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SIMULATION OF THE CONDENSER OF THE SEAWATER GREENHOUSE
Part I: Theoretical development

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A theoretical model is formulated in this Part 1 of the paper for simulating the physical process of condensation of the humid air in the condenser of seawater greenhouse that is located in Muscat, Oman. Analyses to the equations, in addition to the theoretical developments of the proposed model are discussed.

Keywords: *condensation, condenser, heat and mass model, seawater greenhouse, simulation, sultanate of Oman*

Introduction

The seawater greenhouse is a new development that produces fresh water from sea water, and cools and humidifies the growing environment, creating optimum conditions for the cultivation of temperate crops [1]. The use of greenhouses in arid regions decreases crop water requirements by reducing evapotranspiration. The plastic cover utilized on these structures changes locally the radiation balance by entrapping long-wave radiation and creates barrier to moisture losses. As a result, evapotranspiration is reduced by 60 to 85% compared to outside the greenhouse [2]. The greenhouse acted as a solar still providing a controlled environment inside the greenhouse. The greenhouse is equipped with humidification–dehumidification devices which create the proper climate to grow valuable crops and at the same time produce freshwater from saline water.

In this present study, a theoretical model was developed to simulate the physical process of condensation of the humid air in the condenser of seawater greenhouse. The theoretical aspects were given along with the model development.

Seawater greenhouse (SWGH) process description

Seawater greenhouse is mainly used for the production of fresh water from humid air and for providing environmental conditions favorable to the growth of crops in the greenhouse. The idea of its operation depends on creating the natural water cycle in a controlled environment (Fig. 1). The water cycle starts by pumping the seawater into a storage tank named ‘cold tank’ inside the SWGH after filtration of the seawater through layers of sandy soil. The seawater is then pumped to the condenser where it passes through the condenser tubes before reaching the first cooling pad evaporator at the front side of the SWGH. The seawater passes through the first evaporator from top to bottom while air introduces through it into the SWGH,

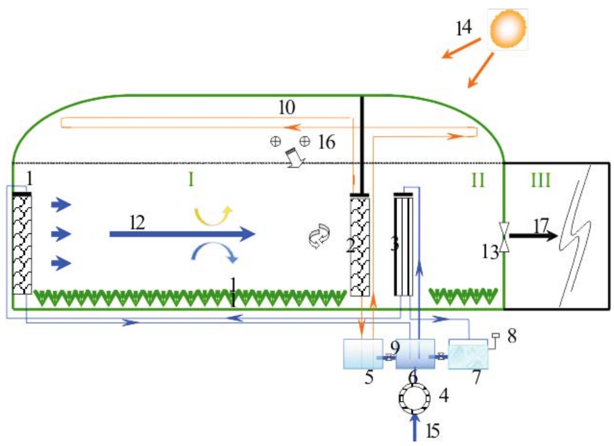

Fig. 1 Seawater greenhouse process schematic diagram [1]. 1 – evaporator 1, 2 – evaporator 2, 3 – condenser, 4 – well, 5 – hot seawater tank, 6 – cold seawater tank, 7 – fresh water, 8 – conductivity apparatus, 9 – supplement valve, 10 – solar heating pipes, 11 – crops, 12 – cooled and humidified air, 13 – fans, 14 – solar energy, 15 – seawater intake, 16 – ventilator, 17 – dry and cold air, I – first compartment, II – second compartment, III – shadow room

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by two sucking fans installed on the other side, in a perpendicular direction to the flow of water. This process contributes to a consistent humidification of the introduced air causing it to be relatively cold and humid through the entire greenhouse to undergo a second humidification process at the second evaporator located 1.3 m from the condenser. Both evaporators work as compact exchanger of mass and heat and they consist of cardboard honey-comb lattices. To provide a homogenous mixture of humid air, the greenhouse is equipped with two ventilators (the sucking fans) whose size depends on the geometric characteristics of the greenhouse.

