Managing humidity in greenhouses using ground air heat exchangers

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Abstract

Greenhouse growing is pivotal in securing food security in Canada, where the harsh climate conditions and permafrost make outdoor farming almost impractical. Canada's intensive and frigid climate conditions create many challenges in managing the air quality within the greenhouse, which impacts crop production. In winter, the duration and period of ventilation and window opening are significantly limited because of the very low outdoor temperature. Simultaneously, 80%–95% of the irrigation water is transferred to the air within the greenhouse through crop transpiration, increasing the Relative Humidity (RH) level in the greenhouse. The current study seeks to address this problem by evaluating the Ground Air Heat Exchangers (GAHEs) that have superior efficiency compared to the conventional heat pumps that rely on electricity for dehumidification. In this study, first, a greenhouse model was used to obtain the temperature and humidity ratio of the air inside the greenhouse. Then, a transient numerical model of the proposed GAHE, relying on a thorough thermodynamic analysis, was developed. This study demonstrated that when the greenhouse air flows through the GAHE pipes, its temperature and humidity ratio decrease as the ground temperature is lower than the temperature of the greenhouse air.

Introduction

Climate control is vital in a greenhouse, providing a micro-controlled environment for plants to ensure optimum growth and high-yield food production throughout the year [1]. The critical climatic factors that need regulation include temperature, humidity, CO2 levels, and additional lighting. Decreasing the temperature of the greenhouse during the summer and increasing it during the winter requires considerable energy. Generally, this energy demand is fulfilled through conventional energy resources such as fossil fuels, further intensifying sustainability issues and environmental impact [2]. Controlling humidity in the horticulture industry is particularly challenging due to its complexity. Relative humidity fluctuates with air temperature, and plants consistently emit moisture into the air. Furthermore, regulating humidity demands significant energy, particularly in colder regions [3]. Numerous researchers have consolidated and reviewed studies focusing on the indoor climate management of agricultural greenhouses. For instance, Singh et al. [4] conducted a comprehensive review of strategies such as natural ventilation, shading, and evaporative cooling to manage the microclimate of greenhouses. Li et al. [5] also delved into the key parameters influencing the heating, ventilation, and air-conditioning systems in greenhouses. Furthermore, Ghani et al. [6] emphasized the advantages of energy-efficient cooling techniques applied in contemporary greenhouses, particularly in hot and arid environments.

High humidity levels increase the likelihood of condensation on leaves, particularly during night-time, fostering the development of Botrytis and other fungal diseases. Additionally, in conditions of high humidity, plants face challenges in evaporating water from their leaves, potentially limiting the uptake of essential nutrients like boron and calcium [7]. The primary method employed for climate control in greenhouses, particularly for dehumidification and sensible cooling, is ventilation [8]. Ventilation achieved by simply opening vents, without active pressurization is termed natural or passive [9]. On the other hand, when fans are employed to enhance air exchange, it is classified as forced ventilation [10]. Nevertheless, relying solely on natural ventilation is often insufficient, and the integration of forced ventilation not only increases greenhouse expenses but also exacerbates sustainability concerns and environmental impact.

To enhance greenhouse sustainability and minimize its environmental footprint, renewable energy sources can be harnessed to supply the required energy for various components, such as heating or cooling systems and dehumidification systems. Ozgener et al. [11] conducted a comprehensive study on the application of earth-air heat exchangers (EAHEs) to efficiently manage the heating and cooling requirements of the greenhouse. Nayaket al. [12] introduced a straightforward mathematical model under quasi-steady-state conditions for evaluating the energy and exergy of a combined opaque PV/T and earth air heat exchanger (EAHE) system integrated with a greenhouse. Mahdavi et al. [13] endeavoured to enhance the efficiency of a solar greenhouse incorporating PVT collectors and EAHE, focusing on energy and exergy efficiencies. Heyat Jilani et al. [2] investigated a Quonset-type Greenhouse integrating Thin-film Photovoltaic (GiTPV). Their conclusion highlights that the Earth Air Heat Exchanger (EAHE) effectively supplies heat within the enclosed space of their proposed GiTPV system during non-sunlit hours, leading to minimized fluctuations in room air temperature and ultimately improving thermal comfort.

