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Parametric Investigation of a Proposed Humidification-Dehumidification Desalination System

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ABSTRACT

A theoretical study with developed mathematical model presented and validated for a proposed desalination system based on air Humidification-Dehumidification (HDH) technique with open air open water cycle. The model was validated by comparing its results with the corresponding experimental results of other studies. Two cases of water and air are examined; case (I) - open air open water - air heating (OAOW-AH) and case (II) - open air open water -water heating (OAOW-WH). The results show that if the air mass flow rate increase above certain value that leads to decrease the condensation rate. The results also show if water mass flow rate increase that leads to increase the condensation rate.

Keywords

Desalination; Humidification - dehumidification system; System productivity; Mathematical model; Heat and mass transfer.

1. INTRODUCTION

The water is a very important requirement to the human. Freshwater consumption is increasing every day. Water is needed in all industrial activities, in agriculture and for domestic purposes [1]. Problem of fresh water shortage could be solved by several desalination techniques. Desalination refers to any process that removes some or all amount of salt and other minerals from saline water. Humidification-dehumidification (HDH) desalination technique is considered a promising technique for small capacity desalination plants. HDH is distinguished by moderate cost, simple operation and maintenance. Humidification-dehumidification process is distinguished from other processes by moderate cost, simple operation and low maintenance. It is based on evaporation of seawater or brackish water and consecutive condensation of the generated humid air, mostly at ambient pressure. Moisture is added to air in a humidification process, while the moisture is removed from air in a dehumidification process.

Nawayseh et al [2] studied experimentally a humidifier equipped with different packing material. The process is a closed air HDH cycle, in which air is circulated in the unit by natural draft between the humidifier and condenser. The mass transfer coefficient in the humidifier was found to be affected mostly by the water flow rate due to its influence on the wetting area of the packing. The results showed the effects of water and air flow rates on the productivity of desalinated water. Yamali et al [3] studied experimentally and theoretically closed water and open air system. Pad humidifier unit is consists of four pads placed in series, made of plastic and it forms the wetted surface of thermally insulated humidifier. The results indicate that significant improvement on the productivity of the system achieved by increasing the initial water temperature inside the storage tank. In addition, the productivity of the system increases with an increase of the feed water mass flow rate and the productivity remains approximately the same when the air mass flow rate is increased. Narayan et al [4] indicated that the air heated systems have higher energy consumption than water heated systems. This is because sensible heat content of water, which is supplied by air in the humidifier is not subsequently recovered from the water, unlike in the water-heated cycle in which the energy transferred to air is recovered in the dehumidifier. Xing Li et al [5] designed and tested a new solar air heater with evacuated tubes for small scale solar humidification-dehumidification desalination process. Firstly, the humidifier and dehumidifier are designed and optimized mathematically. Finally, desalination pilot plant is designed and built to test and analyse operation characteristics. Test results show that heated inlet sprayed water temperature in the pad humidifier can effectively increase both relative humidity and temperature of outlet air. These increases can effectively improve fresh water production of the plant under the same air flow rate and cooling condition. Kabeel et al [6] studied experimentally open-water and closed-air humidification-dehumidification cycles using solar water heater and using two types of cellulose paper as packing materials in a humidifier. They proposed modified design of condenser to determine unit performance. Air was circulated by natural or forced circulation. They studied the effects of different flow parameters such as, water temperature, water mass flow rate, packing material, types of air circulation and fan installation on this unit. They concluded that forced down air circulation, enhances the performance than forced up, forced up-down and natural air circulation. Forced up-down air circulation gives approximately the same system productivity as natural air circulation. Using cellulose 5 mm thickness as a packing material gives a higher productivity than using cellulose 7 mm thickness under both natural and forced flow circulation. Yildirim et al. [7] investigated experimentally an open air and open water cycle (OAOW) integrated with thermo-electric cooling (TEC) technique. The TEC were used for dehumidification process and heating of feed water. The waste heat from the hot side of TEC is used to increase the temperature of feed water. The results indicated that the system productivity is quite low because of the low value of water temperature at the inlet of the humidifier. Inlet water temperature at the entrance of the evaporator is the most important parameter that identifies the system performance. Giwa et al. [8] investigated a closed air open water humidification–dehumidification (HDH) desalination system using a recovered photovoltaic (PV) thermal energy that improves the production of small-capacity sustainable water. Air was considered to be fed to the air gap at the back of the PV through an electric fan. The air would then cool down the PV panel, become heated in the air gap and humidified by sprayed saline water in the humidifier. The air outlet from the humidifier was considered to be saturated with moisture and passed to the dehumidifier shell where it would be dehumidified by cooling it down to another saturation point (designed as 20 °C) with the aid of cooling water (CW) fed to the tubes inside the dehumidifier shell. The results show that the PV-HDH system resulted in 83.6% decrease in environmental impacts when compared with PV-Reverse Osmosis (PV-RO) system. So, the integrated PV-HDH desalination technology is promising and expected to play a key role in the field of water desalination.

