PRODUCING WATER BY CONDENSATION OF HUMID AIR IN BURIED PIPE

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Abstract

This study investigates the possibility of using warm humid air for irrigation and drinking water production, by flowing air over the water surface in a solar still with saline or polluted water. Vapor will be saturated during mixing with the warm air in buried pipe. Finite difference method is employed to simulate the flow of the air long the pipe. The amount of water produced and buried pipe length depends upon the flow velocity, humid air properties and buried pipe diameter. The amount of water produced is 0.02525kg/s (0.0909m^3/h). The length of the buried pipe needed in this study is 77.36m for a selected air flow velocity 5m/s with the properties of 70°C, 100% relative humidity at pipe inlet, 40°C and 100% relative humidity at pipe outlet of 0.2m pipe diameter. The results agree with a previous study (Gustafsson & Lindblom, 2001) with -4.0% deviation in water production and 7% of required pipe length.

Keywords
Water Production, Condensation, Humid Air, Underground, Desalination

1. Introduction
The demand of water is expected to increase by 40% in 2050 comparing to 2014 then there will be a shortage in food production. Moreover, two-thirds of the world's population will not have access to clean drinking water (Elisasson, 2015). Thus, nonconventional resources of water are an urgent need. Some oil countries such as Kuwait, Qatar, Bahrain, Saudi Arabia, and the United Arab Emirate produce 95% of the total fresh water by desalination technology using fossil

fuels (Gustafsson & Lindblom, 2001). However, sustainable resources are also required as the oil will also be in shortage within a few decades, especially in the third world countries (Wheida & Ronny, 2007).
The early work on condensation of brackish-water for irrigation, which started in 1985 (Widegren, 1986), was initiated by Bo Nordell at Luleå University of Technology, Sweden. His idea was to use solar energy to saturate air, which was conducted into buried perforated pipes where clean water is formed by condensation. This was later used by the Swiss company "Ingenieurbüro Ruess und Hausherr" and their experiments showed that the water consumption of tomato plants was reduced by 50% as the was water delivered underground, close to the roots (Gustafsson & Lindblom, 2001). The condensation irrigation technique is not only used for producing water for irrigation but also to produce drinking water. Some years later also a Doctoral Thesis on Condensation Irrigation was published (Lindblom 2012).

2. Physical model

The solar still shown in Figure 1 is nothing but a black pool filled by saline (or polluted) water. This system is covered by glass and has inlet for dry air and outlet for humid air.

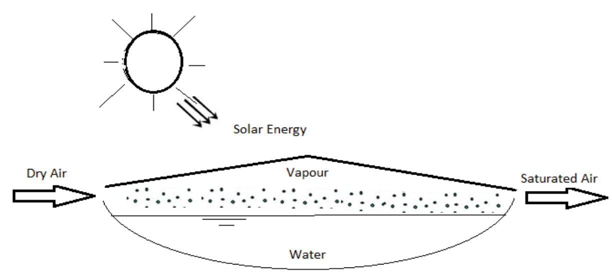

Figure 1. Simple Solar still

The solar energy will heat up the water and evaporate it. This will increase the humidity of the air which will be close to saturation when it leaves the system. The saturated air then will be forced to flow into buried pipe which works as a condenser as shown in Figure 2, and the clean water will be collected at the end of the pipe.