The cold tank supply seawater to an adjacent tank named 'hot tank'. The seawater in the hot tank is pumped to solar panel pipes installed at 1.7 m below the plastic cover surface inside the SWGH to heat up, by solar energy, the seawater before passing through the second evaporator whose operation is identical to that of the first evaporator. The cold air passes through the second evaporator in a perpendicular direction to the flow of hot seawater warms up and reaches its saturation point before comes in contact with condenser tubes. It was noted that only a small fraction of solar radiation involved in photosynthesis, because the roof selectively filter solar radiation incident. Such technology aims maintaining appropriate conditions for the development of plants namely a relatively fresh and fairly light.

The saturated humid air from the second evaporator passes through the condenser where cold seawater flows inside its tubes. Part of the humid air condenses on the walls of vertical tubes of the condenser. The resulting condensate is collected to be routed to the storage reservoir for use in irrigation of crops. Table 1 illustrates the principal parameters of the design of the seawater greenhouse in the Sultanate of Oman. In order to analyze how it works in real time, the seawater greenhouse is equipped with thermocouples and hygrometers arranged in different places to measure the dry bulb temperature (T_{db}) and relative humidity (*RH*). Moreover, the intensity of solar fluxes inside and outside the SWGH are determined as well by solar radiation pyranometers. The dry bulb temperature, relative humidity and solar radiation values were collected from 20 December 2005 to 7 February 2006. The data were taken every half an hour.

Table 1 Design parameters of the seawater greenhouse

Width/m	16
Length/m	45
Maximum height/m	4.8
Maximum speed of the air/m s^{-1}	7.1
First evaporator/m	15.6×2×0.2
Second evaporator/m	15.6×2×0.2
Condenser/m	15×1.9×0.8

Condenser process description

The condenser of the seawater greenhouse is a heat exchanger where the seawater is the coolant and the humid air is the hot fluid (Fig. 2a). The condenser consists of a set of 302 rows of parallel tubes

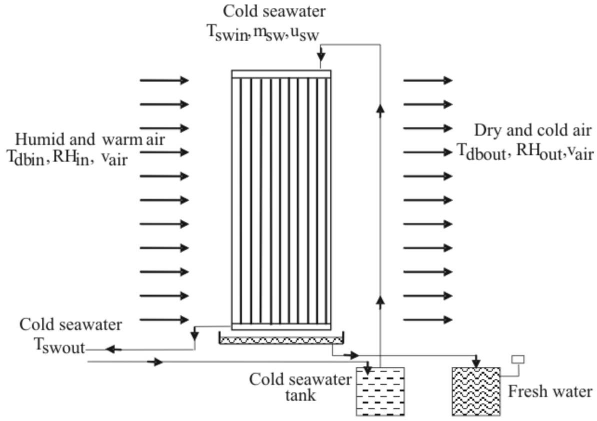

Fig. 2a Process schema of the condenser used in the seawater greenhouse

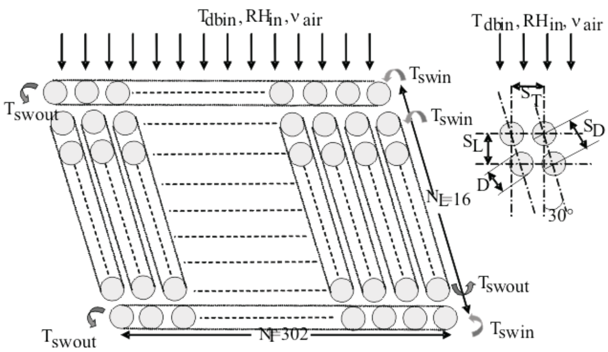

Fig. 2b Detailed schema of the condenser used in the seawater greenhouse

Table 2 Design parameters of the condenser of the greenhouse

Condenser/m	15×1.9×0.8
Thickness of the vertical tube (δ)/μm	200
Height of the vertical tube (*L*)/m	1.8
Diameter of the vertical tube (D_{out})/mm	33
Transverse pitch (S_T)/mm	46.7
Longitudinal pitch (S_L)/mm	40.5
Diagonal pitch (S_D)/mm	46.7
Angle (α)/°	30
Number of longitudinal tubes (N_L)	302
Number of transverse tubes (N_T)	16
Total number of tubes (N_{tot})	4832