Based on the literature review, there is a necessity to develop a full mathematical model to study the desalination humidification-dehumidification system. The present work is achieved to study the water desalination system based on humidification-dehumidification principle. The objective of the present study is to develop a detailed mathematical model of a HDH system for different cases; case (I) - open air open water - air heating (OAOW-AH) and case (II) - open air open water -water heating (OAOW-WH). Moreover, the effects of air and water mass flow rate on the system productivity are examined. The difference between the OAOW-AH and CAOW-AH cycles is that $T_{a,h,i}$ equals the ambient temperature, T_{amb}, instead of $T_{a,c,o.}$ In general, the ambient temperature of the air will be less than the dehumidifier exhaust temperature. As a result, the humidifier in the OAOW cycle is operating with a lower inlet temperature. The lower temperature air stream provides a greater potential for heat and mass transfer exchange with the warm seawater stream. Therefore, the humidifier can run at a higher efficiency which leads to improved cycle performance.

2. MATHEMATICAL MODEL

2.1 Description of the HDH Desalination System

This study aims to investigate the influences of operating and design parameters on the performance of the humidifier integrated with the condenser as shown in Figure 1. The air flows from bottom to top in the humidifier. During its path, it humidifies when contacts the hot water flowing down words. A packing material is placed in the humidifier to increase the contact area between the air and the water. The air exits from the dehumidifier and enters in a two identical condensers to condense the water vapor from the air. The details of the condenser will be explained later in this paper. In the application the inlet air could be heated by solar energy or any source of waste heat. Also the water could be heated primarily by passing through the condenser followed by solar heating or heating by waste heat source. Since that our study is a parametric study, these parameters will be given as input parameters to the theoretical model.

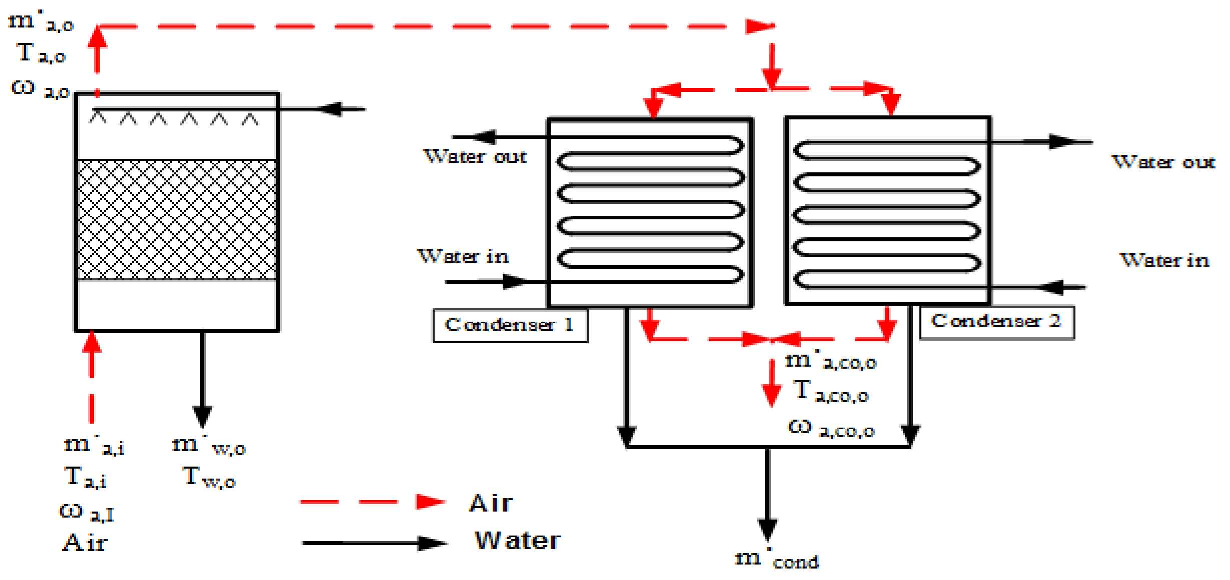

Figure. 1 Sketch of the system considered in the mathematical model

2.2 Model Assumptions

A developed simulation model of the humidifier and condenser described below. To simplify the mathematical model formulation, the following realistic assumptions have been considered:

1- Steady state conditions prevail.
2- The humidifier is well insulated; hence, the heat loss to the surrounding is neglected.
3- The system operates under constant atmospheric pressure of 101.3 KPa.
4- The packing material used in this study is wooden type.
5- Water is assumed to be uniformly distributed over the cross sectional area of the packing.
6- Heat exchanges between water and air in a counter flow process.
7- Lewis number is assumed to be unity.
8- The rate of evaporation of sea water is nearly typical to the rate of evaporation of the pure water [9]. Therefore this study was based on pure water.

2.3 Input Data

The base conditions for the model input data are shown in the following table.

Table. 1 Base case conditions

Parameter	Value	Parameter	Value
$\dot{m}_a$ (kg/s)	0.02	(ω_{amb}) (kg_v / kg_a)	0.01331
$\dot{m}_w$ (kg/s)	0.08	T_a (°C)	30
T_{amb} (°C)	30	T_w (°C)	30

The design parameters of the considered system are taken close to the values reported in [2] and shown in the following table.