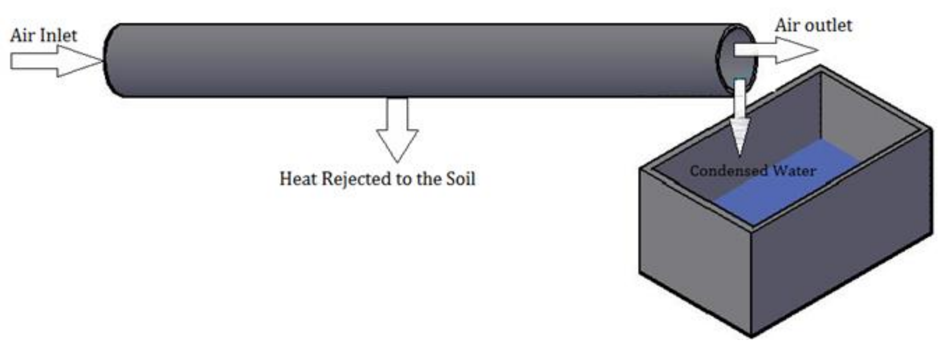

Figure 2. Condensing water from air in a pipe system

3. Mathematical Models

The information sought under the problem of this study does not require a detailed knowledge of the flow. Thus, the control volume approach is more appropriate to use rather than differential form approach. The basic fundamental laws of fluid can be applied on the system in order to achieve a mathematical formulation to the process of condensation in the pipe. Thus, the buried pipe is used as a control volume system. Current paper follows the procedure developed in 2001 by Gustafsson & Lindblom (2001).

3.1 Conservation of Mass

The objective is to determine the amount of fresh water that can be subtracted from moist air flow through a pipe at certain inlet and outlet condition as shown in Figure 3.

Figure 3. Air Inlet and Outlet of the system

Applying mass conservation equation (McQuiston & Jerald D, 2000):

$$(\dot{m}_{da} + \dot{m}_{v})_{i} = (\dot{m}_{da} + \dot{m}_{v})_{o} + \dot{m}_{w} \qquad (1)$$

Note that:

$$(\dot{m}_{da})_{i} = (\dot{m}_{da})_{o} \qquad (2)$$

$$(\dot{m}_{v})_{i} = \dot{m}_{da}\omega_{i} \qquad (3)$$

$$(\dot{m}_{v})_{o} = \dot{m}_{da}\omega_{o} \qquad (4)$$

Substitute from Eqs. 2,3, and 4 into Eq. 1 yields:

$$\dot{m}_{w} = \dot{m}_{da}(\omega_{i} - \omega_{o}) \qquad (5)$$

The last relation represents the amount of fresh water produced by the system.

3.2 Conservation of Energy

The difference in air properties at the pipe inlet (energy in) and the outlet (energy out) of the pipe means that some heat has been transferred through the pipe wall to the ground surrounding the pipe. This heat transfer is caused by the temperature difference between the flow and the wall surface of the pipe. The vapor of the air precipitates as water on the inner surface of the pipe as shown in the Figure 4.

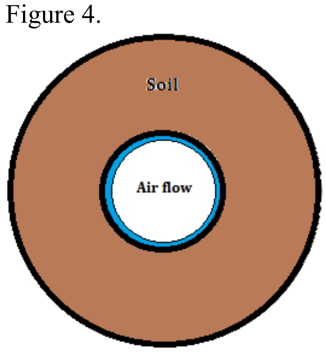

Figure 4. System cross section and the surrounding soil

This phase change will release the latent heat of the humid air. Part of this heat will be carried with the flow. The other part will be conducted from the wall surface of the pipe into the surrounding soil. Both thermal and dynamic boundary layers of the pipe flow can be assumed to be fully developed throughout the pipe length, where the pipe diameter is very small compared to the temperature interval length. The heat balance per unit length for one temperature interval is determined by the following equation:

$$q' = q'_{cout} + q'_{conv} \qquad (6)$$

The following sections illustrate the complete analysis of the energy equation, then it will be shown how the pipe length can be estimated.

3.3 Convective Heat Transfer

Because of the temperature difference between the flowing humid air and the pipe surface, forced heat convection occurs. Besides of the temperature difference, the rate of heat convection typically depends on the surface area of the pipe and the convection coefficient. Both the temperature difference and contact area are measured factors, while the heat convection coefficient depends on the flow properties and can be calculated from Eq. (7) (Incropera F.P & Dewitt D.P, 2011).

$$h_D = \frac{Nu_D k_a}{D} \qquad (7)$$

Where the Nusselt (Nu) correlation for $(3 \times 10^3 \leq Re_D \leq 5 \times 10^6)$ and $(0 \cdot 5 \leq Pr \leq 2000)$ is described by the Gnielinski equation (Welty, Wicks, Rorrer & Wilson, 2007):

$$Nu_D = \frac{(Re_D - 1000)\mathrm{Pr}(\frac{f}{8})}{1 + 12 \cdot 7\left(\frac{f}{8}\right)^{\frac{1}{2}}(Pr^{\frac{2}{3}} - 1)} \quad (8)$$