While renewable energy sources, particularly geothermal energy, have been employed for greenhouse cooling and heating, they have yet to fully replace the reliance on fossil fuel energy in other aspects of greenhouse operations, such as the dehumidification system. This study aims to utilize

Ground Air Heat Exchangers (GAHEs) as an alternative with superior efficiency compared to conventional heat pumps, which typically depend on electricity for greenhouse dehumidification. This contribution presents a numerical model for GAHES to evaluate its performance in achieving greenhouse dehumidification.

Numerical modelling

A schematic of GAHE used as a subsurface greenhouse dehumidification is shown in Figure 1(a). The humid air enters the GAHE. Water vapour condenses when the humid air temperature drops below the humid air dew point, and freshwater is produced. The freshwater produced can be stored and supplied in the root zone when it is needed. The formulation of the computational model for GAHE as a dehumidification system relies on a thorough thermodynamic analysis, incorporating principles of mass and energy conservation. In developing this model, the present study considers several assumptions, which are detailed as follows:

- The air and vapour are treated as ideal gases.
- Changes in kinetic and potential energy are not considered.
- The impact of soil moisture and rainwater evaporation is negligible.
- The thermal resistance of the GAHE pipe wall is not taken into consideration.

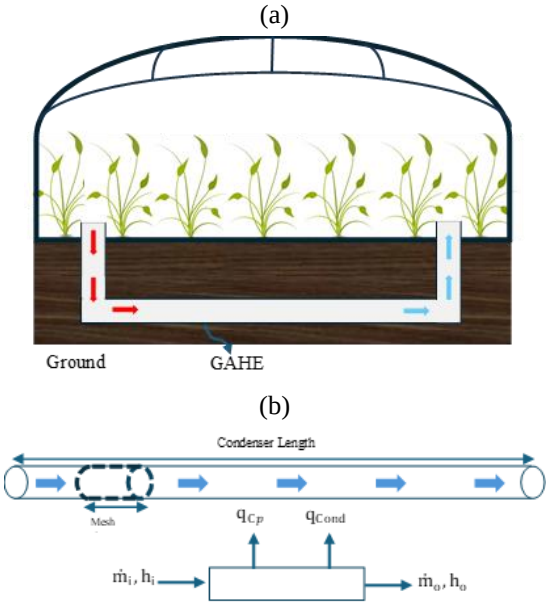


Figure 1: (a) the configuration of GAHE as a subsurface greenhouse dehumidification, (b) Air control volume in the GAHE.

Considering a small control volume of air inside the subsurface condenser (Figure 1(b)), the energy balance equation can be formulated as follows [20]:

$$\sum \dot{m}_i h_i = \sum \dot{m}_o h_o + q_{cp} + q_{cond} \quad (1)$$

$$(\dot{m}_a h_{ai} + \dot{m}_{vi} h_{vi}) = (\dot{m}_a h_{ao} + \dot{m}_{vo} h_{vo}) + q_{cp} + q_{cond} \quad (2)$$

Upon dividing equation (1) by the mass flow rate $\dot{m}_a$, we have:

$$C_a(T_i - T_o) + \omega_i h_{vi} - \omega_o h_{vo} = \frac{1}{\dot{m}_a}(q_{cp} + q_{cond}) \quad (3)$$

