Simulation of a Predominantly Passive Natural Air Cooling System

P.v.Z. Venter and M. van Eldik

Abstract—Climate control is an everyday challenge. With the rapid surge in electricity prices over the past few years, air conditioning operating expenses necessarily increased. The effects, furthermore, of global warming result in increased cooling, and therefore, energy demand.

The purpose of this paper is to propose two models that simulate a natural air cooling system. The first model simulates cooling through an earth to air heat exchanger, utilising the soil as a heat sink. The second model simulates the transient cooling of a control volume, which receives cooled air and is open to the environment.

A scale model of an earth-to-air heat exchanger system was designed, constructed and used to verify results from the proposed models.

Following verification, a real-life size heat exchanger was simulated in order to cool down a room of 60 m^3 within one hour, using only the underground soil as a heat sink.

Results showed that a room at an initial 30 °C can be cooled down to 20.5 °C with a 1.2 m underground heat exchanger and down to 17.8 °C if the length is increased to 2.0 m. Only fan power is needed to increase the air's dynamic pressure, resulting in flow conditions. As a result a coefficient of performance between 60 and 80 can be achieved.

***Index Terms*—Air cooling, control volume, earth to air heat exchanger, simulation models, transient cooling.**

1 INTRODUCTION

In modern days, climate control has become an everyday challenge. Due to global warming, indoor temperatures of residential homes can rise to uncomfortable levels, especially in summer months. By making use of low exergy systems for heating and cooling of residential homes thermal comfort can be improved and energy efficiency can be increased [1].

Through air conditioning, indoor temperatures can be regulated to comfortable levels, despite ambient conditions. However, taking electricity prices and environmental factors, i.e. the supply of energy through coal based electricity plants in consideration; the use of air conditioning should be limited. Above one third of the world's primary energy demand is due heating and cooling appliances, which include air conditioning [2].

Power generation in South Africa is predominantly dependant on fossil fuels. Fossil fuelled power generation is not only limited in available resources, but possess a negative effect on the environment that include the assistance of global warming. In order to possibly ensure a safe future energy demand it is imperative to investigate and implement the use renewable energy.

It is, therefore, of utmost importance to investigate alternative cooling methods that can be implemented, either assisting or in some instances the replacement of traditional air conditioning systems. Two typical types of cooling mechanisms can be considered for residential air cooling; passive - and active systems.

A passive system is defined as a building envelope that uses environmental potentials such as wind or solar energy. Active systems involve various mechanical and electric components, such as fans and heat pumps. The proposed solution for this paper is a passive, supported by an active system.

This paper proposes a solution where ambient air is cooled through an earth to air heat exchanger. Air is forced, by means of a fan, into a piping system which is buried underground. This air is then cooled through a heat exchanger, where the outlet is connected to a room. Allowing cooler 'ambient' air to enter a room will result in less electricity needed by an air conditioning system to obtain the required temperature.

This paper, furthermore, proposes two simulation models. The first model is to simulate the air cooling through the heat exchanger, for design purposes. The second model simulates the transient cooling of a room, open to the environment, receiving an air flow. By utilising this second model, a room's cool-down rate and duration can be simulated. These simulation results, in conjunction with the first proposed model, can then be used to size the heat exchanging system.

In the following section a literature survey is provided, followed by the proposed simulation models. A result section is given in Section 4 where experimental data from a test bench is used to verify the proposed models. A case study follows in Section 5.

2 LITERATURE SURVEY

This section focuses on available literature for four different natural air cooling methods.

1) Night flushing, or night-purge ventilation, is a passive cooling concept where no mechanical or electrical components are used in the cooling system. With a night flushing system, cool ambient air is circulated through a building during evenings.

Natural ventilation such as a night flushing system is most effective in areas where the day time temperatures are relatively high, compared to that of the evenings. A building's features as well as solar and internal heat gain plays an important role in natural ventilation. Furthermore, a

P.v.Z. Venter, North-West University PO Box X6001, Potchefstroom, 2520, South Africa (12330825@nwu.ac.za).