(Table 2) arranged vertically and with an angle of 30° with the direction of flow of humid air (Fig. 2b). Each row has 14 identical vertical tubes with diameter of 33 mm (D) and a height of 1.8 m (L). The arrangement of the tubes was organized to ensure the passage of coolant from one tube to another. All tubes passes in a single row have the form of a coil. Seawater enters with a constant speed (u_{sw}) and a known temperature (T_{swim}) in the first row of each tube and it leaves the last tube in the same row with a temperature T_{swout}. The humid air from the second evaporator runs perpendicular to the condenser. It enters through the tubes with a velocity (v_{air}), temperature (T_{dbin}), and a relative humidity (RH_{in}). This humid air will leave the condenser with the same speed v_{air}, with a temperature of T_{dbout} and a relative humidity of RH_{out}. The contact of humid air with the outer cold surfaces of the tubes of the condenser (in which seawater is flowing) will result in the condensation of the water vapor in the humid air at the outer surfaces of these tubes. The produced condensate which will be formed as a liquid film of low thickness will descend along the tubes to be collected in the reservoir of fresh water.

Theoretical modeling of the condenser

The following steps were followed in modeling condenser of the seawater greenhouse:

- Step I: Determination of the saturation temperature of the air (T_{sat}) for each tube of the condenser.
- Step II: Determination of the average heat transfer coefficient of condensation (h_{ave}) for vertical tubes of the condenser.
- Step III: Determination of the temperature of the outer wall (T_{wout}) of each tube of the condenser.
- Step IV: Determination of the condensate mass flux of each tube of the condenser (m_c) and the total condensate mass flux from all the tubes (m_c)$_{tot}$.

Detailed procedures on how the above steps were accomplished are described as follows:

Step I: Determination of the saturation temperature of the air (T_{sat}) for each tube of the condenser

Data of the dry bulb temperatures of the air together with relative humidity are collected only at the inlet and the exit of the condenser of the seawater greenhouse. Linear interpolation was assumed in order to find out the dry bulb temperature and the relative humidity for each tube of the condenser by using the known information at the inlet and outlet of the condenser. The simulation of the operation of the condenser requires that the adiabatic saturation temperature to be known at each tube in the condenser. For this purpose, Eqs (1)–(8) were used. According to Coulson and Richardson [3], information about these equations can be found in Sherwood and Comings [4].

Calculation of $(P_{sat})_{T_{db}}$

The pressure of the water vapor at the adiabatic dry bulb temperature $(P_{sat})_{T_{db}}$ was calculated for each tube of the condenser according to Eq. (1):

$$\lg(P_{sat})_{T_{db}} = 8 - \frac{1689.52}{230 + T_{db}} \tag{1}$$

where $(P_{sat})_{T_{db}}$ is the vapor pressure of pure water at the dry bulb temperature (mmHg), T_{db} is the adiabatic dry bulb temperature (°C).

Calculation of P_{vap}

The partial pressure of water vapor in the air-water mixture for each tube (P_{vap}) was calculated according to Eq. (2):

$$P_{vap} = (P_{sat})_{T_{db}} RH \tag{2}$$

where P_{vap} is the partial pressure of water vapor in the air–water mixture (mmHg), RH is the relative humidity (%).

Calculation of H

The absolute humidity (H) was calculated according to Eq. (3):

$$H = \left(\frac{18}{29}\right)\frac{P_{vap}}{760 - P_{vap}} \tag{3}$$

where H is the absolute humidity of air at the adiabatic dry bulb temperature (kg water vapor/kg dry air).

Calculation of h_{fg}

The heat of condensation of vapor in the air was calculated by determining the latent heat of vaporization (h_{fg}) according to Eq. (4):

$$h_{fg} = 2502\left[\frac{1 - \left(\frac{T_{sat}}{T_{cr}}\right)}{1 - \left(\frac{273.15}{T_{cr}}\right)}\right]^{0.38} \tag{4}$$

where h_{fg} is the latent heat of vaporization at adiabatic saturation temperature (kJ kg^{-1}), T_{sat} is the adiabatic saturation temperature of air (K), T_{cr} is the critical temperature of water (K), T_{cr}=647.3 K [5].