And

$$Re_{D,n} = \frac{C_n D}{\upsilon} \quad (9)$$

$$Pr = \frac{\upsilon}{\alpha} \quad (10)$$

$$f = (0.790 \ln(Re_D) - 1.64)^{-2} \quad (11)$$

The difference between the air and surface temperature within an interval is determined by:

$$\Delta T = \frac{T_n + T_{n+1}}{2} - T_s \quad (12)$$

Finally, the heat transfers per unit length due to convection can be determining by the following equation:

$$q'_{conv} = h_D \pi D \Delta T \quad (13)$$

3.4 Condensation heat and mass transfer

The latent heat of the humid air flow will be released by convection, due to the change of phase from vapor to liquid of the humid air. The rate of change from vapor to liquid is given by the mass transfer coefficient $h_m(\frac{m}{s})$ which can be calculated as follows (David, 1995).

$$h_m = \frac{h_D D_{AB} Le^{1/3}}{k_a} \quad (14)$$

The Lewis number Le is the ratio of the thermal and mass diffusivities, and it is only relevant when both heat and mass transfer occur at the same time, and can be calculated as in the following equation (Incropera & Dewitt, 2011).

$$Le = \frac{\alpha}{D_{AB}} \quad (15)$$

Once the mass transfer coefficient calculated, the amount of water condensed in pipe for each pipe interval per unit area can be easily calculated by evaluating the properties of the flow at the mean temperature of each interval as in the following equation:

$$\dot{m}''_C = h_m(\rho_s - \rho_a) \quad (16)$$

Consequently, the latent heat released per pipe length caused by conduction can be calculated as following:

$$\dot{q}'_c = \dot{m}''_c \pi D h_{fg} \quad (17)$$

Where (h_{fg}) is the latent heat of vaporization for water at the mean temperature of the interval.

The amount of this latent heat splits into two parts. The first part flows radially due to temperature difference between the soil and the pipe surface while the second part flows in the direction of the fluid flow because of the inlet temperature interval is higher than the outlet one. Thus, the amount of each part depends on the ability of transporting heat in that medium. Therefore, the amount of latent heat which is consumed by soil can be easily computed by the following equation based on its share of transferability.

$$q'_{cout} = \frac{q'_c U_{soil}}{U_{soil} + h_D} \quad (18)$$

U_{soil} is the overall heat transfer coefficients of soil calculated from the pipe surface to a distance r_{soil} (m) in the radial direction out from the pipe where the soil is no longer penetrated by the heat transfer with the pipe. Since the pipe wall is very thin, the heat transfer is calculated directly from the air to the soil using the following equation.

$$U_{soil} = \frac{k_{soil}}{r_s \ln(\frac{r_{soil}}{r_s})} \quad (19)$$

4. Estimating the Interval Length

The total length of the pipe (Z) will be divided into several intervals. Each interval capable of cooling the fluent by five degrees. Now, the required length of each interval should be defined and then added up to find the total length. Basically, The length of one temperature interval is equal to the loss of water content in the air divided by the condensation rate per unit length, as represent by the following equation:

$$z = \frac{\dot{m}_{da}(\omega_n - \omega_{n+1})}{\dot{m}_c''\pi D} \qquad (20)$$

The dry air flow rate $\dot{m}_{da}(\frac{kg}{s})$ can by calculated using following equation:

$$\dot{m}_{da} = \frac{\rho_a C_n \pi r_s^2}{1+\omega} \qquad (21)$$

Where $\rho_a(\frac{kg}{m^3})$ is the density of humid air and is given by Peterson (1978) as:

$$\rho_a = \frac{P_t}{R_{da}(T_a + 273.15)} - \left(\frac{1}{R_{da}} - \frac{1}{R_w}\right)\frac{P_{vsat}\varphi}{(T_a + 273.15)} \qquad (22)$$

Obviously, the amount of water content in the air in later interval will reduced by the amount of water condensed in former interval as shown in the following equation:

$$\dot{m}_{a(n+1)} = \dot{m}_{a(n)} - \dot{m}_c''\pi D z \qquad (23)$$

The mass of humid air in this interval is equal to:

$$\dot{m}_{a(n)} = C_n \rho_{a(n)} r_s^2 \pi \qquad (24)$$

Consequently, the velocity in later interval decreases as well and calculated as in the following equation:

$$C_{n+1} = \frac{\dot{m}_{a(n+1)}}{\rho_{a(n+1)} r_s^2 \pi} \qquad (25)$$

With the inlet velocity known for the next temperature interval, its heat rate and length of the interval can be evaluated as well. The total length of the pipe is easily calculated when all the interval lengths are estimated. The sum of the lengths of temperature intervals will be the total pipe length. Steady state process will be achieved after some time, and no longer change will occur for the pipe length and heat rate. The reason behind reaching the steady state condition in the pipe system is due to the period which is needed for heat distribution process in the soil. This issue will be discussed in the next section.