Table. 2 Values of Design Data

Parameter	Value	Parameter	Value
$\dot{m}_a$ (kg/s)	0.01-0.04	Humidifier height (m)	2
$\dot{m}_w$ (kg/s)	0.04-0.16	Humidifier width (m)	0.5
T_{amb} (°C)	30	Humidifier depth (m)	0.8
(ω_{amb}) (kg_v / kg_a)	0.01331	Condenser height (m)	2
T_a (°C)	40-80	Number of condensers	2
T_w (°C)	50:90	$D_{i,co,w,tube}$ (m)	0.009
$\dot{m}_{w,co}$ (kg/s) (constant)	0.08	$D_{o,co,w,tube}$ (m)	0.011
$T_{w,co}$ (°C) constant	30	Fins height in condenser (m)	0.05

2.4 Humidifier Modelling

Wooden packing material used in this study was described and suggested by Nawayseh [2]. The humidifier was divided to ten control volumes (Figure 2a). The parameters at exit from each control volume is the inlet parameters for the next control volume. Mass and energy balances for both air and water passing through each control volume are presented in Figure.2b.

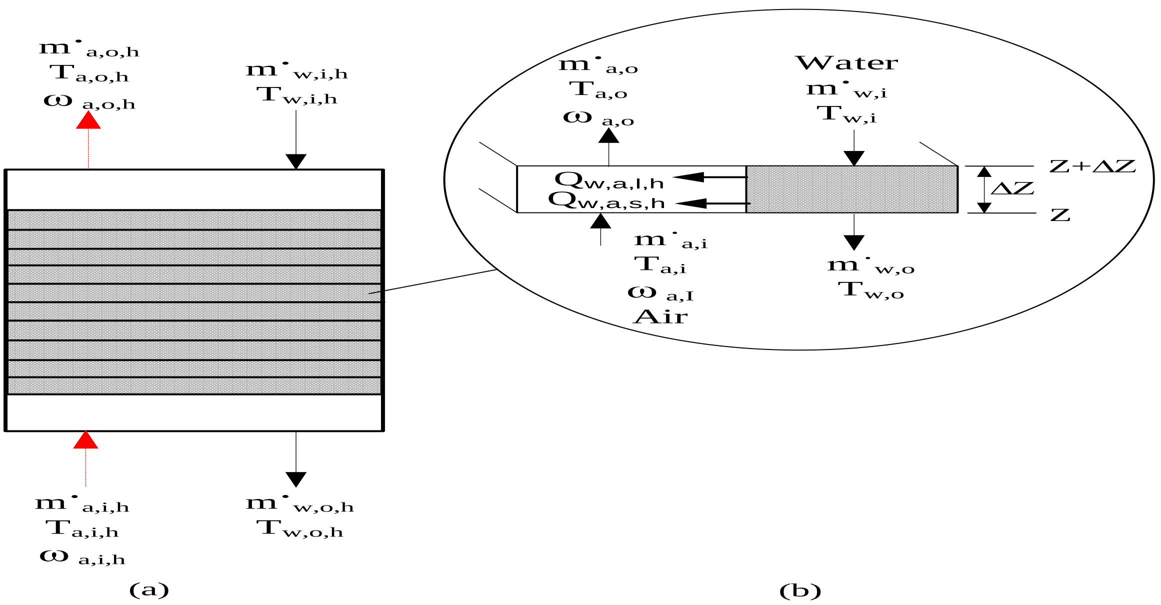

Figure. 2 (a) Energy and mass balances (b) An element of the humidifier

2.4.1 Mass balance for humid air

The mass of air changes due to the amount of vapor ($\dot{m}_{evap}$) that gets suspended in it as it flows inside the humidifier. Since the rate of dry air is constant, the air mass balance is expressed as:

$$\dot{m}_{evap} = \dot{m}_{a,i}.(\omega_{a,h.o} - \omega_{a,h.i}) \quad (1)$$

$$\dot{m}_{evap} = k_{av}.v_{p,el}.(\omega_{s,h.m} - \omega_{a,h.m}) \quad (2)$$

Where $\omega_{a,h.m} = \frac{(\omega_{a,h.i}+\omega_{a,h.o})}{2}$ and $\omega_{s,h.m} = \frac{(\omega_{s,h.i}+\omega_{s,h.o})}{2}$

Where $k_{av} = k.a_v$ is the mass transfer coefficient related to the area per unit volume (a_v)which can be calculated by the following equation [2] :

$$\frac{k_{av}.V}{\dot{m}_{w,i}} = 1.19\,.\left[\frac{\dot{m}_{w,i}}{\dot{m}_{a,i}}\right]^{-0.66} \quad (3)$$

The heat transfer coefficient is calculated using the following equation base on the Lewis relationship as defined by Donald [10]:

$$k_{av} = \frac{h_{av}}{Cp_a} \quad (4)$$

Where $h_{av} = h.a_v$ is the heat transfer coefficient related to the area per unit volume.