Calculation of H_{sat}

The saturation humidity at the adiabatic saturation temperature (H_{sat}) was calculated for each tube of the condenser according to Eq. (5):

$$\frac{H_{sat} - H}{T_{sat} - T_{db}} = -\frac{C_s}{h_{fg}} \tag{5}$$

where H_{sat} is the saturation humidity at the adiabatic saturation temperature (kg kg^{-1}), C_s is humid heat (kJ kg^{-1} °C^{-1}) and was calculated according to Eq. (6):

$$C_s = C_{pair} + HC_{pvap} \tag{6}$$

where C_{pair} is the heat capacity of air (1.005 kJ kg^{-1} dry air °C), C_{pvap} is the heat capacity of water vapor (1.88 kJ kg^{-1} water vapor °C).

Calculation of $(P_{sat})_{T_{sat}}$

The pressure of the saturated vapor–air mixture at the adiabatic saturation temperature $(P_{sat})_{T_{sat}}$ was calculated according to Eq. (7):

$$H_{sat} = \frac{18}{29}\frac{(P_{sat})_{T_{sat}}}{760 - (P_{sat})_{T_{sat}}} \tag{7}$$

where $(P_{sat})_{T_{sat}}$ is the pressure of the saturated vapor–air mixture at adiabatic saturation temperature (mmHg) and H_{sat} is the saturation humidity that was calculated as mentioned before by using Eq. (5).

Calculation of $(T_{sat})_{cal}$

The adiabatic saturation temperature $(T_{sat})_{cal}$ was calculated according to Eq. (8):

$$\lg(P_{sat})_{T_{sat}} = 8 - \frac{1689.52}{230 + (T_{sat})_{cal}} \tag{8}$$

where $(T_{sat})_{cal}$ is the calculated adiabatic saturation temperature (°C). It should be emphasized that the use of Eq. (4) to get the value of h_{fg} is by necessity trial and error because T_{sat} is that is being sought. Hence, using an initial value of T_{sat} in Eq. (4), a value of h_{fg} is computed. Also, using this preliminary initial value of T_{sat} in Eq. (5), a value of saturation humidity H_{sat} is computed. The computed H_{sat} will be used in Eq. (7) and a value of pressure of saturated vapor–air mixture is calculated $(P_{sat})_{T_{sat}}$. The calculated $(P_{sat})_{T_{sat}}$ value will be used in Eq. (8) to get the adiabatic saturation temperature $(T_{sat})_{cal}$. Finally, the value of $(T_{sat})_{cal}$ will be compared with the assumed value of T_{sat}. If the values are different, then the value of $(T_{sat})_{cal}$ calculated from Eq. (8) will be used in Eq. (4) and the rest of the equations and then a new value of $(T_{sat})_{cal}$ is computed from Eq. (8). The trial is repeated until the values of $(T_{sat})_{cal}$ with successive trials do not change.

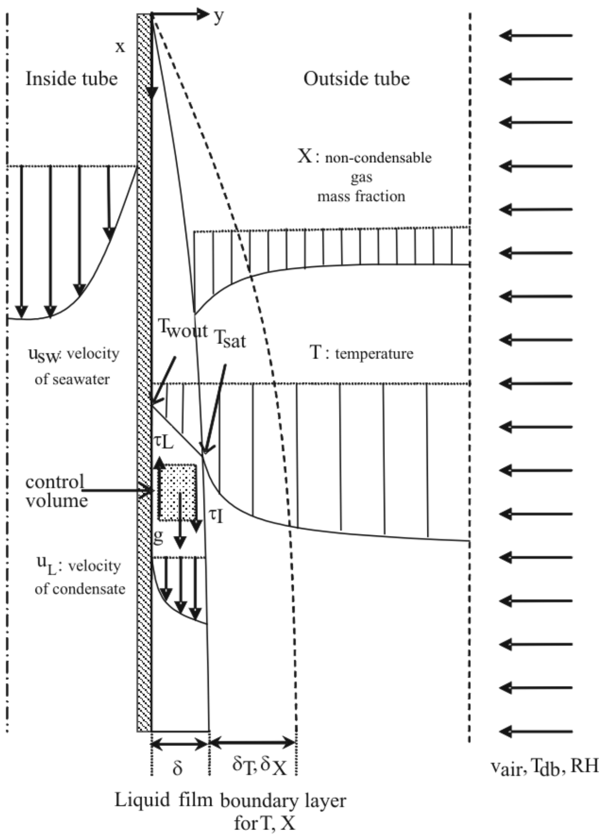

Fig. 3 Physical model of the film condensation outside vertical tube of the condenser

