarily on the 'vapor pressure cycle' to produce cooling. Now, we want to introduce a better, more efficient, and less expensive method EAHE (Earth-Warm Temperature). The Earth-air heat exchanger (EAHE) is a promising method that can be used effectively to reduce the heat / cooling load of a building by first cooling the air in summer. Over the past two decades, much research has been done to develop analytical models and analysis numbers for EAHE systems. The method of calculating the fixed temperature of the earth (EUT) and the recently developed correlation of friction factor and Nusselt number are used to ensure maximum accuracy in calculating heat transfer. Advanced calculations allow designers to calculate heat transfer, coefficient of flexible heat transfer, pressure drop, and length of the EAHE system pipeline. A long pipe of small diameter buried at high depth and having a low flow rate of air causes an increase in the efficiency of the EAHE system.

Keywords: cold storages, exchanger, Earth-air, temperature

Introduction

Over the past two decades, much research has been done to develop analytical and numerical models to analyze EAHE systems (Mihalakakou *et al*. 1994; Bojic *et al*. 1997; Gauthier *et al*. 1997; Hollmuller and Lachal 2001; Su *et al*. 2012; Sehli *et al*. 2012; Ozgener *et al*. 2013) [9]. The performance analysis of EAHE involved the calculation of the continuous heat transfer from the pipe to the substrate or the calculation of the variable heat transfer from the circulating air to the pipe and the changes in air temperature and humidity. A number of computer modeling tools are commercially available. Energy Plus and TRSYS have well-functioning EAHE modules; however, these are analytical tools and are not used for immediate design.

Currently, Computational Fluid Dynamics (CFD) is best known to researchers for modeling and evaluating the performance of EAHE systems. CFD uses a very simple rule to divide the entire system into smaller grids. Then, governing standards were applied to these different factors to obtain numerical solutions for motion parameters, pressure distribution, and temperature gradients in the short term and at reasonable cost due to the reduction of the required test function (Kanaris *et al*. 2006; Wang *et al*. 2007). A thorough analysis of the EAHE system, the use of CFD is recommended, but limited to those with a good command over it. For the first design of the EAHE system, the use of basic heat transfer calculators is best suited to determine the geometric size of the system.

Many researchers such as De Paepe and Janssens (2003), Badescu and Isvoranu (2011) [2], and T'Joen *et al*. (2012) developed EAHE design equations and processes.

In this paper, the author has developed a one-sided model of the EAHE system. The EUT calculation method and the recently developed correlation of friction factor and Nusselt number are used to ensure maximum accuracy in calculating heat transfer.

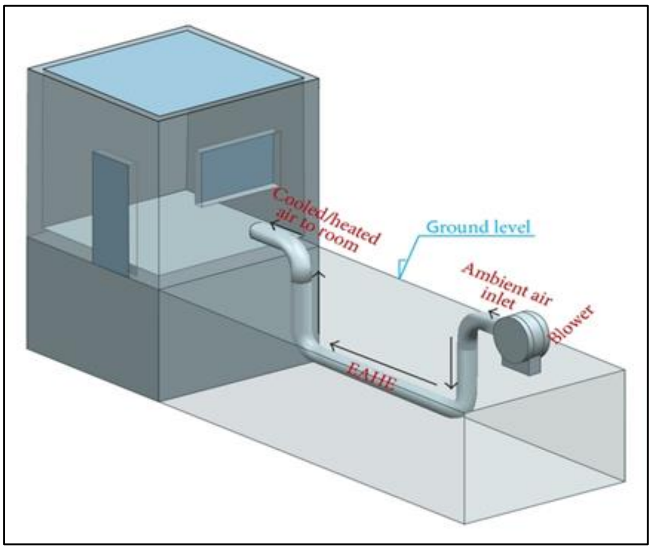

Fig 1

Review of technologies

The development of the EAHE system model involves the use of basic heat transfer calculators. The geometric measurements of the EAHE system are determined by considering the amount of heat or cooling that will be met to adjust the building surface. The design process involves identifying user input parameters as well as parameters that affect the output of the desired design. Once the output output has been adjusted, heat transfer gauges are used to meet the output output in terms of input parameters.

The geometric sizing parameters of an EAHE comprises the diameter of the pipe, *D*; length of the pipe, *L*; and number of pipes in parallel, N_p, in the heat exchanger.