The calculation of the heat released by condensation, denoted as q_{cond}, is established in the following manner:

$$q_{cond} = L_w \dot{m}_{cond} \quad (4)$$

where the mass of condensed water, $\dot{m}_{cond}$, can be expressed as [14]:

$$\dot{m}_{cond} = A_p h_m \left(\varphi \rho_{v,s}(T) - \rho_{v,s}(T_p)\right) \quad (5)$$

where the convective mass transfer coefficient, h_m, is obtained as follows:

$$h_m = \frac{\mathrm{Sh} D_{AB}}{D_p} \quad (6)$$

In a pipe characterized by laminar fully developed flow, the Sherwood Number (Sh) is 3.66 [15]. Additionally, in the case of turbulent flow, the Sherwood Number is determined as follows [15]:

$$\mathrm{Sh} = 0.023 \mathrm{Re}^{4/5} \mathrm{Sc}^{0.4} \quad (7)$$

The calculation process for the binary diffusion coefficient, D_{AB}, proceeds as follows [16]:

$$D_{AB} = -2.775 \times 10^{-6} + 4.479 \times 10^{-8} T + 1.656 \times 10^{-10} T^2 \quad (8)$$

Furthermore, the expression for convective heat transfer from the air to the pipe, denoted as (q_{cp}), is as follows [17]:

$$q_{cp} = h_f A_p (T - T_p) \quad (9)$$

where for a laminar flow, h_f is defined [18]:

$$h_f = \frac{k}{D_p}\left(49.028 + 4.173\left(\frac{\mathrm{Re} \times \mathrm{Pr} \times D_p}{L_p}\right)\right)^{\frac{1}{3}} \quad (10)$$

and, for a turbulent flow [19]:

$$h_f = \frac{k}{D_p} \frac{(0.79 \ln(\mathrm{Re}) - 1.64)^{-2}}{8} \times \frac{\mathrm{Pr}}{1 + 12.7\sqrt{\frac{(0.79 \ln(\mathrm{Re}) - 1.64)^{-2}}{8}}\left(\mathrm{Pr}^{\frac{2}{3}} - 1\right)} (\mathrm{Re} - 100) \quad (11)$$

The humidity ratio of the air exiting each control volume is determined via mass balance, as outlined below:

$$\omega_o = \omega_i - \frac{\dot{m}_{cond}}{\dot{m}_a} \quad (12)$$

Changes in soil temperature occur as a result of heat exchange with the subsurface condenser pipes, a phenomenon termed as the disturbed soil temperature. In this study, the hybrid analytical–numerical model proposed by Bahmani and Hakkaki-Fard [20] for horizontal ground heat exchangers is adapted for the subsurface condenser.

This model integrates Duhamel's theorem, the modified infinite line source method, and the superposition principle. Initially utilized to compute the disturbed soil temperature without condensation in the context of a horizontal ground heat exchanger [20], the same methodology is applied here to derive the subsequent equation for calculating soil temperature in the presence of condensation:

$$T(r,t) = T_{undis} + \sum_{m=1}^{n} \frac{(Q_m - Q_{m-1})/L_p}{4\pi k_{soil}} Ei\left(\frac{C_{soil}\rho_{soil} r^2}{4k_{soil}(t_n - t_{m-1})}\right) \quad (13)$$

In this context, Q encompasses both convection and condensation heat transfers. The symbol 'n' represents the total number of time steps. The exponential integral function, denoted as Ei(x), is calculated according to the following expression for $0 \le x \le 1$ [21]:

$$\text{Ei(x)} = ln(x) + 0.577215660.99999193x + 0.24991055x^2 - 0.05519958x^3 + 0.00976004x^4 - 0.00107857x^5 \quad (14)$$

and, for x>1, $Ei(x)$ is presented in [21].

$$\text{Ei(x)} = (-(x^4 + 8.5733287x^3 + 18.059017x^2 + 8.637609x + 0.2677737)/(x^4 + 9.573233x^3 + 25.6329561x^2 + 21.0996531x + 3.9684969)xexp(x)) \quad (15)$$

Equation (13) does not include the thermal interaction among adjacent GAHE pipes. To address this, considering N GAHE pipes, the disturbed ground temperature can be determined utilizing the spatial superposition method, as described below [20]:

$$T(r,t) = T_{undis} + \sum_{j=1}^{n} (T_j((r - r_j, t) - T_{undis}) \quad (16)$$

The assumption is made that neglecting the heat resistance of the subsurface condenser pipe wall allows for considering the disturbed ground temperature at the pipe wall location as the temperature of the pipe wall.