5. The Heat Distribution in the Soil

The surface temperature of the pipe is higher than the temperature of the surrounding soil due to the flow of the humid air. This makes the heat flows radially out of the pipe. This process will be function of time for a period until a maximum penetration depth achieved and the processes becomes independent of time.

To simulate this process, Finite Difference Method FDM will be used. However, this technique provides an approximate solution and variables are founded only at discrete points in time and space. In other words, the solution is not continuing. The space coordinate called nodal points, or nodes for short, and the distance between two discrete points in time are called time step. These discrete points are chosen depending on geometry and the desired accuracy. The nods are set for this calculation is shown below in Figure 5.

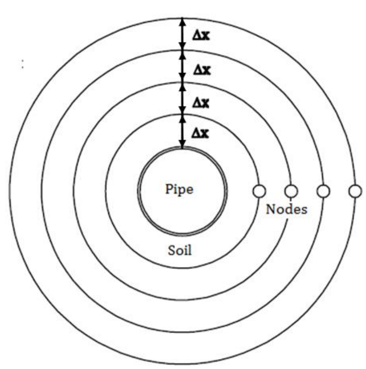

Figure 5. Nodes at Δx distance from each other

By using any numerical solution technique, there are errors should be minimized as much as possible to converge the results and eliminate instability. These errors mainly caused by finite precision of computations, and truncation error. Thus, the values of nodal distance and time step must be chosen to be with a certain value. This can be assured by satisfying the following inequality (Incropera & Dewitt, 2011):

$$Fo(1+Bi) \leq \frac{1}{2} \qquad (26)$$

Where:

$$Fo = \frac{\alpha_{soil}\Delta t}{(\Delta x)^2} \qquad (27)$$

$$Bi = \frac{U_a \Delta x}{k_{soil}} \qquad (28)$$

Here U_a $(\frac{W}{m^2K})$ is the total convection coefficient for the air and it is calculated as the following equation:

$$U_a = \frac{q'}{\Delta T \pi D} \qquad (29)$$

When the time step and the node distance finally chosen then the time dependent temperature distribution in the soil may be estimated. The initial temperature in each node must be known. This boundary condition is known at time equal zero $(t = 0)$, at which the pipe surface and the surrounding soil have one uniform temperature along the pipe. The surface temperature after time of Δt seconds from the time t can be estimated by the equation below:

$$T_s^{t+\Delta t} = 2Fo(T_a^t + Bi\bar{T}) + (1 - 2Fo - 2Bi.Fo)T_s^t \qquad (30)$$

Where $\bar{T}$ is the mean temperature of the air in the interval of interest, which is estimated by the following equation).:

$$\bar{T} = \frac{T_n + T_{n+1}}{2} \qquad (31)$$

The temperature in the nodes outside the pipe is only influenced by conduction in the soil, whereas the surface temperature also is affected by the convection heat transfer from the air. The equation which approximate this behavior in the nodes is given below:

$$T_j^{t+\Delta t} = Fo\left(T_{j-1}^t + T_{j+1}^t\right) + (1 - 2Fo)T_j^t \qquad (32)$$

Eqs.(30-32) are all from Incropera & Dewitt (2011)

6. Calculations

All the physical equations of this model were developed. It is time now of specifying the boundary conditions and develop a computer program code using Q-Basic (David, 1995) to simulate the system behavior, the flow chart program is shown in Figure 6[1].

