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DYNAMIC MODELING AND SIMULATION OF A MULTI EFFECT HUMIDIFICATION-DEHUMIDIFICATION DESALINATION UNIT

Mohammed Imroz Sohel[1] and Belal Dawoud [2]

[1]Ananda Group, City Heart Building, 67 Naya Paltan,
Dhaka 1000, Bangladesh.
Email: sohelimroz@yahoo.com
[2]Chair of Technical Thermodynamic, RWTH Aachen University,
Schinkelstrasse 8, D-52062 Aachen, Germany.

ABSTRACT
Recently solar distillation using Multi Effect Humidification-dehumidification (MEH) cycle is receiving special attention. In this communication detailed models of MEH are presented that take into account most of the dynamic behavior with reasonable accuracy. Using these models simulation was carried out and the results of simulation were discussed in this paper. To find out the competence of the newly developed models, some of the simulation results were compared with the available experimental data in the literature. It turned out that the result of simulation give a very good agreement with the experimental data. The maximum deviation between simulation and experimental results was found to be less than 5%.

Keywords: Solar distillation, MEH, Solar energy.

1. INTRODUCTION
Solar energy employed to seawater desalination is a topic of old interest. A comprehensive study of seawater desalination using humidification and dehumidification is available in [1]. According to Graeter at el. [2], efficient evaporation and condensation can be achieved at high temperature (close to 100°C); however, the thermal efficiency even for the highest quality flat plate collector drops significantly at such elevated temperature. At moderate operating temperatures, intensive heat and mass transfer must be maintained in the evaporator and condenser. This necessitates the development of a new generation of solar desalination units. Grune [3] introduced the multi-effect humidity process in the early 1960s and his 1970 study gives interesting insight into the process. The most attractive feature of this technology is the recovery of heat of condensation. As high as 70% heat recovery is reported in the literature.

An investigation of various parameters used in such process is presented in [4], a computer simulation and experimental data are available in [5]. These investigations concentrate mostly on steady state operation. The dynamic models used in these investigations suffer much from detailed information as well as are not competent enough to take into account complicated phenomenon such as heating up of the MEH unit. However, references [4] and [5] are the basis for most of the heat and mass transfer phenomenon that takes place in the MEH process and are used in this investigation.

2. PRINCIPLE OF MEH
Because of ease of operation and maintenance MEH Units based on Open-Water/Closed-Air Cycle will be discussed in this paper. In this type of plants, heat is recovered by air circulation between a humidifier and a condenser using natural or forced draft circulation. As shown in figure 1, the saline water feed fed to the condenser is preheated by the evolved latent heat of condensation of water. This heat is usually lost in the single-basin still. The saline water leaving the condenser is further heated in a flat plate solar collector and then sprayed over the packing in the humidifier. The reported efficiency of these desalination units was significantly higher than that of a single basin still. These types of desalination units are very suitable for small production capacities in remote areas.

3. CONSTRUCTION OF AN MEH UNIT BASED ON OPEN-WATER/CLOSED-AIR CYCLE
Only a selected design by Nawayseh et al. [4], constructed in Malaysia, is presented here. The reason is that production rate of desalinated water by this unit is much higher than that of others in the same paper [4]. Detailed dimensions of this unit are presented later in this work.

3.1 Comprehensive Study of The Humidifier
The humidifier is nothing but a cooling tower of counter flow type. Water is dropped from the top of the tower whereas air is flown through the bottom of this tower.

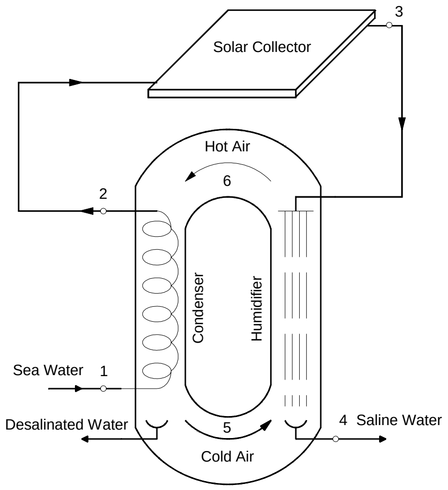

Fig 1. Schematic diagram of an experimental MEH desalination unit operated with forced or natural air circulation [1]

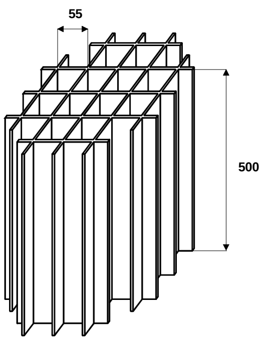

Fig 2. Details of humidifier's packing used in the desalination units in Malaysia [4]

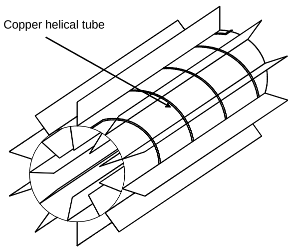

Fig 3. Schematic diagram of the Condenser used in the desalination units in Malaysia [4]

Hot water is cooled down by direct contact with air. Total six numbers of such humidifiers were used in this particular design.