For simulating the transition in the greenhouse, the model developed by Katzin [22] is utilized. The indoor environment comprises various factors such as temperature, humidity, and CO2 concentrations. These factors are typically represented using different equilibrium models, specifically involving energy, water vapor, and CO2. The indoor conditions are characterized by factors including temperature, humidity, and CO2 levels, which are commonly modelled using energy, water vapor, and CO2 balances. The temperature, relative humidity, and absolute humidity of the greenhouse air serve as inputs for the developed model of the GAHE. The greenhouse under study is located in Quebec City, and its dimensions and properties are detailed in Table 1. All parameters related to numerical modelling are defined in Appendix.

Table 1: The dimensions and properties of the studied greenhouse.

Parameter	Value
Mean greenhouse cover slope (°)	22
Floor area of greenhouse (m^2)	4×10^4
Surface of the cover including side walls (m^2)	4.84×10^4
Mean height of the greenhouse (m)	6.9
Maximum roof ventilation area (m^2)	0.169×10^4
Length of pipe rail system (m)	1.125

Mesh and time steps independence study

In this study, a numerical model was utilized to simulate GAHE dehumidification. A mesh independence analysis was carried out using a GAHE pipe with specific dimensions: a length of 20 m, a diameter of 20 cm, an inlet velocity of 0.2 m/s, buried at a depth of 0.5 m, and with a pipe spacing of 0.5 m. The location of the system was assumed to be in Bandar-Abbas, Iran. The simulation covered a period of ten years, during which the total condensed water amount was computed. The findings of the mesh independence study are presented in Table 1. As demonstrated in the table, decreasing the mesh size from 10 cm to 2.5 cm resulted in a relative difference in the predicted condensed water amount of less than 0.06 percent. Consequently, a mesh size of 10 cm was selected for the numerical simulations of the GAHE.

Similarly, a time-step independence investigation was conducted under the same conditions. The total condensed water was computed using three different time steps: 10, 5, and 2.5 minutes. The results of the time-step independence study are summarized in Table 2. As shown in the table, reducing the time step from 10 to 2.5 minutes led to a relative difference of less than 0.03 percent. Hence, a time step of 10 minutes was utilized for the numerical simulations.

Table 2: The results of the Mesh independence study.

Mesh size (cm)	Total condensed water (kg)	The relative variation in condensed water content with the finest mesh
10	7976.62	0.06
5	7977.23	0.03
2.5	7981.52	-

Table 3: The results of the time step independence study

Time step (min)	Total condensed water (kg)	The relative variation in condensed water content with the finest time step
10	7976.62	0.03
5	7977.32	0.02
2.5	7978.87	-

Model validation

The developed model of the GAHE is verified using the experimental results of Asgari et al. [23]. The experimental setup consists of two main components: a humidifier and a dehumidifier. The humidifier consists of a fan, a damper, a

galvanized steel box measuring 50 × 50 × 60 cm, a 5000 W electric heater, and connecting pipes. Conversely, the dehumidifier comprises a wooden box measuring 5 × 1 × 1 m filled with clay soil and a 5m polyethylene pipe. As illustrated in Fig. 5, the fan draws ambient air into the galvanized steel box, with the airflow regulated by a damper. The electric heater increases water temperature and evaporation rate, generating water vapor that humidifies the incoming air. A float valve controls the water level in the galvanized steel box. Humid air is directed into a buried polyethylene pipe at a depth of 0.75 m within the clay. The pipe has outer and inner diameters of 60.3 mm and 53.2 mm, respectively. The clay has properties including a density of 1450 kg/m³, heat capacity of 880 J/(kg·K), and thermal conductivity of 1.28 W/(m·K). Acting as a subsurface condenser, the buried pipe produces freshwater, with a slight downward tilt of 2% ensuring water collection at the pipe's end.