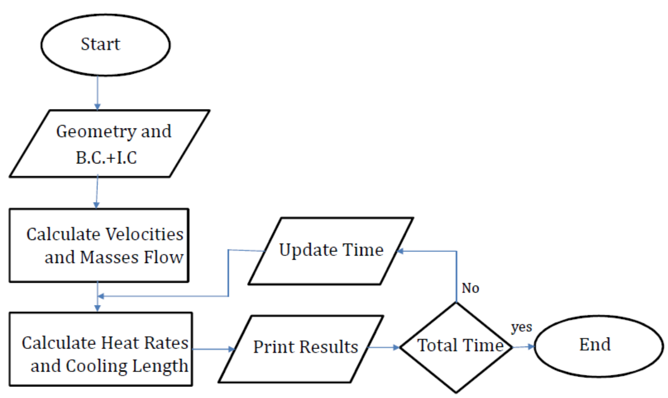

Figure 6. computer Program flow chart

The presumptions made as the following:

- Cooling the air flow from 70 C° and 100% relative humidity to 40 C° and 100% relative humidity.

[1] Some other relations were used too such as equation of state. Also, some fluid properties were fed directly to the program at different conditions.

- Pipe diameter is set to be 0.2 m.
- Inlet velocity is set to be 5m/s.
- The pipe is divided into temperature intervals 5 C^o each.
- The radial distance between two nodes in the soil is 0.125 m.
- The time step is set to be 900 s.
- The soil conductivity coefficient is set to be 1.5 W/(mK).
- The undistributed soil temperature is set to be 18 C^o.
- The thermal resistance of the pipe wall is neglected

7. Results

The results of the current study and reference study, in terms of the amount of condensed water and the total cooling length are presented in table (1) below. The deviation of the current study from Gustafsson & Lindblom (2001) is only (-0.4%) in term of condensed water and (+7%) in term of total length.

Table 1. Final results for current study and reference study

Boundary Condition			Current Study		(Gustafsson & Lindblom, 2001).	
Interval	Inlet Temperature (C^o)	Outlet Temperature (C^o)	Condensed Water (kg/s)	Cooling Length (m)	Condensed Water (kg/s)	Cooling Length (m)
1	70	65	0.00806	23.63	0.00807	22.00
2	65	60	0.00573	17.29	0.00576	13.40
3	60	55	0.00419	13.85	0.00422	11.20
4	55	50	0.00312	11.43	0.00314	10.30
5	50	45	0.00235	9.27	0.00237	10.10
6	45	40	0.00178	7.66	0.00180	10.60
Total			0.02525	83.13	0.02536	77.60

The total cooling length and condensed water flow rate vary with time as shown in Figure 7. The condensed water flow rate is steady and depend on the inlet and outlet conditions only, based on the presumption made in the previous section. The total cooling length will change with time as long unsteady heat transfer exist, after some time of running, steady state heat transfer will be reached and no longer total cooling length change will occur.