M. van Eldik, North-West University PO Box X6001, Potchefstroom, 2520, South Africa (martin.vaneldik@nwu.ac.za).

building must typically be designed for night flushing to increase the potential effect thereof.

Night flushing can easily be implemented; however, it is dependent on the opening and closing of windows at specific times. Also note that for security or noise pollution reasons it might not be ideal for open windows at the premises or residential home [3].

2) A solar chimney is a passive solar system that transfers solar energy without the usage of mechanical or electrical devises. The concept of a solar chimney is to limit undesired heat gain during the day through natural ventilation. Warm interior air is replaced with cooler exterior air when possible. Furthermore, cold night time air is stored and used to cool down warm air during the day [3].

Results reported by [3] showed that at times up to a 50% saving in electricity usage was obtained by means of a solar chimney.

3) Depending on the desired work temperature, geothermal heat can be used for varies purposes. For space heating and cooling geothermal energy can be used directly due to the low and medium-enthalpy sources. It should be noted that a typical soil temperature for a sufficient depth below the surface is in the region of 13˚C [4].

Geothermal heat exchangers can be used for heating or cooling ambient air. These systems make use of tubes buried underground. The ground is used indirectly with a solution of water and antifreeze that is pumped and circulated through the loop of tubes in a closed system [5]. The tubes in the ground form a large heat exchanger with the soil. During the winter the warmer earth heats the water-antifreeze solution that circulates in the loop. These systems are typically connected to a unit located in a home or building.

To warm-up ambient air; the unit transfers the heat from the water in the loop, compresses the heat to a higher temperature then uses that heat to warm the air. For cooling during a typical summer's day, the system operates in reverse. Heat is absorbed from the air in the house and transferred to the geothermal unit. The heat in the tubes is then absorbed by the ground.

A geothermal system is expensive to install with a typical payback period within six to ten years according to [5].

4) The final air cooling method is the incorporation of earth to air heat exchangers. Note that this is a specific case of geothermal heating or cooling. Ambient air is forced through tubes buried underground. The soil serves as an energy buffer by reducing the temperatures during the summer when the ground temperature is lower than the outside ambient conditions. During winter the soil serves as an energy source by heating the air temperature inside the tubes when the ground temperature is higher than that of ambient conditions [6]. As stated earlier, the underground soil temperature maintains a nearly constant temperature of about 13˚C throughout the year.

The earth to air cooling system's performance is mostly affected by the following parameters; pipe span, - length, - depth, - material and air flow rate [7]. Research shows that by increasing the tube length or depth at which it is buried and by decreasing its pipe diameter or mass flow rate, the outlet temperature of the air will decrease [8].

Furthermore, another advantage of an earth to air heat exchanger is low operational and maintenance costs [9, 10].

3 SIMULATION MODELS

Two simulation models are formulated in this section. The first model simulates an earth to air heat exchanger, where the outer tube surface temperature is kept at a constant. The second proposed model, numerically determines the transient cooling of an enclosed volume where colder air enters and mixes with that inside the control volume. It should be noted, as cooler air enters, a mixture of air simultaneously exits the control volume.

All theoretical equations can be found in [11].

The following model simulates the temperature distribution of an earth to air heat exchanger:

An evenly distributed mass flow, $\dot{m}$ (kg/s), enters a header, connected to n tubes. Mass flow is calculated by:

$$\dot{m} = \rho V A \tag{1}$$

where ρ is the density (m^3/kg), V the velocity (m/s) and A the cross sectional area (m^2), perpendicular to the flow.