The experiment lasted four hours, maintaining a humid air velocity of 1.0 m/s in the dehumidifier pipe throughout. Room and initial soil temperatures were both set at 20 ℃. Fig. 6 presents a comparison between the values predicted by the developed model and the actual experimental results, utilizing Mean Absolute Difference (MAD) and Root-Mean-Square Deviation (RMSD) for evaluation.

$$MAD = \frac{1}{n}\sum_{i=0}^{n}|e_i - m_i| \qquad (15)$$

$$RMSD = \sqrt{\frac{1}{n}\sum_{i=0}^{n}(e_i - m_i)^2} \qquad (16)$$

The mean absolute deviation (MAD) and root mean square deviation (RMSD) averages for the predicted dehumidifier outlet air temperature are 0.9 ℃ and 1.3 ℃, respectively. Similarly, the average MAD and RMSD for the predicted dehumidifier outlet relative humidity are 1.3% and 1.5% respectively. These results indicate that the developed models are sufficiently reliable for analysing the performance of the GAHE in dehumidifying a greenhouse.

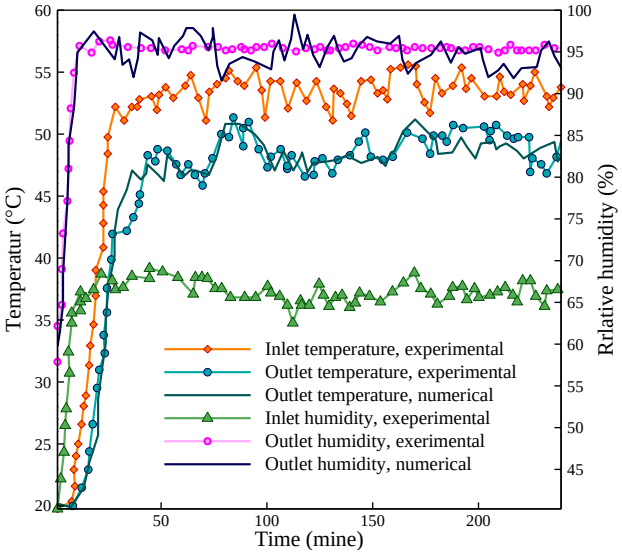


Figure 2: The predicted outlet air temperature and relative humidity from the current numerical model compared to the experimental results of Asgari et.al [23].

Results and discussion

In this study, the performance of the proposed subsurface greenhouse dehumidification is investigated in the winter and summer for a greenhouse located in the Quebec City. Besides the effect of the GAHE diameter and the number of used GAHE on the performance of the system will be investigated.

The performance of the system in the summer

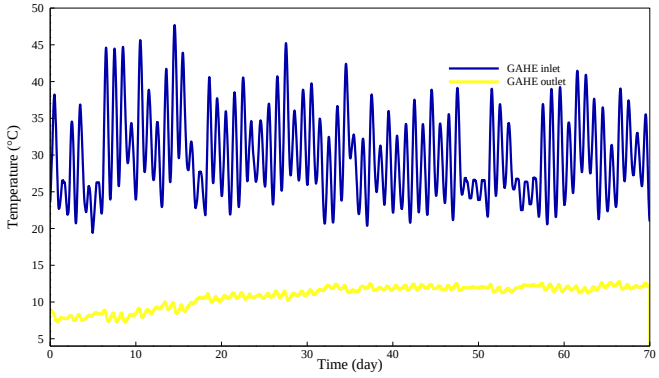
Table 4 presents the dimensions and characteristics of the GAHE utilized in this study. In Figure 3, the variations in (a) temperature and (b) humidity ratio at the inlet and outlet of the GAHE in summer are depicted. As previously mentioned, the temperature, relative humidity, and humidity ratio of the greenhouse air are determined using the code developed by Katzin [22] and set as the inlet boundary condition for the GAHE.