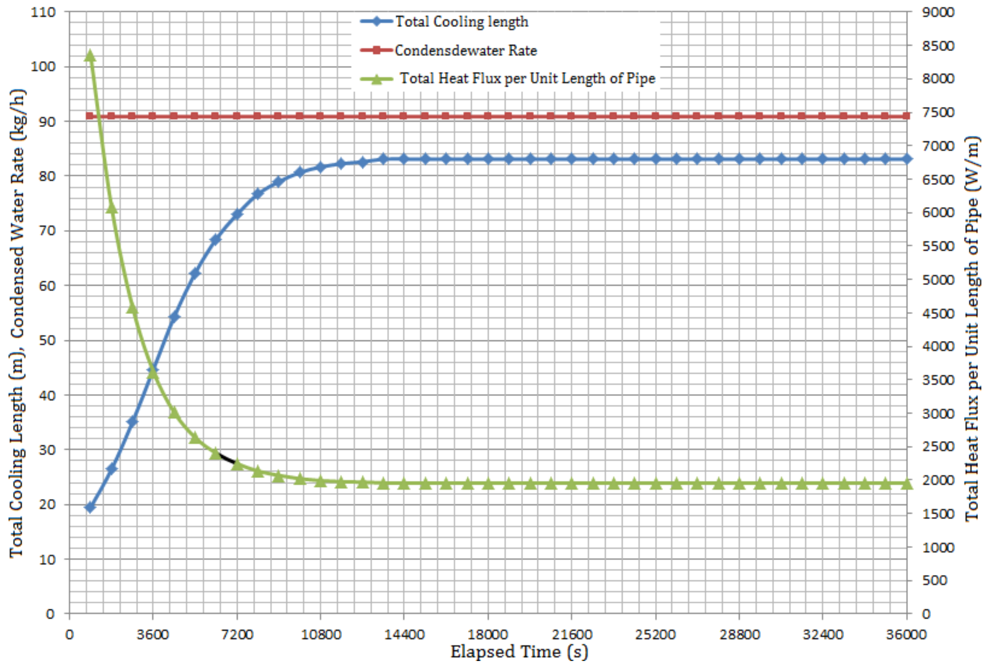

Figure 7. Results behavior

8. Sensitivity Analysis for chosen variables

In this section it will investigated how the pipe length and condensed water rate vary with different inlet velocities, pipe diameters and soil conductivities coefficient. The results are shown after ten hours of running.

8.1 Sensitivity Analysis for inlet velocity

The changes of cooling length and the amount of condensed water flow rate with inlet velocities $(1\frac{m}{s} \leq C \leq 9\frac{m}{s})$ is shown in Figure 8 ,while keep all the other parameters as the presumption made in section (6).It is noticed that, with increasing the inlet velocity the mass flow rate will increase which increase the water vapor inflow and gives an opportunity to extract more liquid water, but consequently the cooling process will needs more time because the flow moves faster and the total cooling length will be longer despite of increasing the heat transfer coefficient with the velocity increase.

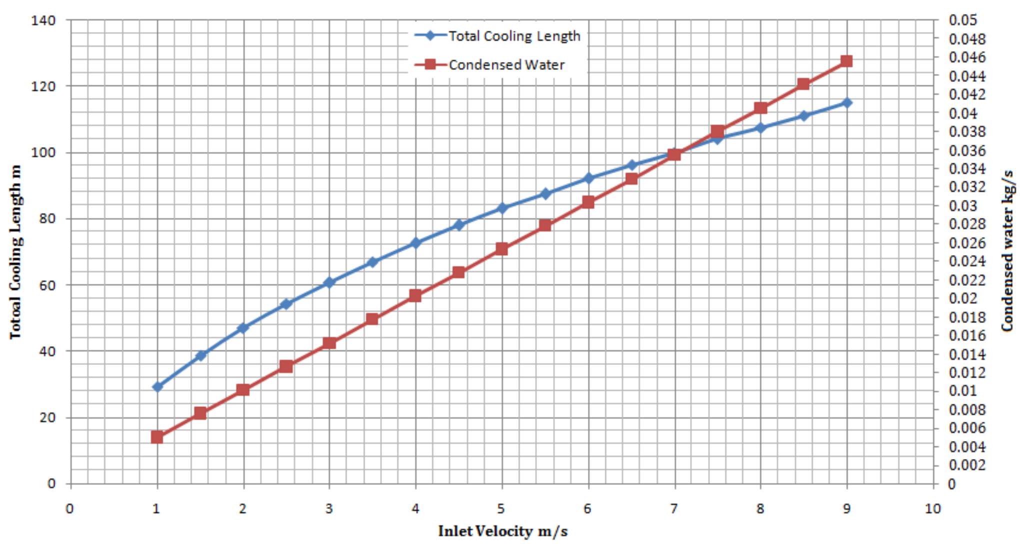

Figure 8. Sensitivity Analysis for Inlet Velocity

8.2 Sensitivity Analysis for Pipe diameter

The change of cooling length and the amount of condensed water flow rate with pipe diameter $(0 \cdot 1m \leq D \leq 0 \cdot 3m)$ is shown in Figure 9, while keep all the other parameters as the presumption made in section (6). Increasing the pipe diameter will increase the cross-section area and consequently will increase the mass flow rate, followed up with increasing in both cooling length and condensed water.