2.4.2 Mass balance for water

The mass of water also decreases due to the amount of vapor ($\dot{m}_{evap}$) that spreads into the air. The water mass balance is expressed as:

$$\Delta\dot{m}_w = \dot{m}_{w,i} - \dot{m}_{w,o} \quad (5)$$

$$\Delta\dot{m}_w = \dot{m}_{evap} \quad (6)$$

Using Equation (2) and Equation (6), the following equation can be deduced:

$$\Delta\dot{m}_w = k_{av}.v_{p,el}.(\omega_{s,h.m} - \omega_{a,h.m}) \quad (7)$$

2.4.3 Energy balance for humid air

Sensible heat energies of water and air can be computed using the following equations:

$$Q_{w,a,s.h} = \dot{m}_{a,i}.Cp_{a,i}.(T_{a,h.o} - T_{a,h.i}) \quad (8)$$

$$Q_{w,a,s.h} = h_{av}. v_{p,el} . (T_{w,h,m} - T_{a,h.m}) \quad (9)$$

where $T_{a,h,m} = \frac{T_{a,h,in} + T_{a,h,o}}{2}$ and $T_{w,h,m} = \frac{T_{w,h,in} + T_{w,h,o}}{2}$

2.4.4 Energy balance for water

Latent heat energy can be computed using the following equation:

$$Q_{l.h} = \dot{m}_{evap}. H_{fg} \quad (10)$$

From Equation (2) and Equation (10), the following equation can be deduced:

$$Q_{l.h} = k_{av}. v_{p,el} . (\omega_{s,h.m} - \omega_{a,h.m}). H_{fg} \quad (11)$$

Total rate of heat transfer from air to water can be computed by the following equations:

$$Q_{to,w,a} = Q_{s.h} + Q_{l.h} \quad (12)$$

$$Q_{to,w,a} = \dot{m}_{w,i}. Cp_{w} . T_{w,h.i} - \dot{m}_{w,o}. Cp_{w} . T_{w,h.o} \quad (13)$$

2.5 Condenser

The condenser used in this model was suggested by [2] using galvanized steel sheets 1.0 mm thickness. The condenser cylinder is 3 m long and 0.170 m diameter. Ten longitudinal fins were soldered on the cylinder outer surface and similarly nine 1.0 mm-thickness fin were soldered on the inner surface with height 50 mm for the inside and outside. Copper tube with 16.02 m long having 9 mm inside diameter was soldered to the surface of the cylinder as shown in Figure.3. Water was used as cooling medium inside Copper tube. The condenser section is separated to two similar condensers.

2.5.1 Mass balance

The condensation rate of water can be computed by:

$$\dot{m}_{cond} = \dot{m}_{a,co,in}. (\omega_{a,co.in} - \omega_{a,co,o}) \quad (14)$$

$$\dot{m}_{cond} = k_{m,a,co}. \eta_{co.fin}. (A_{co,cyl} + A_{co,fin}). (\omega_{a,co.} - \omega_{a,s.co}) \quad (15)$$

Where $\omega_{a,co} = \frac{(\omega_{a,co.i} + \omega_{a,co.o})}{2}$ and $\omega_{a,s,co} = \frac{(\omega_{s,co.i} + \omega_{s,co.o})}{2}$

With neglecting the outer surface area of the water tube.

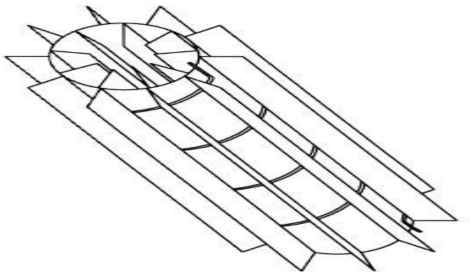

Figure. 3 Geometrical description of the condenser

2.5.2 Energy balance for air

The energy balance in the condenser is described by the following equations:

Sensible heat energy can be computed by:

$$Q_{a,w.s.co} = \dot{m}_{a,co,in}.\left(Cp_{a,co,in} + \omega_{a,co.in}.Cp_v\right).T_{a,co,in} - \dot{m}_{a,co,in}.\left(Cp_{a,co,o} + \omega_{a,co.o}.Cp_v\right).T_{a,co,o} \quad (16)$$

$$Q_{a,w.s.co} = h_{conv,co,a}.\eta_{co.fin}.\left(A_{co,cyl}+A_{co,fin}\right).(T_{a,co} - T_{wall,co}) \quad (17)$$

Where $T_{a,co} = \frac{T_{a,co,in}+T_{a,co,o}}{2}$ and $T_{w,co} = \frac{T_{w,co,in}+T_{w,co,o}}{2}$

Latent heat energy can be computed by:

$$Q_{a,w,l.co} = \dot{m}_{cond}.H_{fg,co} \quad (18)$$

Total rate of heat transfer from air to water can be computed by:

$$Q_{a,w,to.co} = Q_{a,w,s.co} + Q_{a,w,l.co} \quad (19)$$

The following equations describe the air side balance in the condenser using reference [11]:

The convection heat transfer coefficient of air in the condenser ($h_{conv,co,a}$) can be computed by:

$$Nu_{a,co,duct} = \frac{h_{conv,co,a}.D_{eq,co}}{k_{a,co}} \quad (20)$$

Where the equivalent hydraulic diameter ($D_{eq,co}$).can be computed by:

$$D_{eq,co} = \frac{4.A_{co,cs}}{2.\pi.(D_{co,pvc,tube}+D_{co,cyl})+2.N_{co,o,fin}.H_{co,fin}+2.N_{co,i,fin}.H_{co,fin}} \quad (21)$$