As observed in Figure 3(a), the temperature of the greenhouse air decreases because the ground temperature is lower than that of the greenhouse air. It's noteworthy that the average initial ground temperature is 8.5°C. Additionally, Figure 3(b) illustrates a reduction in the humidity ratio of the greenhouse air as it passes through the GAHE. This phenomenon occurs due to the condensation of water vapor in the greenhouse atmosphere when the temperature of the humid air falls below its dew point, attributed to the lower temperature of the ground compared to the greenhouse air. Consequently, the humidity level in the greenhouse air diminishes.

Table 4: The dimension and properties of the studied GAHE used as subsurface dehumidification.

Parameter	Value
Length of the GAHE pipe (m)	40
Diameter of the GAHE pipe (m)	0.2
Number of the GAHE pipes	4
Spacing between tubes buried beneath the soil (m)	0.5
The buried depth of GAHE pipes (m)	0.75
Initial inlet velocity (m/s)	0.2

(a)



(b)

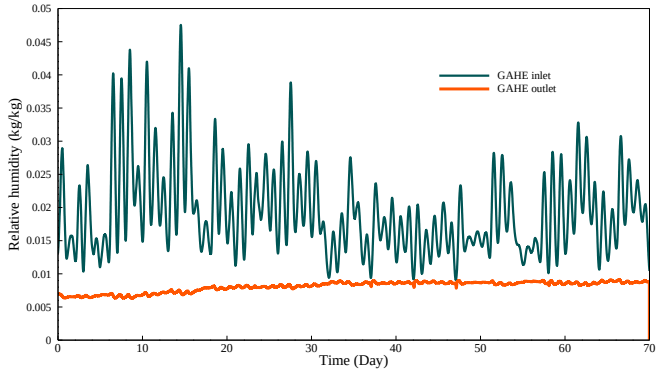
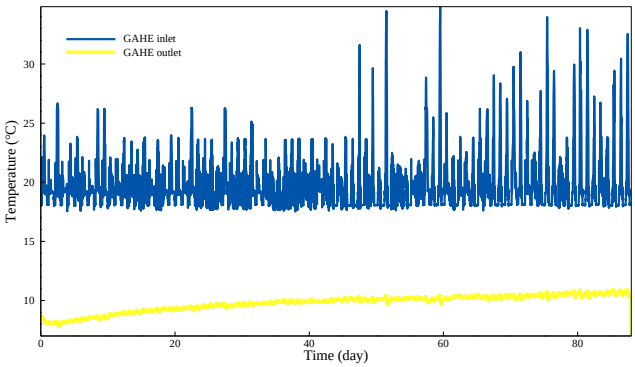


Figure 3: the variation of the (a) temperature and (b) humidity ratio in the inlet and outlet of the GAHE in summer.

The performance of the system in the winter

Figure 4 depict the variations in (a) temperature and (b) humidity ratio at the inlet and outlet of the GAHE in winter. According to this Figure, both the temperature and humidity ratio of the greenhouse decrease as they pass through the GAHE pipes. During winter, the average humidity ratio of the greenhouse air decreases from 0.01216 kg/kg to 0.0079 kg/kg when flowing through the GAHE, indicating a 39% reduction. Similarly, in summer, the humidity ratio decreases from 0.019326 kg/kg to 0.00891 kg/kg, demonstrating a 58% decrease.

(a)



(b)

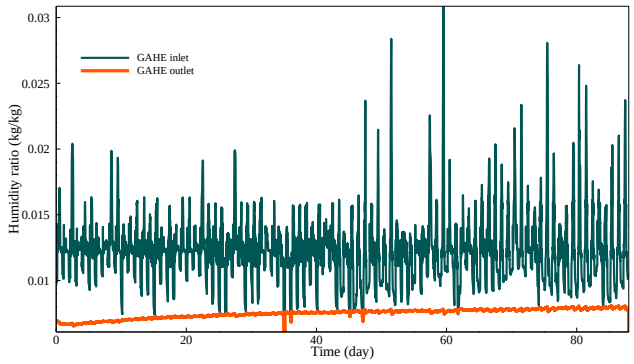


Figure 4: the variation of the (a) temperature and (b) humidity ratio in the inlet and outlet of the GAHE in winter.