It is common practice in the design of cooling towers [6] to use the performance characteristic, Ka/L, defined as follows

$$KaV/L = \int_{T4}^{T3} \frac{cp_w dT}{H_f - H_a} \quad (1)$$

The mass transfer coefficient is calculated based on Lewis relationship as defined by Donald [6]:

$$K = \frac{h_a}{cp_a} \quad (2)$$

h_a is calculated by the following relation [7]:

$$Nu_L = 1.75(\frac{\mu_b}{\mu_w})^{0.14}[Gz + 0.012(GzGr^{1/3})^{4/3}]^{1/3} \quad (3)$$

Where, $h_a = \frac{k}{d} Nu_L$ (4)

Grashof number is defined as,

$$Gr = \frac{\rho^2 g\beta(T_w - T_b)d^3}{\mu^2} \quad (5)$$

Graetz number is defined as:

$$Gz = \mathrm{Re}\,\mathrm{Pr}\frac{d}{L} \quad (6)$$

Reynold's number is defined as:

$$\mathrm{Re}_d = \frac{\rho v d}{\mu} \quad (7)$$

Prandtl number, Pr, is a physical property. All of the physical properties of air and water that have been used in this work were taken from [8].

The characteristic dimension used in the Re and Nu is the hydraulic diameter of the flow area enclosed by the adjacent wooden slats. The distance between the wooden slats was used as the characteristic dimension in the Grashof number.

3.1.1 Mass Balance of the Humidifier

$$\{\dot{m}_{a,dry}\}_{in} = \{\dot{m}_{a,dry}\}_{out} \quad \text{and}$$

$$\{\dot{m}_w + w\dot{m}_{a,dry}\}_{in} = \{\dot{m}_w + w\dot{m}_{a,dry}\}_{out} \quad (8)$$

3.1.2 Energy Balance of the Humidifier

Energy balance of water as a thermodynamic system

$$\dot{Q}_w = m_w cp_w \frac{dT_w^{outlet}}{dt} + \dot{m}_w cp_w (T_w^{outlet} - T_w^{inlet}) \quad (9)$$

According to energy conservation, energy given by water is taken by both wood and air,

$$\dot{Q}_w = \dot{Q}_a + \dot{Q}_{wood} \quad (10)$$

Energy balance of air is given by

$$\dot{Q}_a = m_a \frac{du_a}{dt} + \dot{m}_{a,dry}(h_{1+x}^{outlet} - h_{1+x}^{inlet}) \quad (11)$$

Here ideal homogeneous mixing is assumed. u_a represents the internal energy of air and calculated by following equation:

$$u_a = (h_{1+x} - pv) \quad (12)$$

Since the amount of air hold up is very small, the first term of equation 11 can be neglected. Thus equation 11 can be reduced to:

$$\dot{Q}_a = \dot{m}_{a,dry}(h_{1+x}^{outlet} - h_{1+x}^{inlet}) \quad (13)$$

Enthalpy of wet air as well as basic equations of thermodynamic was taken from [9].

Heat taken by air is the result of combined heat and mass transfer from the water to air:

$$\dot{Q}_a = KaV((h_w^{outlet} + h_w^{inlet})/2 - (h_{1+x}^{outlet} + h_{1+x}^{inlet})/2) \quad (14)$$

Energy balance in the wood:

$$\dot{Q}_{wood} = m_{wood} cp_{wood} \frac{dT_{wood}}{dt} \quad (15)$$

This energy is given by the water. Thus

$$\dot{Q}_{wood} = h_w A\left(\frac{T_w^{inlet} + T_w^{outlet}}{2} - T_{wood}\right) \quad (16)$$

In the above equation it was assumed that there is no mass transfer from water to wood. In reality mass transfer takes place. But mass transfer from water to wood is limited in the first start up phase and after that the significance is ignorable. Thus for normal operation the mass transfer from water to wood can be neglected.

Water side convective heat transfer coefficient can be calculated using the following equation [7]:

$$Nu_L = \Pr^{1/3}(0.37\,\mathrm{Re}_L^{0.8} - 871) \quad (17)$$

Where, $10^7 > \mathrm{Re} > 5\times10^5$

3.1.3 Calculation of Hold up Mass of Water in the Humidifier

The amount of water hold up in the humidifier is dependent on the film thickness developed on the surface of the humidifier. Let's assume the thickness to be δ. The humidifier is made of wooden slats combined in cross arrangement. It has surface area of A. Now, if it is assumed a single plate with equivalent area has a width of b. Then the width can be calculated as:

$$b = A/L \quad (18)$$

Where, L is the height of the humidifier.