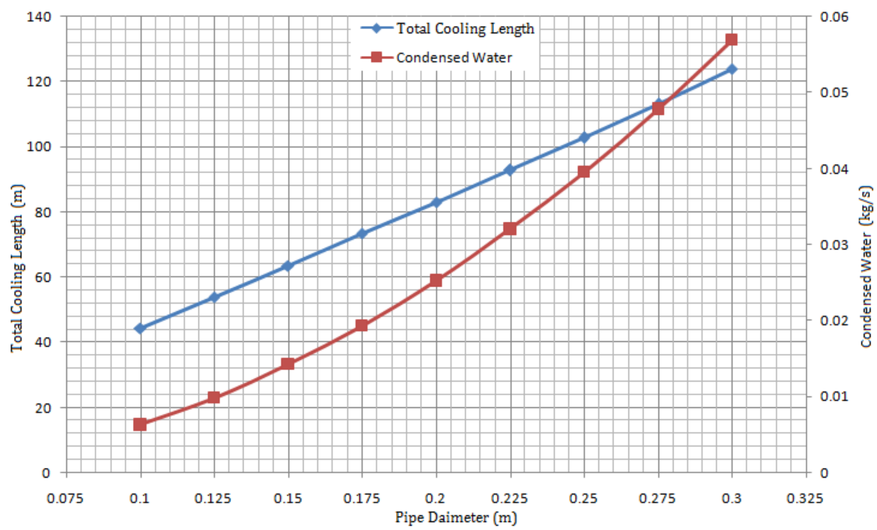

Figure 9. Sensitivity Analysis for Pipe Diameter

8.3 Sensitivity Analysis for Soil Conductivity Coefficient

The change of cooling length and the amount of condensed water flow rate with soil conductivity coefficient $(1W/mK \leq k \leq 1.5W/mK)$ is shown in Figure 10, while keep all the other parameters as the presumption made in section (6).When the soil conductivity coefficient increases, the rate of heat transfer will increase, so the total required cooling length will be shorter, while the amount of condensed water flow rate will remain the same because the mass flow rate does not affected by the change of soil conductivity coefficient as long as the inlet and outlet conditions still as they are.

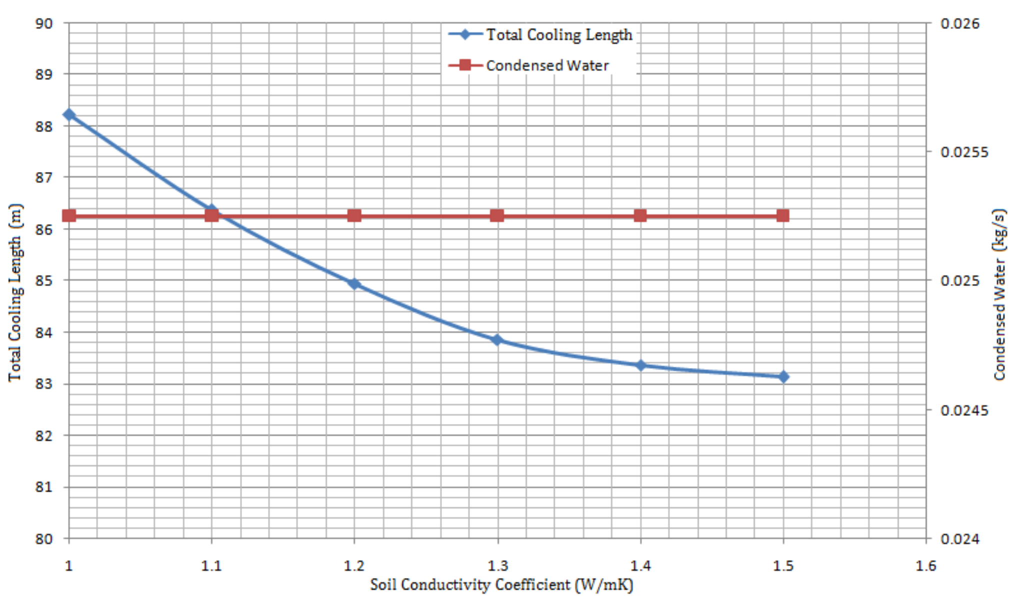

Figure 10. Sensitivity analysis for the soil conductivity coefficient

9. Conclusions

From the results obtained from this study, the following conclusion can be drawn:

- The amount of water produced depends on the inlet and outlet conditions only.
- The total cooling length of the pipe depends on the inlet, outlet conditions and also it is time dependent.
- Increasing inlet velocity will be followed up with increasing in the water produced and the total cooling length.
- Increasing pipe diameter will be followed up with increasing in the water produced and the total cooling length.
- Increasing soil conductivity coefficient will be followed up with decreasing in the total cooling length and will never affect the water produced.
- Amount of water produced can be used for irrigation system besides using it for drinking.