Assumptions

The following assumptions are made to facilitate the development of a one-sided EAHE system model:

- The surface temperature is defined as the relative temperature of the local air, which is equal to the temperature of the air intake.
- EUT can be measured at the average annual temperature of the region (Bhopal, India).
- The polyvinyl chloride (PVC) pipe used in EAHE is a uniform cross-section.
- The size of the pipeline used in EAHE is very small; therefore, the heat resistance of the pipe material is not taken into account.
- The temperature on the surface of the pipe is the same as the axial direction.

Boundary conditions

The following boundary conditions were used in the one-dimensional model of the EAHE system.

Inlet boundary conditions

At the entrance to the EAHE pipe, the values of air flow velocity, v_a (m/s), and static temperature of air, T_{in} (°C), at inlet were to be defined. The thermodynamic properties (density and specific heat capacity) and transport properties (dynamic viscosity and thermal conductivity) of air were to be defined at static temperature of air at inlet.

Outlet boundary conditions

In a subsonic flow regime, the relative pressure at the outlet of the EAHE pipe was defined as equal to zero atm.

Wall

The temperature on the surface of pipe (wall) was uniform in axial direction and was defined as equal to earth's undisturbed temperature at Bhopal city (25.2 °C). No slip condition with smooth wall was assumed at the inner surface of the pipe.

Mass flow rate of air

The mass air flow rate is an important factor, and should be known by the designer in order to determine the choice of size and number of pipes. No unique size and pipeline size can meet EAHE performance. Therefore, the designer should consider the best combination of EAHE performance and pumping power required to ensure mass air flow. For a pipe of diameter, *D*; air density, *ρ*; air flow velocity, v_a; and number of parallel pipes, N_p, the mass flow rate of air through a pipe, m_a, is given by:

$$\dot{m} = \frac{\frac{\pi}{4} D^2 \rho v_{\mathrm{a}}}{N_{\mathrm{p}}} \qquad (1)$$

For the designer, these parameters have to be determined in such a way that the boundary conditions and the heat exchanger performance are met.

Earth's undisturbed temperature

The uninterrupted global temperature is an important parameter in designing an EAHE system. Assuming that the homogeneous soil of continuous thermal diffusivity, the temperature at any depth z and time t can be measured by the following expression (Labs 1989):

$$T_{z,t} = T_m - A_s exp\left[-z\left(\frac{\pi}{365\alpha_s}\right)^{\frac{1}{2}}\right] \cos\left\{\frac{2\pi}{365}\left[t - t_o - \frac{z}{2}\left(\frac{365}{\pi\alpha_s}\right)^{\frac{1}{2}}\right]\right\} \qquad (2)$$

where $T_{z,t}$ is representing the ground temperature at time *t* (s) and depth *z* (m), T_m is the average soil surface temperature (°C), A_s is the amplitude of soil surface variation (°C), α_s is the soil thermal diffusivity (m^2/s; m^2/day), *t* is the time elapsed from beginning of the calendar year (day), and t_o is the phase constant of soil surface (s; days). It is very difficult to accurately calculate the uninterrupted temperature of the earth because the boundaries of the soil are often unknown. In addition, the central soil structures are described. Therefore, the uninterrupted global warming is an estimated value that can be considered to be equal to the annual average of global warming. The surface temperature is considered to be equal to the prevailing air temperature. Thus, the uninterrupted global temperature of Bhopal (Central India) is defined as 25.2 ° C equal to the average annual temperature (source: Department of Meteorology, Bhopal).

Methods

If the of the EAHE system is known, the calculation of the heat transfer rate can be done using the log mean temperature difference method (LMTD) or the ε – number of transfer units (NTU) method. In this paper the ε – NTU method is used. Exhaust temperature is determined using the efficiency of EAHE (ε) which is the function of the number of transmission units (NTU).

Heat exchanger effectiveness and NTU

In the system of earth–air heat exchanger, the area used to transport air heaters only. Heat is released or pumped air into the pipe walls by moving and from the pipe walls to the surrounding soil and vice versa by the conveyor. If the contact of the pipe wall with the ground is considered complete and the groundwater flow is considered to be very high compared to the high resistance, then the temperature of the wall inside the pipe can be considered permanent. The NTU discourse relies on different types of EAHE system flow configurations. In this paper, an evaporator or condenser (with a constant temperature on one side, i.e., a wall) is used.