Step II: Determination of the average heat transfer coefficient of condensation (h_{ave}) for vertical tubes of the condenser

This section presents the procedure of calculating the average heat transfer coefficient for film condensation (h_{ave}) for laminar film condensation on the outer surfaces of vertical tubes (i.e., liquid film is assumed to be laminar for the film thickness calculation). Figure 3 shows the physical model for film condensation on a vertical tube considered in Nusselt's analysis. The outer wall temperature is below the saturation temperature ($T_{Wout}<T_{sat}$) and thus the water vapor condenses on the outer surface of the tube. The liquid film flows downward under the influence of gravity. The film thickness and thus the mass flow rate of the condensate increases with x as a result of continued condensation on the existing film [6]. Then the heat transfer from the vapor to the tube must occur through the film, which offers resistance to heat transfer. Clearly the thicker the film, the larger its thermal resistance and thus the lower the rate of heat transfer. For the liquid film, the force balance in the control volume (Fig. 3.) can be expressed as

$$\tau_L = (\rho_L - \rho) g (\delta - y) + \tau_I \tag{9}$$

where τ_L is the shear stress of liquid film (N m^{-2}), τ_I is the shear stress of liquid film in interface (N m^{-2}), ρ is the density of air (kg m^{-3}), ρ_L is the density of liquid film (kg m^{-3}), g is the gravitational constant (m s^{-2}), δ is the film thickness (m), y is wall coordinate (m).

According to Seungmin *et al.* [7], the velocity profile in the liquid laminar film (u_L) can be described from the Eq. (10):

$$u_L(y) = \frac{(\rho_L - \rho) g}{\mu_L} \left(\delta y - \frac{y^2}{2} \right) + \frac{\tau_I}{\mu_L} y \tag{10}$$

where u_L is the velocity of liquid film (m s^{-1}), μ_L is the dynamic viscosity of liquid film (kg m^{-1} s^{-1}).

The first term in the right hand side of Eq. (10) is the parabolic velocity distribution, which is exactly same with Nusselt analysis for no interfacial shear. The second term is the linear velocity distribution due to the interfacial shear. The liquid film mass flow rate (m_c) can be calculated by integrating the velocity profile equation. Then, mass balance in liquid film can be expressed with respect to film thickness as indicated in Eq. (11):

$$\Gamma \equiv \frac{m_c}{\pi D_{out}} = \left[\frac{(\rho_L - \rho) g}{3\nu_L} \delta^3 + \frac{\tau_I}{2\nu_L} \delta^2 \right] \tag{11}$$

where Γ is mass flow per unit length (kg s^{-1} m^{-1}), m_c is the liquid film mass flow (kg s^{-1}), ν_L is the kinematic viscosity of liquid film (m^2 s^{-1}), D_{out} is the outer diameter of the tube condenser (m).

From Eq. (11), the film thickness can be calculated. If there is no interfacial shear, the Nusselt thickness $\delta(x)$ [6] can be explicitly defined as

$$\delta(x) = \left[\frac{4 k_L \mu_L (T_{sat} - T_{wout}) x}{g \rho_L (\rho_L - \rho_v) h_{fg}} \right]^{\frac{1}{4}} \tag{12}$$

where T_{wout} is the temperature of outer wall of vertical tube of condenser (°C), ρ_v is the density of vapor (kg m^{-3}), x is the stream wise coordinate (m).

For laminar film, the temperature distribution in the film region is almost linear. Therefore the heat transfer coefficient for film condensation (h_x) at location x can be written as shown in Eq. (13):

$$h_x = \frac{k_L}{\delta(x)} \tag{13}$$

where k_L is the thermal conductivity of liquid water (W m^{-1} °C^{-1}), h_x is the heat transfer coefficient for film condensation at x location (W m^{-2} °C^{-1}).