10. Recommendations

Based on the results of this study and the experience of other researches, the following recommendation can be drawn:

- It is expected that the produced water for drinking in this process is lacking some kinds of mineral.
- In order to model the system for real design, the roughness effect should be put into account and soil parameters which are differ from place to place.
- Experimental study is needed to obtain results about the properties effect on condensation flow rate taking into account the local weather conditions.
- Further studies are needed to evaluate the economics of condensation water production especially for agriculture use.

NOMENCLATURE

Bi	Biot number	Dimensionless
C	Velocity	m/s
D	Diameter	m
D_{A-B}	Diffusion coefficient of A in B	m^2/s
f	Friction factor	Dimensionless
Fo	Fourier number	Dimensionless
h_D	Convective heat transfer coefficient	W/m^2K
h_{fg}	Latent heat of vaporization	kJ/kg
h_m	Convective mass transfer	m/s
k	Conductivity coefficient	W/mK
Le	Lewis number	Dimensionless
m	Mass	kg
$\dot{m}$	Mass flow rate	kg/s
$\dot{m}''$	Mass flow rate per unit Area	kg/m^2s
n	Interval Number	
Nu_D	Nusselt number	Dimensionless
P	Pressure	Pa
Pr	Dimensionless	Prandtl number
q'	Heat flux per unit length	W/m
R	Gas constant	kJ/kgK
Re_D	Reynold number	Dimensionless
T	Temperature	C^o
V	Volume	m^3
X	Mole fraction	Percent or Fraction
Z	Interval Length	m

Greek Characters

α	Thermal diffusivity	m^2/s
ν	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
φ	Relative humidity	Percent or Fraction
ω	Humidity ratio	kg_v/kg_{da}
Δt	Time step	s
Δx	Distance between two nodes	m

Subscript

a	Air
C	Condensation
Conv	Convection
da	Dry air
i	Inlet
n	Interval
o	outlet
s	Surface
sat	Saturated
t	Total
v	Vapor
w	Water

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Biography

Dr. M.A.Muntasser is presently professor in Mechanical Engineering Department, Tripoli-University. He is also the Chairman of the board of LAPEDI (Oil and Gas Co.) registered in Spain. He started his carrier in ESSO Standard Company in Libya, as pipe line Construction Engineer. He worked also different committees at Tripoli University, he joined as member of Supervisor committee for the construction of Brega – Misurata new Pipe Line. He introduced many companies to the oil industry. His degree includes, B.Sc. in Mechanical Engineering (Thermal Science), University of Tripoli, Libya (1969), M.Sc. in Mechanical Engineering (Thermal Science) Purdue University, W. Lafayette, Indiana, USA (1972) and PhD in Ph.D. Mechanical Engineering (Thermal Science) North Carolina State University, Rayleigh, North Carolina, USA (1978). He authored 100 papers on energy and environment and contributed numerous technical presentation at major conference. Dr. M.A. Muntasser served different posts at Tripoli university till become the Associate Dean of Engineering. He served as a chairman of MPC's series conference.

Currently, Mr. A Elhemmali is studying on a master program in mechanical engineering at Memorial University in Newfoundland, Canada. He has a BSc from Tripoli-University, Libya (2007) in Mechanical and Industrial Engineering (Thermal science). Mr. Elhemmali has almost five years experience in oil industry as a Field Engineer who specialized in service well testing, applying a verity of measurement, sampling, and analysis technologies and techniques to provide information such as petroleum composition, flow rate, and downhole pressure and temperature.

Dr. Jenny Lindblom has presently a research position at Soil Mechanics, Mining and Geotechnical Engineering, Luleå University of Technology. Her degree includes a MSc (2001), LSc (2006) and PhD (2012) in Water Resources Engineering at Luleå University of Technology, Luleå, Sweden.

Dr. Bo Nordell is since 2001 professor in Water Resources Engineering, Luleå University of Technology. His degree includes a MSc (1979) in Mining Eng., PhD (1994) in Water Resources Eng., Docent (1999) in Water Resources Eng., all at Luleå University of Technology, Luleå, Sweden