For a laminar flow if ($Re_{a,co} \leq 2300$), Nusselt number for the humidified air, which enters the condenser can be computed by the following equation:

$$Nu_{a,co,duct} = 3.66 + \frac{0.065.\frac{D_{eq,co}}{L_{co,cyl}}.Re_{a,co}.Pr_{a,co}}{1+0.04.(\frac{D_{eq,co}}{L_{co,cyl}}.Re_{a,co}.Pr_{a,co})^{\frac{2}{3}}} \quad (22)$$

For turbulent flow If ($Re_{a,co} > 2300$), Nusselt number can be computed by:

$$Nu_{a,co,duct} = \frac{\frac{f}{8}.(Re_{a,co}-1000).Pr_{a,co}}{1+12.7.(\frac{f}{8})^{0.5}.((Pr_{a,co})^{\frac{2}{3}}-1)} \quad (23)$$

Where the friction factor (f) can be computed by:

$$\frac{1}{\sqrt{f}} = -2.\log\left(\frac{2.51}{Re_{a,co}.\sqrt{f}} + \frac{\mathcal{E}_{cu}}{3.7\,.D_{eq,co}}\right) \quad (24)$$

Reynolds number as well as Prandtl number of the humidified air entering the condenser section can be computed by the following equations:

$$Re_{a,co} = \frac{\rho_{a,co}.V_{a,co}.D_{eq,co}}{\mu_{a,co,i}} \quad (25)$$

$$Pr_{a,co} = \frac{\mu_{a,co}.Cp_{a,co,i}}{k_{a,co}} \quad (26)$$

The convection mass transfer coefficient of air in condenser $k_{m,a,co}$can be computed by:

$$k_{m,a,co} = \frac{h_{conv,co,a}}{Cp_{a,co,in}+\omega_{a,co.in}.Cp_{v}} \quad (27)$$

2.5.3 Energy balance for water

$$Q_{a,w,to.co} = h_{conv,co,w}.A_t.(T_{wall,co} - T_{w,co}) \quad (28)$$

$$Q_{a,w,to.co} = \dot{m}_{w,co,in}.Cp_{w,co,in}.\left(T_{w,co,o} - T_{w,co,in}\right) \quad (29)$$

The convection heat transfer coefficient of water side in the condenser ($h_{conv,co,w}$)can be computed by:

$$Nu_{w,co} = \frac{h_{conv,co,w}.D_{i,co,w,tube}}{k_{w,co}} \quad (30)$$

Nusselt, Reynolds, and Prandtl numbers of water side in the condenser can be computed by the following equations, respectively:

$$Nu_{w,co} = 0.023.(Re_{w,co})^{0.8}.(Pr_{w,co})^{0.3} \quad (31)$$

$$Re_{w,co} = \frac{\rho_{w,co}.V_{w,co}.D_{i,co,w,tube}}{\mu_{w,co,in}} \quad (32)$$

$$Pr_{w,co} = \frac{\mu_{w,co,in}.Cp_{w,co,in}}{k_{w,co,in}} \quad (33)$$

The efficiency of the fin of the condenser can be computed by:

$$\eta_{co.fin} = \frac{\tanh(H_{co,fin}.\sqrt{2.\frac{h_{conv,co,a}}{k_{iron}.th_{fin}}})}{(H_{co,fin}.\sqrt{2.\frac{h_{conv,co,a}}{k_{iron}.th_{fin}}})} \quad (34)$$

The equations 1-34 were solved using Engineering Equation Solver (EES) program.

3. Mathematical Model Validation

Figure.4 shows the temperatures of water and air at the inlet and exit of the humidifier versus time from 10 AM to 18 PM [2]. Since the model of the present study is steady state model, the inlet temperatures of water and air to the humidifier of [2] were considered constant during the hour and it were given as an input parameters to the humidifier of the present study. Then, the outlet temperatures from the humidifier of the present study were compared with the experimental data of [2]. The comparison shows good agreement as shown in Figure.4, which gives confidence in extending model results. The model given by [2] was based on overall balance of the humidifier but in the present study, the heat and mass balances were based on dividing the humidifier into ten control volumes from bottom to top. The present study used open air open water configuration with ability of heating both air and water instead of closed air open water configuration with heating water which used by [2].The approach of model differ from the approach which was given by [2].