Substituting the $\delta(x)$ expression from Eq. (12) into Eq. (13), the local heat transfer coefficient for film condensation h_x can be expressed as

$$h_x = \left[\frac{g \rho_L (\rho_L - \rho_v) h_{fg} k_L^3}{4 \mu_L (T_{sat} - T_{wout}) x} \right]^{\frac{1}{4}} \tag{14}$$

However, the condensate in actual condensation process is cooled further to some average temperature between T_{sat} and T_{wout}, releasing more heat in the process. Therefore, the actual heat transfer will be larger. According to Çengel [6], Rohsenow [8] showed that the cooling of the liquid below the saturation temperature can be accounted for by replacing h_{fg} by the modified latent heat of vaporization h^*_{fg}, defined as

$$h^*_{fg} = h_{fg} + 0.68 C_{pL} (T_{sat} - T_{wout}) \tag{15}$$

where h^*_{fg} is the modified latent heat of vaporization (J kg^{-1}), C_{pL} is the specific heat of liquid water (J kg^{-1}).

The average heat transfer coefficient for laminar film condensation over the vertical tube of height L is determined from its definition by substituting the h_x relation and performing the integration. It gives:

$$h_{ave} = 0.943 \left[\frac{g \rho_L (\rho_L - \rho_v) k_L^3 h^*_{fg}}{\mu_L (T_{sat} - T_{wout}) L} \right]^{\frac{1}{4}} \tag{16}$$

where h_{ave} is the average heat transfer coefficient for film condensation (W m^{-2} °C^{-1}), L is the height of vertical tube (m).

Step III: Determination of the temperature of the outer wall (T_{wout}) of each tube of the condenser

The temperature of the seawater flowing inside the tubes of the condenser is only known at the entrance and exit of the condenser. Linear assumptions were used in order to find out the temperature of seawater for each tube of the condenser by using known information that was known at the entrance and exit of the condenser. Moreover, the outside wall tube temperature (T_{wout}) need to be determined for each tube included in the condenser. For determination of T_{wout}, Eqs (17)–(29) were used.

Calculation of T_{filmin}

The properties of seawater (k_{sw}, ρ_{sw}, μ_{sw}) were calculated at inner film temperature (T_{filmin}). The film temperature in the tube of condenser (T_{filmin}) was calculated according to Eq. (17):

$$T_{filmin} = \frac{\frac{(T_{swin} + T_{swout})}{2} + T_{win}}{2} \tag{17}$$

where T_{filmin} is the film temperature inside tube of condenser (°C), T_{swin}, T_{swout} are the inlet and the outlet seawater temperatures in each tube of condenser, respectively (°C), T_{win} is the wall temperature inside tube (°C).

It should be noted that the solution procedure of Eq. (17) requires iterations as the inner wall temperature T_{win} is unknown.

Calculations of Re, Nu, and Pr

The Reynolds number (Re), Nusselt number (Nu) and Prandtl (Pr) were calculated according to Eqs (18)–(20):

$$\mathrm{Re}=\frac{\rho_{sw}u_{sw}D_{in}}{\mu_{sw}} \tag{18}$$

$$\mathrm{Nu}=\frac{h_{in}D_{in}}{k_{sw}} \tag{19}$$

$$\mathrm{Pr}=\frac{C_{psw}\mu_{sw}}{k_{sw}} \tag{20}$$

where Re is the Reynolds number, Nu is the Nusselt number, Pr is the Prandtl number, D_{in} is the inner diameter of tube (m), u_{sw} is the mean seawater velocity in the tube and it was assumed constant at each tube (m s^{-1}), ρ_{sw} is the density of seawater (kg m^{-3}), μ_{sw} is the dynamic viscosity of seawater (kg m^{-1} s^{-1}), k_{sw} is the thermal conductivity of seawater (W m^{-1} °C^{-1}), h_{in} is the heat transfer coefficient of the inner tube (W m^{-2} °C^{-1}), C_{psw} is the heat capacity of seawater (kJ kg^{-1} °C^{-1}).