The cooling experienced by the air between two points along any segment along the heat exchanger is calculated by:

$$\dot{Q} = \dot{m}\Delta h \tag{2}$$

where $\dot{Q}$ (W) is the heat rejection and Δh the change in enthalpy (J/kg), which is dependent on temperature and pressure. The initial inlet enthalpy is known; therefore the heat rejection and outlet conditions are unknown. A second heat rejection calculation follows from:

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

The overall heat transfer coefficient UA (W/K) is given by:

$$\frac{1}{UA} = \frac{\ln(D_o/D_i)}{2\pi kL} + \frac{1}{h_c \pi D_i L} \tag{4}.$$

The heat exchanger's outer and inner tube diameters are denoted by D_o (m) and D_i (m), respectively. The tube's conductivity is given by k (W/mK), with a length L (m). The log mean temperature difference is represented by ΔT_{lm}:

$$\Delta T_{lm} = \frac{\Delta T_i - \Delta T_{i+1}}{\ln(\Delta T_i - \Delta T_{i+1})} \tag{5}$$

where

$$\Delta T_i = T_i - T_{si} \tag{6}$$

and T_{si} the inner tube surface area at point i.

The convection heat transfer coefficient h_c (W/m^{2}K) is determined by:

$$h_c = \frac{Nu \cdot k}{D_i} \tag{7}$$

where the dimensionless Nusselt number is determined from the Dittus Boelter correlation for a fluid in cooling:

$$Nu = 0.023 Re^{0.8} Pr^{0.3} \tag{8}.$$

The dimensionless Prandtl number is defined by:

$$Pr = \frac{C_p \mu}{k} \tag{9}$$

whereas the dimensionless Reynolds number is defined by:

$$Re = \frac{\rho V D_i}{\mu} \tag{10}$$

where μ is the fluid's viscosity (kg/ms).

By utilising equations (1) to (10) and discretising the heat exchanger tubes into consecutive segments, the steady state temperature profile of the air can be simulated.

A second model is needed to simulate the transient temperature change in a control volume. As air enters and exits a control volume, mixing occurs. It is assumed that uniform mixing takes place during this transient process. The following differential equation needs to be set up and solved to determine the temperature distribution in the control volume over time:

$$\frac{dT}{dt} = f\left(\frac{dM_t}{dt}\right) \qquad (11)$$

where M_t represents the total mass (kg) within the control volume at time t, which is open to the environment. The following numerical model is proposed that solves this transient process, through a combination of an implicit as well as an explicit iterative approach.

The mass equation (not conservation) at time t is defined by (12):

$$\dot{m}h_t\Delta t + M_{t-1}h_{M_{t-1}} = \left(\dot{m}\Delta t + \frac{1}{2}(M_{t-1} + M_t)\right)h_{M_t}$$

where $T_t = f(h_t)$ and

$$P\bar{V} = M_tRT_t \qquad (12).$$

The gas constant for air is given by R (J/kgK) and $\bar{V}$ is the control volume's volume (m^3).

4 Practical Setup with Experimental and Simulation Results

4.1 Test bench sizing

A test bench was designed, implementing both models as proposed in the previous section. The rationale behind the test bench is for proof of concept. This testing facility, therefore, does not portray an actual size heat exchanger or room. A schematic representation is given in Fig. 1. Results from the simulation models will be verified against the test bench, in order to validate the concept.

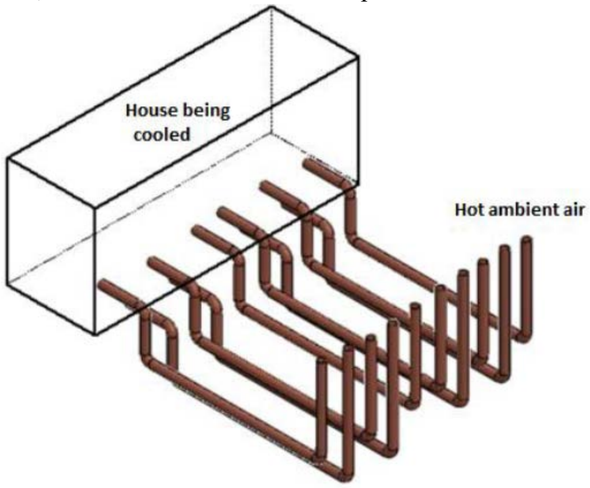

Fig. 1: A schematic representation of the scale model. Ten copper tubes submerged in water are used to cool down a control volume of $0.5m^3$, which is open to atmosphere.