The flow cross sectional area of the flow at any height in the humidifier can be defined as $b\delta$.

Now let's define flow velocity of water at the entry point of humidifier to be v_1 and the flow velocity of water at the exit point (at the bottom of the humidifier) to be v_2.

Using Bernoulli equation [10] between the entry and exit points of the humidifier one can write,

$$\frac{v_1^2}{2g} + L = \frac{v_2^2}{2g} + \frac{CAv_{av}^2}{2g} \quad (19)$$

Where, C is the drag coefficient, it is taken to be 0.005 in this work. Equation 19 can be solved for v_2 as follows

$$v_2 = \sqrt{v_1 + 2gL - CAv_{av}^2} \quad (20)$$

For calculation of drag loss instead of point-to-point velocity, only average velocity was used, difference between velocities at the entry to the humidifier to that of at the exit is significant. Thus the use of average velocity is not a good assumption. But the model was solved using discretization technique. Here, the whole humidifier is divided hypothetically into several elements across the height. All of the phenomenon described earlier, were calculated for one element and then the output from this element used as the input of the next element. Subsequently, the whole humidifier is simulated. Discretization was carried out for the whole MEH unit.

There is an ignorable density difference of water in the humidifier. According to Continuity equation [10]:

$$A_1 v_1 = A_2 v_2 \quad (21)$$

Inserting values of A_1 and A_2 in equation 21 one gets:

$$b\delta_1 v_1 = b\delta_2 v_2 \quad (22)$$

From equation 22 film thickness at various locations along the height of the humidifier can be obtained:

$$\delta_2 = \frac{\delta_1 v_1}{v_2} \quad (23)$$

Let's assume δ_{av} is the average film thickness in the humidifier which is the arithmetic mean of δ_1 and δ_2. Total volume of the hold up water is:

$$V_w = A\delta_{av} \quad (24)$$

Therefore the mass of water hold up in the humidifier is given by

$$m_w = V_w \rho_w \quad (25)$$

Detail analysis hold up mass in the humidifier is presented in [11].

3.2 Comprehensive Study of the Dehumidifier

The condenser is the most sensitive component of the whole unit. The amount of condensation is very much dependent on the performance of the condenser. In order to utilize the latent heat of condensation of water efficiently, the condenser area was made large by incorporating fin to it.

3.2.1 Necessary Mass Balance in the Condenser

$$\{w\dot{m}_{a,dry}\}_{in} = \{w\dot{m}_{a,dry} + \dot{m}_{con}\}_{out} \quad (26)$$

3.2.2 Energy balance of the condenser

Heat gain of water while flowing through the tube is given by,

$$\dot{Q}_w = m_w cp_w \frac{dT_w^{outlet}}{dt} + \dot{m}_w cp_w (T_w^{outlet} - T_w^{inlet}) \quad (27)$$

The quantity of this heat can be calculated as follows:

$$\dot{Q}_w = U_{cond} A_{cond} \Delta T_{mean} \quad (28)$$

$$\Delta T_{mean} = ((T_a^{inlet} + T_a^{outlet})/2 \quad - (T_w^{inlet} + T_w^{outlet})/2) \quad (29)$$

It is a general practice to use the log mean temperature difference while modeling heat exchanger. Since the simulation will be conducted using discretization, arithmetic mean temperature will give result with good accuracy.

The amount of heat given out by air can be calculated as:

$$-\dot{Q}_a = m_a cp_a \frac{dT_a^{outlet}}{dt} + m_s cp_s \frac{dT_a^{outlet}}{dt}$$

$$+\dot{m}_a (h_a^{outlet} - h_a^{inlet}) + m_{con} cp_w T_a^{outlet} \quad (30)$$

Here also ideal homogeneous mixing is assumed for heat transfer phenomenon. Reference temperature is assumed to be 0ºC. Now using energy conservation law one can write:

$$\dot{Q}_w = \dot{Q}_a \quad (31)$$

3.2.3 Calculation of Overall Heat Transfer Coefficient in the Condenser

In the condenser the heat flows from the humid air to the cooling water either through the fins or through the cylinder itself. The following two resistances may represent these two heat flow paths:

$$R_1 = \frac{1}{h_w A_t} + \frac{1}{\eta_f A_f h_c} \quad (32),$$

$$R_2 = \frac{1}{h_w A_t} + \frac{1}{\eta_f A_{cyl} h_c} \quad (33)$$

The overall heat transfer resistance may be defined as follows, assuming parallel resistance:

$$U_{cond} A_{cond} = \frac{1}{R_1} + \frac{1}{R_2} \quad (34)$$

The fin efficiency is defined as:

$$\eta_f = \frac{\tanh \sqrt{\frac{2h_c}{k_t}} L_f}{\sqrt{\frac{2h_c}{k_t}} L_f} \quad (35)$$

The characteristic fin dimension L_f was taken equal to half the tube spacing in 32 and 34. For the true fin in equation 33, it was taken equal to the true fin height plus half the tubes spacing.