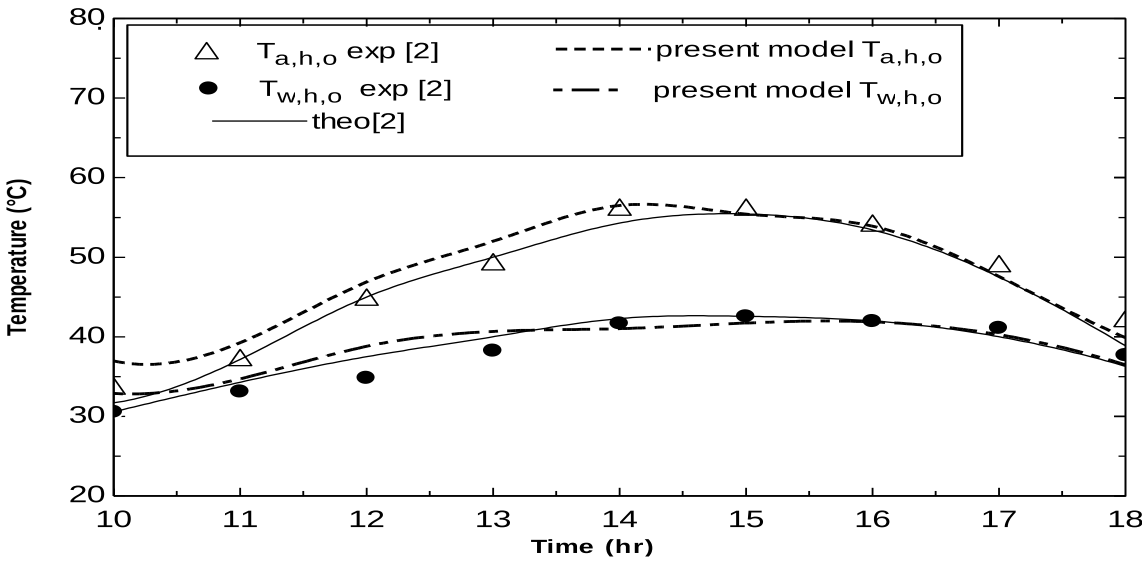

Figure. 4 Validation of the mathematical model

4. RESULTS AND DISCUSSION

Effects of varying air and water mass flow rates, individually, on system performance were investigated. Two cases of water and air are examined; case (I) - open air open water - air heating (OAOW-AH) and case (II) - open air open water -water heating (OAOW-WH).

4.1 The Effect Of Air Mass Flow Rate

The effect of air mass flow rate on the condensation rate (system productivity) for two cases is illustrated in **Figures. 5** and **6.** The air mass flow rate is varied within the range 0.02:0.08 kg/s for different water mass flow rate (0.04:0.1). As shown in Figure. (5) for case (I) (OAOW-AH) (T_{a-h-in} =60 °c – T_{w-h-in} =30 °c). The figures shows that the productivity decreases with increasing the air flow rate more than a certain value. This can be attributed to the increase in air velocity which leads to decreasing the contact time between air and the surface of the condenser. The figure also indicates that to increase the condensation rate and consequently to increase the productivity, the system should operate at high water flow rate and at low air flow rates. Also the figure shows that the effect of increase water flow rate is more profound at low values.

As shown in Figure. (6) for case (II) (OAOW-WH) (T_{a-h-in} =30 °c – T_{w-h-in} =60 °c). The figure shows that the productivity decreases with increasing the air flow rate. The figure also indicates that to increase the condensation rate and consequently to increase the productivity, the system should operate at high water flow rate and at low air flow rates. Also the figure shows that the effect of increase water flow rate is more profound at low values. Productivity more than case one could be obtained in this case as the water heating increasing the driving force in the humidifier which leads to increase evaporation rate and consequently to increase the condensation rate.

4.2 The Effect Of Water Mass Flow Rate

The effect of water mass flow rate on the condensation rate (system productivity) is studied as illustrated in **Figures. 7** and **8**. The water mass flow rate is varied within a range from 0.04 to 0.16 kg/s at an air mass flow rate. It is observed from **Figure. 7** for case (I) that increasing the water mass flow rate increases the production of fresh water that is attributed to the increase in the evaporation rate and consequently, more vapor condenses in the condenser. At low value of air flow rate (0.01 kg/s), condensation rate mostly independent of water flow rate since the productivity is nearly asymptotic to a constant value. At high values, as water flow rate increase the productivity increase.

The effect of water mass flow rate on the system productivity for case (II) is presented in **Figure. 8.** At low value of air flow rate, the condensation rate mostly independent of water flow rate similar to the case (I). Moreover, increasing the water mass flow rate rather than 0.08 kg /s increases the system productivity for any value of air mass flow rate. The system productivity in case (II) is more than case (I) as the water heating increases the driving force in the humidifier which leads to increase evaporation rate and consequently to increase the system productivity.

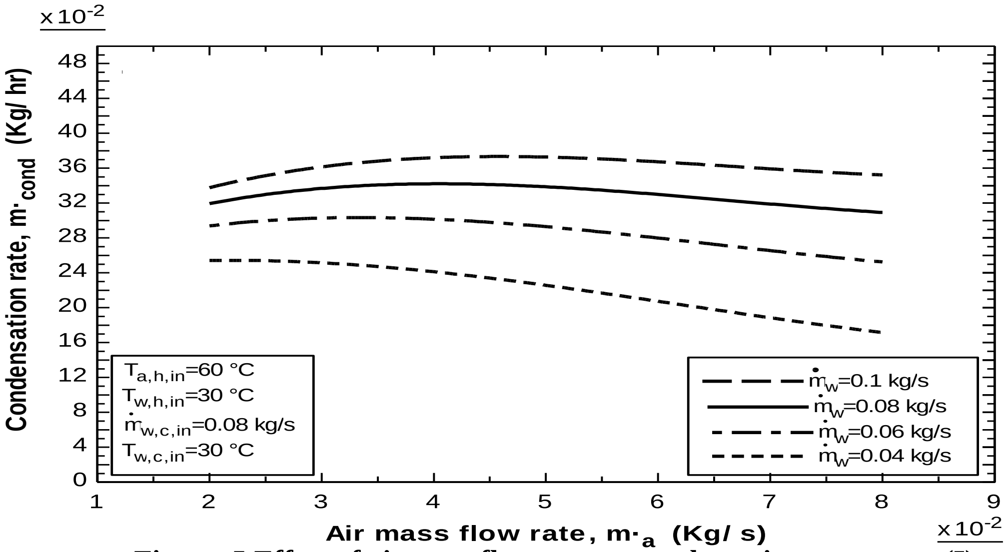

Figure. 5 Effect of air mass flow rate on condensation rate - case (I)