The test bench consists of 10 copper tubes with an inner diameter of 13 mm. Air is forced through a fan into a header (not shown in the figure), which flows into the copper tubes and exits into a control volume. The horizontal sections of the tubes are placed in a water reservoir, simulating the soil temperature at 13.0 °C.

Air velocity entering the header is measured and used to calculate the mass flow rate. Temperatures are measured inside the header, at the exit of some copper tubes and inside the control volume. The test bench heat exchanger is sized so that air entering at 30 °C will be cooled to approximately 21 °C for a velocity ranging from 3.0 m/s to 3.5 m/s. Simulation results show that with a heat exchange length of 0.4 m, an exit air temperature between 20.89 °C and 21.08 °C can be obtained for a velocity from 3.0 m/s to 3.5 m/s.

For the control volume sizing it was decided that the inside air temperature should be cooled down within 5 minutes from an initial 30.0 °C. This will not portray a typical house or office scenario and will only be used to demonstrate the simulation of cool-down times.

Fig. 2 depicts simulation result for the cooling of a 0.5 m^3 volume, when the initial air temperature is at 30 °C and 10 copper tubes are used to deliver cooled air at 3.3 m/s. Take note that the horizontal line at 20.89 °C indicates the simulated steady state outlet temperature of the heat exchanger.

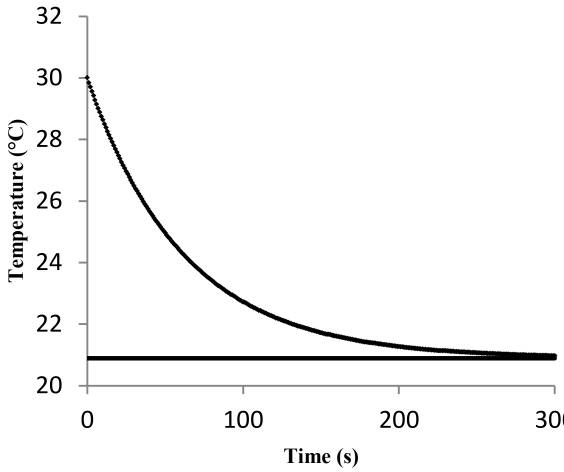

Fig 2. Simulation results of a cool-down inside a control volume over five minutes.

Results from the transient (second) simulation model indicate that the expected temperature after five minutes is 20.97 °C. Take note that just over four and a half minutes (276 s), the control volume is cooled down to 21.00 °C. A 0.5 m^3 volume was therefore chosen as the bench 'room', with dimensions of 0.5 m x 0.5 m x 1.0 m.

4.2 Experimental testing and validation of heat exchanger simulation

A number of tests were performed on the steady state heat transfer through the test bench's heat exchanger and, separately, transient cooling of the control volume. Differences between experimental and simulated values showed to be in the same order of magnitude.

To improve numerical accuracy, the heat exchanger simulation model was discretised into 10 equal length intervals. The test bench, however, only measured the inlet and outlet air temperatures. The inlet and outlet temperatures for a number of test scenarios are provided in Table I, together with the air velocity. A percentage error, with respect to the change in temperature is furthermore provided. Note that the inlet air temperature and velocity, for each scenario, are equivalent for the simulation and experimental results.

Table I. Simulation and experimental results for five test conditions.

Test	Inlet conditions		Outlet Temperature (°C)		% error
	V (m/s)	T (°C)	Simulation	Experimental	
1	3.2	26.2	19.26	19.7	6.3
2	3.2	28.9	20.48	20.9	5.0
3	3.2	29.4	20.70	21.4	8.0
4	3.3	30.1	21.05	21.6	6.1
5	3.5	32.7	22.30	22.9	5.8

All results from Table I lie within a 5.0 % to 8.0 % error range and is deemed as sufficiently accurate. It should be noted that air velocity is determined from the average between measurements at different placements. Any inaccuracy in these measurements may influence the error percentages. Furthermore, the change in temperatures between the inlet and outlet is relatively small, which influences the error percentages.