The total heat transferred to the air when flowing through a buried pipe is given by:

$$Q_{\mathrm{h}} = \dot{m}C_{\mathrm{p}}\left(T_{\mathrm{out}} - T_{\mathrm{in}}\right) \tag{3}$$

where $\dot{m}$ is the mass flow rate of air (kg/s), C_{p} is the specific heat of air (J/kg-K), T_{out} is the temperature of air at outlet of EAHE pipe (°C), and T_{in} is the temperature of air at inlet of EAHE pipe (°C).

Due to convection between the wall and the air, the transferred heat can also be given by:

$$Q_{\mathrm{h}} = hA\Delta T_{\mathrm{lm}} \tag{4}$$

where h is the convective heat transfer coefficient (W/m^2-K) and A is the internal surface area of the pipe (m^2).

The logarithmic average temperature difference (ΔT_{lm}) is given by ($T_{\mathrm{EUT}} = T_{\mathrm{wall}}$):

$$\Delta T_{\mathrm{lm}} = \frac{T_{\mathrm{in}} - T_{\mathrm{out}}}{\ln\left[\frac{(T_{\mathrm{in}} - T_{\mathrm{wall}})}{(T_{\mathrm{out}} - T_{\mathrm{wall}})}\right]} \tag{5}$$

The temperature of air at the outlet of the EAHE pipe can be obtained in an exponential form as a function of the wall temperature and inlet air temperature by eliminating Q_{h} from Eqs. (3) and (4).

$$T_{\mathrm{out}} = T_{\mathrm{wall}} + \left(T_{\mathrm{in}} - T_{\mathrm{wall}}\right)e^{-\left(hA/\dot{m}C_{\mathrm{p}}\right)} \tag{6}$$

If a pipe of infinite length ($A = \infty$) is used, the air will be heated or cooled to the wall temperature. The effectiveness (ε) of EAHE for winter heating application can thus be defined as:

$$\varepsilon = \frac{T_{\mathrm{out}} - T_{\mathrm{in}}}{T_{\mathrm{wall}} - T_{\mathrm{in}}} = 1 - e^{-\left(hA/\dot{m}C_{\mathrm{p}}\right)} \tag{7}$$

The non-dimensional group is called the number of transfer units (NTU):

$$\mathrm{NTU} = \frac{hA}{\dot{m}C_{\mathrm{p}}} \tag{8}$$

Which gives

$$\varepsilon = 1 - e^{-\mathrm{NTU}} \tag{9}$$

The operation of the earth-air heating system is determined by the flawless NTU group. The variance in the efficiency of the earth-air temperature switch as the function of the number of transmission units is shown in Figure 2. It was noted that with the increase in NTU value, efficiency also increases but the curve decreases rapidly. The associated profitability in its operation is very small after the NTU value exceeds 3. There are several ways to create a global air temperature changer to get a given NTU as well as a desirable performance. Similar results have been observed by De Paepe and Janssens (2003).

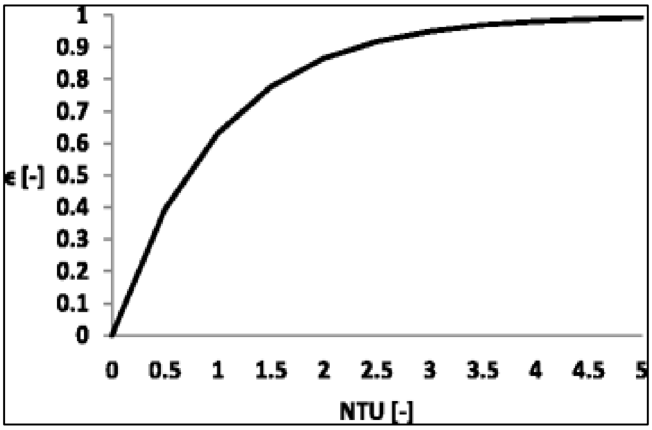

Fig 2: Earth–air heat exchanger effectiveness as a function of number of transfer units

The impact of design parameters on NTU can be studied by heat transfer and pressure reduction. The NTU consists of three parameters, namely, the convective heat transfer coefficient (h), the internal area of the pipe (A) and the air flow rate (ṁ) which can vary.

The inner surface of the pipe is a function of the width, D, and length of the EAHE pipe, L, both:

$$A = \pi DL \tag{10}$$

The convective heat transfer coefficient inside the pipe is defined by:

$$h = \frac{N_{\mathrm{u}}K}{D} \tag{11}$$

where K is the thermal conductivity (W/m-K).