The effect of the diameter and length of the GAHE pipes on the system performance

Figure 5 illustrates the impact of the diameter and length of GAHE pipes on the proposed subsurface dehumidifier. According to the figure, an increase in the diameter of the GAHE pipe leads to a higher mean humidity ratio at the GAHE outlet. For instance, when the diameter increases from 0.2 to 0.3 m, the mean humidity ratio at the outlet rises by 9.3%. This rise is attributed to the increased mass flow rate of greenhouse air and its water vapor content.

Moreover, Figure 5 demonstrates that an increase in the length of the GAHE pipe results in a decrease in the mean humidity ratio at the outlet. For example, when the length increases from 10 to 20 m, the mean humidity ratio decreases by 10%. This reduction is due to the enlarged heat transfer area between the air and the ground, facilitating greater heat loss from the greenhouse air.

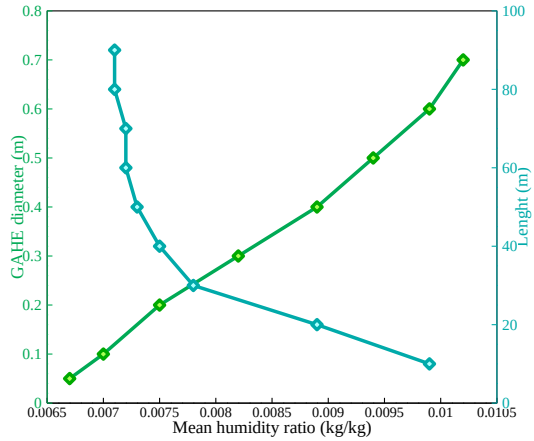


Figure 5: The impact of the diameter and length of GAHE pipes on the performance of the proposed subsurface dehumidifier.

Conclusion

This study investigated the performance of the GAHE for decreasing the humidity of the greenhouse air, which is a challenge in operating greenhouses in the north Canada. The greenhouse model developed by Katzin [22] was used to obtain the temperature and humidity ration of the air inside the greenhouse, and then transient numerical model of the proposed GAHE was developed. This model was also use for evaluating the effect of the diameter and length of the GAHE pipes on the system performance. The main idea of this research can be simplified as follows:

- When the greenhouse air flows in the GAHE pipes, its temperature and humidity ratio decrease as the ground temperature is lower than the temperature of greenhouse air.
- The performance of the system is dependent on the diameter of the GAHE pipes, and by increasing the diameter the mean humidity ratio in the GAHE outlet increases.
- The performance of the system is dependent on the length of the GAHE pipes, and by increasing the length the mean humidity ratio in the GAHE outlet decreases.

Appendix

Nomenclature			
A	area of the collector plate surface (m^2)	Sh	Sherwood number
C	specific heat capacity (J kg^{-1} K^{-1})	t	time (s)
D_{AB}	binary diffusion coefficient (m^2s^{-1})	T	temperature (K)
D_p	pipe diameter (m)	*Greek symbols*	
h	enthalpy (J kg^{-1})	ρ	density (kg m^{-3})
h_m	mass transfer coefficient (m/s)	φ	relative humidity
k	thermal conductivity (W m^{-1} K^{-1})	ω	specific humidity (kg kg^{-1})
L_p	pipe length (m)	Subscripts	
L_w	latent heat of water evaporation/condensation (J kg^{-1})	a	air
$\dot{m}$	flow rate (kg s^{-1})	cond	condensation
Q	heat exchange rate (W)	i	inlet
q_{cp}	convective heat transfer from the air to the pipe (W)	o	outlet
q_{cp}	convective heat transfer from the air to the pipe (W)	p	pipe
q_{cond}	the heat released by condensation (W)	s	saturated
Re	Reynolds number	soil	soil
Sc	Schmidt number	v	vapor
		undis	undisturbed

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