It should be further noted that all simulation results show that more heat transfer is expected. Due to the low velocity, the Reynolds numbers are slightly above that of 2300, indicating that turbulent flow is just evident. A slightly lower velocity than what was measured might therefore result in laminar air flow with a constant Nusselt number of 4.36.

4.3 Experimental testing and validation of transient cooling model

Experiments were performed on the cooling of the control volume. The control volume was heated up while steady state conditions were reached by the heat exchanger. During this time a natural flow of air developed within the control volume. At a temperature of 30.3 °C the heat source was removed, so that only cooled air could enter the well-insulated control volume. Temperature readings within the control volume were logged every second.

Figure 3 shows the transient simulated air temperatures of the control volume, together with the experimentally obtained values. The heat exchanger simulation indicates that cooled air at 21.14 °C enters the control volume. Results from the transient simulation indicates that just after 150 s (163 s), the air mixture inside the control volume should be at 22.00 °C, provided a continuous uniform mixture of molecules.

From Fig. 3 it is evident that the theoretical results initially under predict the control volume's temperature and that a larger change in temperature is observed. It should be noted that the colder air, with a higher density, enters the control volume at the bottom, whereas air is discarded into the surrounding environment at the top. The warmer, lower density, air is therefore initially forced out of the control volume through the natural convection, without uniformly mixing with the incoming stream. As a result, the transient model initially under-predicts the heat rejection.

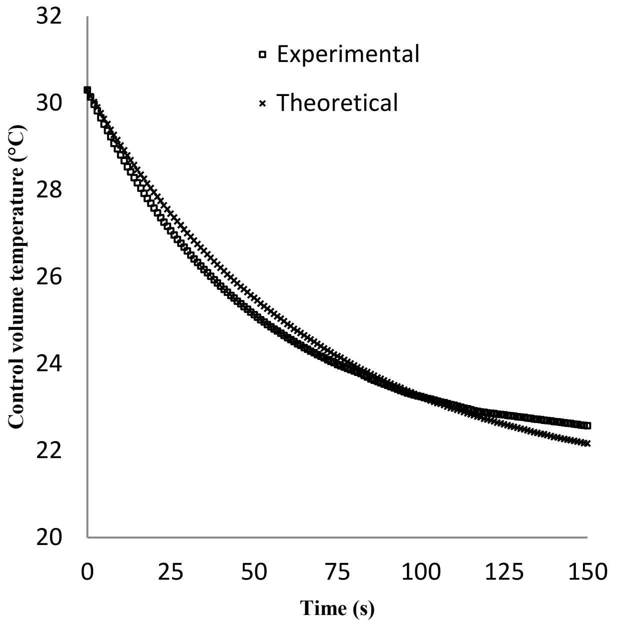

Fig. 3: Simulated and experimental temperatures for cooling of the control volume over 150 s.

As the control volume's temperature decreases, the experimental temperature moves towards the simulated temperatures. At some point the experimental temperatures crosses over with the simulated results, so that a lower temperature is predicted. It should be noted that at 150 s, the control volume's temperature is 1.3 °C from the simulated inlet temperature and practically very close towards the obtainable limit. The simulated temperature is 0.9 °C from the inlet temperature at 150 s.

These results verify the use of the numerical transient model and demonstrate how it can be incorporated to simulate the natural cooling of a control volume.

5 CASE STUDY

In the previous section the validity of the two proposed models were demonstrated. A case study follows on a full scale volume of a typical large room. A heat exchanger sizing follows on the design parameters that a 60 m^3 is required to be cooled down from 30 °C to an adequate temperature. A typical satisfactory room temperature is 22 °C. The design must allow cooling within an hour with a design 'buffer' of 1.5 °C for air flowing at 3.3 m/s. Therefore, air at 30 °C should be cooled to 20.5 °C within the allowable time.