For the case of Re≤2300, the Nusselt number (Nu) was obtained by Eq. (21):

$$\mathrm{Nu}=1.86\left(\mathrm{RePr}\frac{D_{in}}{L}\right)^{\frac{1}{3}}\left(\frac{\mu_{sw}}{\mu_{sww}}\right)^{0.14} \tag{21}$$

where μ_{sww} is the dynamic viscosity of seawater at inner wall temperature (kg m^{-1} s^{-1}), Pr is the Prandtl number.

For the case Re>2300, the Nusselt number (Nu) was calculated according to Eq. (22):

$$\mathrm{Nu}=0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}\left(\frac{\mu_{sw}}{\mu_{sww}}\right)^{0.14} \tag{22}$$

Calculation of h_{in}

The inner heat transfer coefficient h_{in} was calculated according to Eq. (23):

$$h_{in}=\frac{\mathrm{Nu}k_{sw}}{D_{in}} \tag{23}$$

where k_{sw} is the thermal conductivity of seawater (W m^{-1} °C^{-1}), h_{in} is the heat transfer coefficient of the inner tube (W m^{-2} °C^{-1}).

Calculation of $T_{filmout}$

The outer film temperature ($T_{filmout}$) was calculated according to Eq. (24):

$$T_{filmout}=\frac{T_{sat}+T_{wout}}{2} \tag{24}$$

Since the thickness of the tube (δ_{tub}) is very small, it is assumed that the inner (T_{win}) and the outer wall (T_{wout}) temperatures are the same. This implies that T_{wout}=T_{win}. This value of T_{wout} will be used in Eq. (24) to find $T_{filmout}$.

Calculation of h_{ave}

The average heat transfer coefficient for film condensation (h_{ave}) was calculated for each tube by applying Eq. (16) where the properties of condensate water (ρ_L, ρ_v, k_L, μ_L) were defined at $T_{filmout}$ and the modified latent heat of vaporization (h^*_{fg}) was defined at T_{sat}. In the presence of non-condensable gases (air), the average heat transfer coefficient for film condensation (h_{ave}) was corrected (h_{vert}) according to the graph of Sacadura [9] as shown in Eq. (25). It is shown that the corrected value is a function of the non-condensable gas mass fraction (X).

$$h_{vert}=h_{ave}f(X) \tag{25}$$

where h_{vert} is the corrected heat transfer coefficient for film condensation in the presence of non-condensable gas (W m^{-2} °C^{-1}), X is the non-condensable gas mass fraction (kg non-condensable gas/kg total air) which was calculated for each tube according to Eq. (26):

$$X=\frac{m'_{NC}}{m'_{tot}}=\frac{1}{1+H} \tag{26}$$

where m'_{NC} is the non-condensable gas mass (kg), m'_{tot} is the air total mass (kg).

Calculation of U

The overall heat transfer coefficient (U) was calculated according Eq. (27):

$$\frac{1}{UA_s}=\frac{1}{h_{vert}A_{in}}+\frac{1}{2\pi k_{tub}L}\ln\left(\frac{D_{out}}{D_{in}}\right)+\frac{1}{h_{in}A_{out}} \tag{27}$$

where U is the overall heat transfer coefficient for each tube (W m^{-2} °C^{-1}), k_{tub} is the tube thermal conductivity calculate at $T_{filmout}$ (W m^{-1} °C^{-1}), A_s is the surface area (m^2), A_{in} is the area of the inner surface of the wall tube (m^2), A_{out} is the area of the outer surface of the wall tube (m^2).

Calculation of Q

The heat flux (Q) was calculated for each tube according to Eq. (28):

$$Q = UA_s \frac{(T_{sat} - T_{swin}) - (T_{sat} - T_{swout})}{\ln\left(\dfrac{(T_{sat} - T_{swin})}{(T_{sat} - T_{swout})}\right)} \quad (28)$$

where Q is the heat flux for each tube (W).