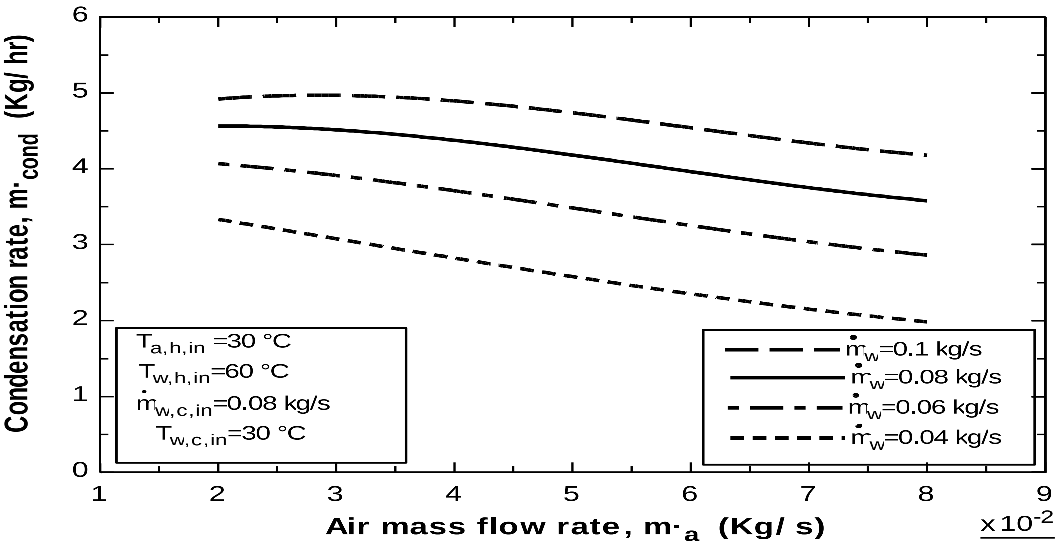

Figure. 6 Effect of air mass flow rate on condensation rate - case (II)

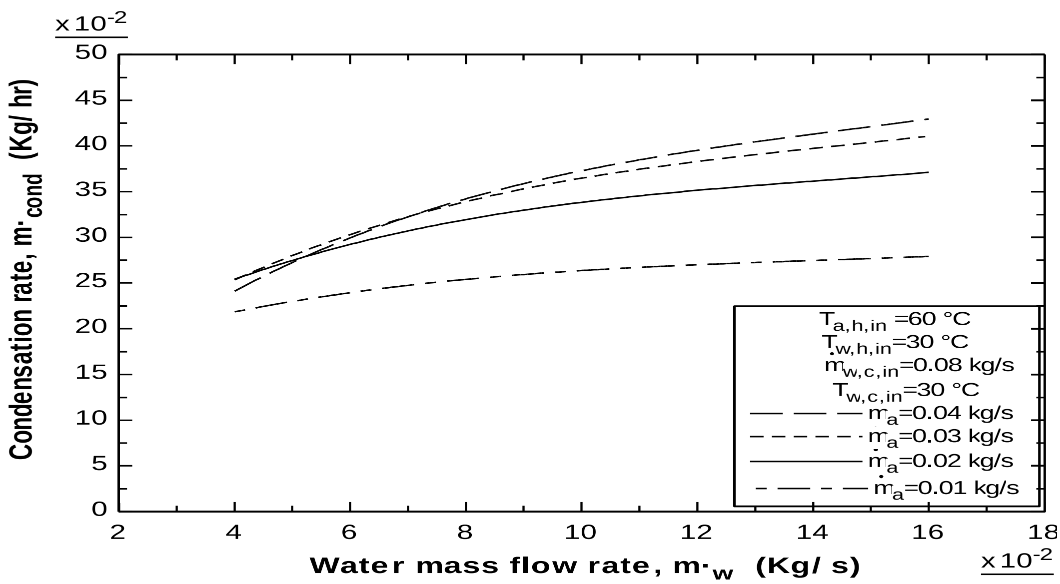

Figure. 7 Effect of water mass flow rate on condensation rate - case (I)