By choosing a copper tube with an inner diameter of 0.0254 m, and utilising the heat exchanger simulation model, a sizing is determined. If 10 tubes are used with a heat exchange length of 1.2 m, assuming a surrounding soil temperature of 13.0 °C, the 60 m^3 volume will be cooled from 30.0 °C to 20.5 °C within an hour. Cooled air at 19.0 °C will enter the room.

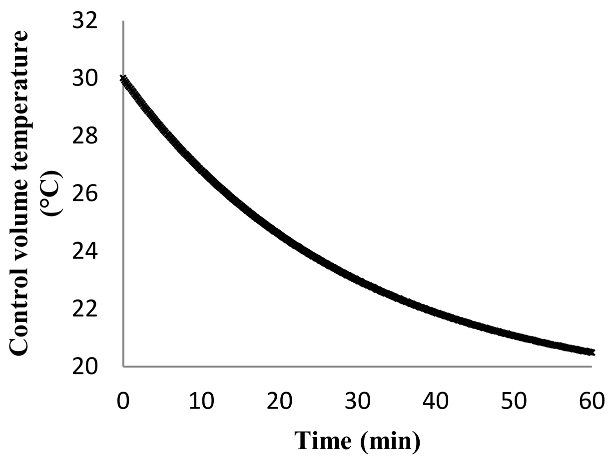

Fig. 4: Simulation results for the cooling of a 60 m^3 room over one hour. .

Figure 4 depicts a realistic cool-down graph for the 60 m^3 room, when ten 1.2 m copper tubes of a 0.0254 m diameter exchanges heat underground. The time-change interval, Δt, from (12) is fixed at 10 second intervals, resulting in 360 numerical time-steps to be solved. As can be seen from Fig. 4, apart from the x-axis interval, it is conceptually the same as the simulation results observed in Fig. 3.

If a lower temperature is to be designed for, based on an ambient temperature of 30 °C, the length of the heat exchanger tubes need to be increased. Table II depicts the outlet air temperature from heat exchangers with a 1.6 m and a 2.0 m design. Accompanied by these temperatures are the simulated room temperatures after one hour. The simulations are performed on a 60 m^3 room with an air velocity of 3.3 m/s.

Table II. Simulated outlet heat exchanger -, including a 60m^3 room temperatures after one hour.

Length of a single tube	Outlet temperature of heat exchanger	Room temperature after one hour
1.2 m	19.0 °C	20.5 °C
1.6 m	17.3 °C	19.0 °C
2.0 m	16.0 °C	17.8 °C

As mentioned earlier, a fan is used to force the air through the heat exchanger and into a room. In order to measure the efficiency of such an earth to air heat exchanger system, the power consumed by the fan needs to be accounted for. The dynamic pressure added towards the air, including all the pressure drops needs to be calculated in order to determine the power supply needed for a fan.

For all three scenarios, as depicted in Table II, the required fan power is calculated and tabulated in Table III. The dynamic pressure added is calculated at 5.5 Pa for an air velocity increase of 3.3 m/s.

Table III. Simulated outlet heat exchanger -, including a 60m^3 room temperatures after one hour.

Length of a single tube	Total pressure added (Pa)	Fan work required (W)	Cooling load (W)	COP
1.2 m	96.6	2.3	186.1	80.6
1.6 m	127.4	3.0	216.1	71.0
2.0 m	158.2	3.8	237.3	62.8

In Table III total pressure head that needs to be added for the three scenarios are given, followed by the work required by the fan. Take note that a fan operating efficiency of 70% is chosen throughout.

The final results given are the cooling load for the 60 m^3 room, together with the system's coefficient of performance (COP). The extremely high COP's are due to the fact that the underground soil is used as a zero-cost heat sink. Note that this zero-cost does not refer to the cooling system that needs to be manufactured and installed. It, therefore, only refers to the cooler underground heat sink at a constant 13 °C.