Zhang (2009) presented in his PhD thesis that in conventional earth-to-air heat exchanger (ETAHE) systems, it is typical to have buried ducts with 10 cm < D_h < 40 cm and lengths longer than 20 m. Such sizes mean the ratios of the lengths to the hydraulic diameters (D_h) are at an order of magnitude of 100. The hydraulic diameter is defined as four times the ratio of the cross-section area to wetted perimeter of the cross-section.

$$D_\mathrm{h} = \frac{4A}{P} \tag{12}$$

where A is the cross-section area and P is the wetted perimeter of the cross-section
The hydraulic diameter of a circular tube is simply the diameter of the tube. Therefore, it is reasonable to assume that air flow is fully optimized for EAHE of such sizes and to adapt to the corresponding empirical relationship to calculate the convection heat transfer coefficient (CHTC). To test this assumption, eight Nusselt (N u) numbers are related to other ETAHE simulation studies (Arzano and Goswami 1997; Bojic *et al*. 1997, cooling and heating; Singh 1994; De Paepe and Janssens 2003; Hollmuller 2003; Sodha; *et al*. 1994; Benkert and Heidt 1997) [1, 9, 3] were used. Since all correlations were found to fully improve the flow of turbulent air, appropriately, it is expected to produce the same values in the same operating condition. Nusselt's numerical variance with respect to Reynold's standard ETAHE design was drawn using all eight links to calculate CHTC, and the largest difference was observed among the eight correlation outcomes. This may be due to a variety of test conditions, which are adopted to determine the correlation, for example, the overlap of the test channels. Significant differences indicate that the appropriate correlation should be chosen when one uses any available models to simulate the performance of the EAHE system.
The EAHE system described in this paper contains cylindrical pipes of 0.1016 m in diameter made of PVC with a total burial length of 19.228 m. Assuming that the internal surface area of the PVC pipes used in the EAHE system is smooth, the N u links provided by De and Janssens (2003) can be used to measure system performance.

$$N_\mathrm{u} = \frac{f/8\,(R_\mathrm{e} - 1000)\,P_\mathrm{r}}{1 + 12.7\sqrt{(f/8)}\left(P_\mathrm{r}^{2/3} - 1\right)} \tag{13}$$

(For turbulent flow in tubes with smooth internal surface)
where R_e is the Reynolds number, P_r is Prandtl number, and f is the friction factor for smooth pipes

With $f = (1.82 \log R_\mathrm{e} - 1.64)^{-2}$ (14)

If $2300 \le R_e < 5 \times 10^6$ and $0.5 < P_r < 10^6$
The Reynolds number is related to the average air velocity and diameter:

$$R_\mathrm{e} = \frac{\rho v_\mathrm{a} D}{\mu} \tag{15}$$

where v_a is the velocity of air through pipe (m/s), D is the diameter of the pipe (m), and μ is the dynamic viscosity of air (kg/m-s).

The Prandtl number is given by

$$P_r = \frac{\mu c_\mathrm{p}}{K} \tag{16}$$

where c_p is the specific heat of air (J/kg-K)

Results and discussion

The thermo-physical properties of materials used in design calculations of EAHE needed to be installed in the cold storages are shown below:

Table 1 Thermo-physical properties of materials used in design calculations of EAHE

The value of Reynolds number was calculated for air flow velocities of 2, 3.5, and 5 m/s and thermo-physical properties of air at 16.7 °C for winter heating application. The corresponding values of friction factor were calculated by using Eq. (14). The value of Prandtl number for thermo-physical properties of air at 16.7 °C was calculated as 0.717. After evaluation of Reynolds number, friction factor, and Prandtl number, the Nusselt number was calculated using Eq. (13) corresponding to air flow velocities of 2, 3.5, and 5 m/s. The variation of Nusselt number with respect to Reynolds number is shown in Fig. 3. It was observed that the value of Nusselt number increases with increase in Reynolds number. Many other researchers (Nakamura and Tamotsu 2004; Luciu *et al*. 2009) also found that the value of Nusselt number increases with increase in Reynolds number.

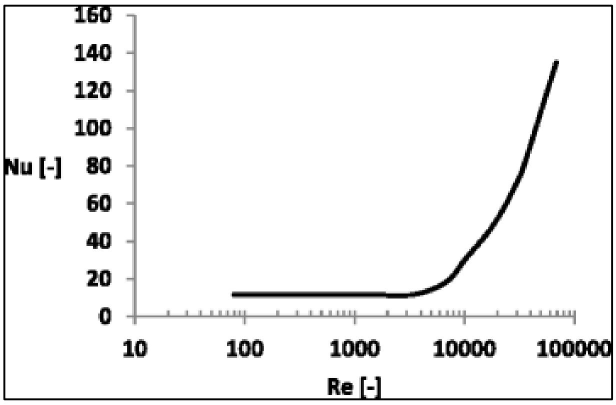

Fig 3: Variation of Nusselt number with respect to Reynolds number

The CHTC for different air flow velocities was calculated by using Eq. (11). The CHTC increases with increase in air flow velocity as shown in Fig. 4. Xiao *et al*. (2011) found that the variation in value of CHTC with a change in air flow velocity has shown similar trends.