Calculation of $(T_{win})_{cal}$

The inner wall temperature $(T_{win})_{cal}$ was then calculated for each tube according to Eq. (29):

$$Q = h_{in} A_{in} \frac{((T_{win})_{cal} - T_{swin}) - ((T_{win})_{cal} - T_{swout})}{\ln\left(\dfrac{((T_{win})_{cal} - T_{swin})}{((T_{win})_{cal} - T_{swout})}\right)} \quad (29)$$

The calculated $(T_{win})_{cal}$ from Eq. (29) was then compared with the assumed value of T_{win} (Eq. (17)). If the values are different, then the value of $(T_{win})_{cal}$ will be used in Eq. (17) and the rest of the equations and then a new value of $(T_{win})_{cal}$ is computed from Eq. (29). The trial is repeated until the values of $(T_{win})_{cal}$ with successive trials do not change.

Step IV: Determination of the condensate mass flux of each tube of the condenser (m_c) and the total condensate mass flux from all the tubes (m_{tot})

The heat and mass analogy method is based on the heat balance at the liquid–air interface, where the heat transferred from the humid air boundary layer is equated to the heat transferred through the condensate film. The heat transfer from the gas region is the sum of the sensible cooling of bulk mixture and latent heat of condensation [7]. At the mixture–liquid film interface, the heat flux can be expressed by Eq. (30):

$$Q = Q_{lat} + Q_{sen} = m_c h_{fg}^* + h_{sen} A_{out} (T_{sat} - T_I) \quad (30)$$

where Q_{lat} is the latent heat flux (W), Q_{sen} is the sensible heat flux (W), h_{sen} is the sensible heat transfer coefficient (W m^{-2} °C^{-1}), T_I is the interface temperature (°C), A_{out} is the external surface of tube ($\pi D_{out} L$).

The second term of Eq. (30) which is responsible about the sensible heat flux is generally very small compared to the first term (i.e., latent heat flux). Then Eq. (30) can be written as shown in Eq. (31):

$$Q = Q_{lat} = m_c h_{fg}^* \quad (31)$$

In addition, the heat flux can be defined as shown in Eq. (32):

$$Q = h_{vert} A_{out} (T_{sat} - T_{wout}) \quad (32)$$

By equating the above two equations, the mass flux of condensate for each tube can be determined as shown in Eq. (33):

$$m_c = \frac{h_{vert} A_{out} (T_{sat} - T_{wout})}{h_{fg}^*} \quad (33)$$

Calculation of m_{tot}

The mass flux of the total condensate from all tubes of the condenser (m_{tot}) was calculated according to Eq. (34):

$$m_{tot} = \sum_{i=1}^{N_{tot}} m_c \quad (34)$$

where N_{tot} is the total number of tubes of the condenser (N_{tot}=4832 tubes).

Conclusions

This paper discussed the simulation of the condenser of the seawater greenhouse at Al Hail in Muscat, Oman. In this Part 1 of the paper, a theoretical model was developed to describe the process of the condensation by using the heat and mass transfer equations. For this purpose, analyses to the equations and to the theoretical development of the proposed model were given.

Nomenclature

A	heat transfer area
C_p	specific heat
C_s	humid heat
D	diameter
g	gravitational constant
H	absolute humidity
h_{sen}	Heat transfer coefficient
h_{fg}	latent heat of vaporization
h_{fg}^*	modified latent heat of vaporization
k_{tub}	thermal conductivity
L	height
m	mass flux
m'	mass
N	tube number
P	pressure
Pr	Prandtl number
Q	heat flux
Re	Reynolds number
RH	relative humidity

S	pitch
T	temperature
U	overall heat transfer coefficient
u	axial velocity
v	velocity
y	wall coordinate
X	non-condensable gas mass fraction
x	streamwise coordinate

Greek symbols

Γ	mass flux rate per unit length
α	angle
δ	film thickness
μ	dynamic viscosity
ν	kinematic viscosity
ρ	density
τ	shear stress

Subscripts

ave	average
c	condensation
cal	calculated
cr	critical
db	dry bulb
filmin	film inside
filmout	film outside
I	interface
in	inside, inlet
L	liquid
L	longitudinal
lat	latent
NC	non-condensable gas
out	outside, outlet
sat	saturation
sen	sensible
sw	seawater
swin	seawater inlet
swout	seawater outlet
sww	seawater at wall temperature
T	transverse
tot	total
tub	tube
v	vapor
vap	vapor
vert	corrected
win	wall inner
wout	wall external

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