The authors do acknowledge that these results are based on simulation outcomes, without taking any costs or instalment into account. The purpose, however, of this paper, is to present a feasible alternative approach towards cooling, rather than traditional air conditioning. Through the proposed simulation models the applicability of an earth to air heat exchanger system was demonstrated. Such systems could potentially be used at a fraction of the electrical input required by standard air conditioning units.

REFERENCES

[1] G. C.da Graca and P. Linden. (2016.). Ten questions about natural ventilation of non-domestic buildings. *Buidling and Enviroment, (107),* pp. 263–273.

[2] K. S. Cetin and C. Kallus. (2016). Data-Driven Methodology for Energy and Peak Load Reduction of Residential HVAC Systems. *Procedia, (145),* pp. 852–859.

[3] M. Macias, A. Mateo, M. Schuler and E. M. Mitre. (2006). Application of night cooling concept to social housing design in dry hot climate. *Energy and Buildings*, *38*, pp. 1104-1110.

[4] M. Macias, J. A. Gaona, J. M. Luxan and G. Gomez. (2009). Low cost passive cooling system for social housing in dry hot climate. *Energy and Buildings*, *41*, pp. 915-921.

[5] A. Angrisani, G. Diglio, M. Sasson, F. Calise and D. M. d'Accadia. (2015). Design of a novel geothermal heating and cooling system. *Energy Conversion and Management*, *(108),* pp. 144-159.

[6] G. Florides, S. Kalogirou. (2007). Ground heat exchangers—A review of systems, models and applications. Renewable Energy, *(32)*, pp. 2461 - 2478.

[7] M. Khabbaz, B. Benhamou, K. Limam, P. Hollmuller, H. Hamdi and A. Bennouna. (2016). Experimental and numerical study of an earth-to-air heat exchanger for air cooling in a residential building in hot semi-arid climate. *Energy and Buildings*, *125*, pp. 109-121.

[8] S. F. Ahmed, M. T. O. Amanullah, M. M. K. Khan, M. G. Rasul and N. M. S. Hassan. (2016.). Parametric study on thermal performance of horizontal earth pipe cooling system in summer. *Energy Conversion and Management, (114),* pp. 324–337.

[9] J. Pfafferott. (2003). Evaluation of earth-to-air heat exchangers with a standardised. *Energy and Buildings*, *3(35)*, pp. 971-983.

[10] A.K. Misra, M. Gupta, M. Lather and H. Garg. (2015). Design and performance evaluation of low cost Earth to air heat exchanger model suitable for small buildings in arid and semi arid regions. *KSCE Journal of Civil Engineering*, *19(4)*, pp. 853-856.

[11] *Fundamentals of Heat and Mass Transfer*, 6th, Wiley and sons, Danvers, 2005, pp. 96-400.

AUTHORS BIOS AND PHOTOGRAPHS

Dr. Philip Venter received his B.Sc. and B.Sc. Hons. in Actuarial science, B.Sc. Hons. Mathematics, B.Eng. and M.Eng. (Mechanical) and Ph.D. in Business Mathematics from the North-West University (NWU) Potchefstroom. He is currently employed as a lector at the School for Mechanical and Nuclear Engineering at the NWU. His research focus area is in thermal fluid simulations, with his main emphasis on mathematical and statistical modelling for system, control and investment optimisations.

Before joining the NWU he worked as a Mechanical Engineer, gaining industry experience.

Prof Martin van Eldik has a PhD in Mechanical Engineering from the North-West University in Potchefstroom. He is an associate professor in the School of Mechanical and Nuclear Engineering where his research group focuses on the next generation of heat pumps including the use of CO_2 as refrigerant for high temperature applications. Martin is also a Program Manager with M-Tech Industrial (Pty) Ltd mainly focusing on turnkey heat pump solutions for the commercial and industrial sectors. He is also involved with the technical and R&D aspects of their water heating heat pumps and mine cooling units.

Presenting author: The paper will be presented by Philip Venter.