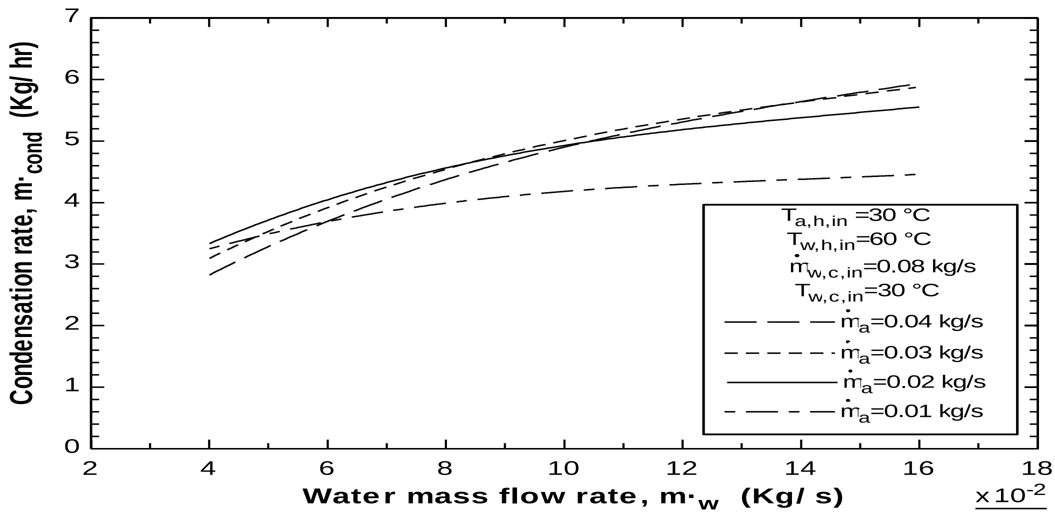

Figure. 8 Effect of water mass flow rate on condensation rate - case (II)

5. CONCLUSION

The results of the present study of typical modelling operation and design parameters are presented and analysed in this study. Effects of these parameters were investigated to evaluate their influences on the performance of the proposed HDH system. The present developed mathematical model has been validated by comparing the present model results with the results reported by [2]. The results for two cases show that increasing air mass flow rate results in decreasing condensation rate. However, increasing water mass flow rate leads to an increase in condensation rate. The effect of increasing water mass flow rate on the productivity is preferable than increasing air mass flow rate through the system. The system productivity in case (II) is more than case (I) as the water heating increases the driving force in the humidifier which leads to increase evaporation rate and consequently to increase the system productivity.

6. NOMENCLATURE

A	surface area (m^2)
a_v	humidifier surface area per unit volume (m^2/m^3)
Cp	specific heat capacity (J/kg K)
D_{eq}	equivalent diameter (m)
f	friction factor
H	height (m)
H_{fg}	latent heat of water (kJ/kg)
h	convective heat transfer coefficient (W/m^2 K)
h_{av}	volumetric heat transfer coefficient (W/m3 K)
K	mass transfer coefficient (kg/ m^2 s)
K_{av}	volumetric mass transfer coefficient in humidifier (kg/m^3s)
Km_{co}	mass transfer coefficient in condenser section (kg/ m^2 s)
k	thermal conductivity (W/m K)
L	Length (m)
$\dot{m}$	mass flow rate (kg/s)
Q	heating power (Watt)
RH	Relative humidity (%)
T	temperature (°C)
th	thickness (m)
V	volume of humidifier (m^3)

Non-dimensional number

Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

Greek letters

ρ	density (kg/m^3)
Δ	change
μ	dynamic viscosity (kg/s.m)
η	efficiency
ε	Roughness(-)
ω	specific humidity($kg_v/kg_{dry\ air}$)

Subscript

a	air
amb	ambient
co	condenser
col	collector
cond	condensate
conv	convection
cu	Copper
cyl	cylinder condenser
el	element
evap	evaporation
fin	fins
h	humidifier
i	in
lat	latent
o	out
p	packing
s	sensible or saturation
t	tube
to	total
v	vapour
w	water

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الملخص

تعنى تحلية المياه إلى العديد من العمليات التي تعمل على إزالة بعض كميات من الأملاح والمعادن الأخرى من المياه المالحة. تقنية التحلية بترطيب الهواء وازالة الرطوبة منه تعتبر احدى التقنيات الواعدة فى هذا المجال .حيث انها عملية بسيطة و رخيصة التكاليف وصيانتها بسيطة. وهى تعتمد على تبخير مياه البحر فى الجزء المسمى المرطب واعادة تكثيفه فى المكثف لازالة الاملاح والشوائب من الماء.

هذه دراسة نظرية على نظام تحلية مقترح قائم على تقنية الترطيب،وازالة الرطوبة من الهواء عبارة عن دورة مفتوحة لكل من الماء والهواء . تم استخدام حشو خشبى ومكثف محددا المواصفات و تم دراسة حالتين فى هذا البحث. الحالة الاولى هى دورة مفتوحة لكلا من الماء والهواء مع تسخين الهواء فقط .والحالة الثانية هى دورة مفتوحة لكل من الماء والهواء مع تسخين الماء فقط.

وأظهرت النتائج أن إنتاجية النظام فى كلا الحالتين تقل بزيادة كمية الهواء الداخل للنظام وأن إنتاجية النظام تزيد بزيادة كمية الماء الداخلة للمرطب. وأن إنتاجية النظام تزيد فى الحالة الثانية (حالة تسخين الماء) عن الحالة الاولى (تسخين الهواء) وذلك بسبب أن تسخين الماء يعمل على زيادة القوة الدافعة فى المرطب والتى تؤدى الى زيادة معدل التبخير وبالتالى تؤدى الى زيادة معدل التكثيف وإنتاجية النظام .