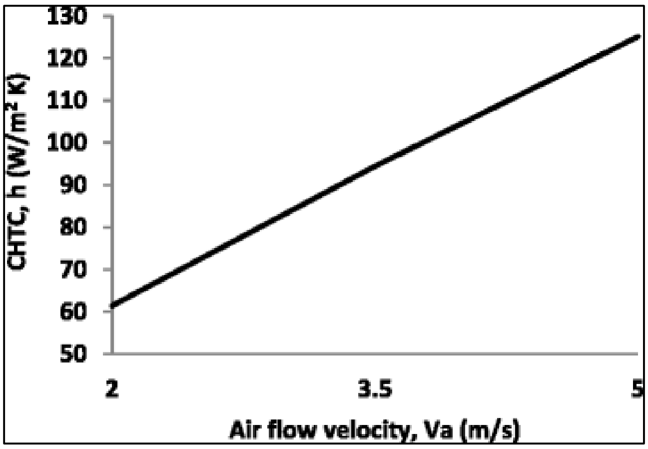

Fig 4: Convective heat transfer coefficient vs. air flow velocity

The length, L, is an independent parameter influencing the NTU. There is a linear variation of NTU with length. Changing either the diameter, D, or mass flow rate, $\dot{m}$, changes the air velocity inside the tube. This result in a changing Reynolds number, R_e. D and $\dot{m}$ have thus no independent influence on the NTU. 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.

Influence on pressure drop

The pressure drop in a smooth pipe is given by

$$\Delta p = f\frac{L}{D}\rho\frac{{v_a}^2}{2} \tag{17}$$

$$f = \frac{64}{R_e} \text{ If } R_e < 2300$$

$$f = (1.82 \log R_e - 1.64)^{-2} \; If \; R_e \geq 2300$$

It is noted from Eqs. (8) and (17) that both NTU and Δp are proportional to the length of the pipe, and the designer can use NTU/L and $\Delta p/L$ as the main performance measures to determine the required length of pipe for design purposes. The length of pipe, L, is an independent parameter which has a linear influence on pressure drop. The diameter of the pipe and air flow velocity has a combined effect on pressure drop. The decrease in air flow velocity and increase in diameter of pipe results in decrease in pressure drop. This is in disagreement with the thermal demand of a small diameter. In each case, a large number of pipes are beneficial. The combination of pipe length and diameter has to be optimized. To evaluate the overall thermo-hydraulic performance of a specific configuration, the J-factor, introduced by De Paepe and Janssens (2003) was used. It is the ratio of the pressure drop to the NTU value and was shown to be a good performance metric of an earth–air heat exchanger.

$$J = \frac{\Delta P}{\text{NTU}} \tag{18}$$

Conclusions

The earth–air heat exchanger is a promising technique which can effectively be used in the cold storages to provide the cooled air in summer. Many researchers have developed EAHE design equations and procedures. For a complete analysis of the EAHE system, the use of CFD is recommended but it is limited to those who have a good command over it. For the initial design of an EAHE system, the use of basic heat transfer equations is more suitable to determine the geometrical dimensions of the system. In this paper, the author has developed a one-dimensional model of the EAHE system. The method to calculate the EUT and more recently developed correlations for friction factor and Nusselt number are used to ensure higher accuracy in the calculation of heat transfer. The value of EUT for Bhopal (Central India) was calculated as 25.2 °C. It was observed that Nusselt number increases with increase in Reynolds number.

The design of earth–air heat exchanger mainly depends on the heating/cooling load requirement of a building to be conditioned. After calculation of heating/cooling load, the design of the earth–air heat exchanger only depends on the geometrical constraints and cost analysis. The diameter of pipe, pipe length, and number of pipes are the main parameters to be determined. With an increase in length of pipe, both pressure drop and thermal performance increase. A longer pipe of smaller diameter buried at a greater depth and having lower air flow velocity results in an increase in performance of the EAHE system.

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