3.2.4 Calculation of the Airside Heat Transfer Coefficient h_c

The values of h_c vary along the condenser height, and hence, a point-to-point calculation is required [12]. However, in order to obtain an average value to be used in the simulation, the approximate analysis [13] was followed:

$$h_c = h_a / Z \quad (36)$$

The same relation, equation 4, was used to calculate the convective heat transfer coefficient h_a . The condensation factor Z represents the ratio of the sensible to the total heat load in the condenser:

$$Z = cp_a \frac{dT}{dh} \quad (37)$$

Values of Z were obtained from the derivative of the enthalpy expression presented in [9].

3.2.5 Calculation of Waterside Heat Transfer Coefficient in the Condenser

The waterside heat transfer coefficient was calculated from the known empirical correlation of flow through a pipe [7].

$$Nu_d = 0.23 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.3} \quad (38)$$

4. RESULTS AND DISCUSSION OF SIMULATON OF THE MEH PLANT

The models were solved using software package “gPROMS” [14]. “gPROMS” is a general all purpose modeling, simulation and optimisation tool [15]. All properties of air and water that are necessary for the simulation of the MEH process were written in “C++” separately. An interfacing was made between functions written in “C++” and “gPROMS”. It was established by generating so called “dynamic library link (.dll)” from “C++”. The Comprehensive simulation and effect of various parameters were discussed in detail in [11].

4.1 Assumptions

1.Air enters the humidifier saturated and leaves the humidifier saturated.

2.There is no heat loss to the environment. In practical operation the loss is very low as reported by Nawayseh et al.[5], to be 1 W per square meter of the MEH unit.

4.2 The Heating up Process

To simulate the heating up process of the MEH unit, a constant heating power of 1 kW was assumed instead of solar collector (see Fig 1). Heating up process plays a vital role for the system. Shorter required time of heating results in better performance of MEH unit because less heat will be wasted. Consequently, higher production of condensate can be anticipated. The values of the parameters used for simulation are presented in table 1.

Table 1: The values of the parameters used for simulation

Humidifier	L=3m, P= 1bar, $\dot{m}_w$ = 0.01 kg/s, $\dot{m}_a$ = 0.01 kg/s, A_{hum} = 11.9 m^2, d = 0.055 m.
Condenser	L=3m, P=1bar, $\dot{m}_w$ =0.01 kg/s, $\dot{m}_a$ = 0.01 kg/s, m_w = 5.443 kg, m_s = 240 kg, A_{con} = 8.9 m^2, A_t = 0.57 m^2, A_{cyl} = 1.6 m^2, A_f = 5.73 m^2, k_t = 0.465 kW/mK.

Inlet water temperature to the condenser was kept constant at 20°C. It can be seen from the figure 4 that the heating up process takes approximately 1000s (about 17 min.) after this period the curves are not so stiff. The longest time required to come to steady state is for

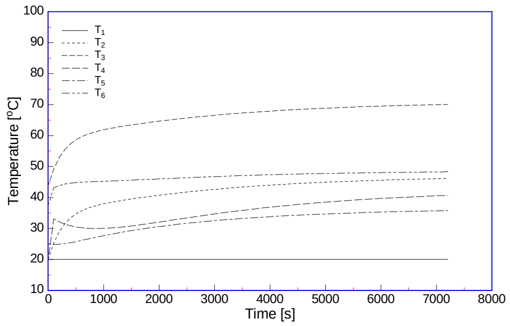

Fig 4. Heating up process of the MEH unit under 1 kW heating power

T_4 and T_5. Because these two temperatures are resulting from combined heat and mass transfer phenomenon. There was an assumption that air enters the humidifier saturated and leaves the humidifier saturated. In reality this is not the case. To satisfy the assumption the process has to prevail for longer time here it is about 3600s or 1h. Modeling and simulation of the heating up process is considered as the most attractive part of this work.

4.3 Fresh Water Production in the Heating up Process

Figure 5 shows the instantaneous fresh water production in heating up process. It is evident from the figure that the instantaneous water production increases rapidly at initial phase and reaches maximum and then drops and comes to steady state. The higher value of instantaneous water production at initial phase is due to the fact, at initial condition the condenser is cold and cold water flows through the condenser to cool down the incoming hot moist air. The cold water does not get enough time to be preheated in the condenser. This leads to lower average temperature of water flowing through the condenser. Thus greater temperature difference exists between hot moist air coming from the humidifier and cooling water of condenser. Consequently, higher amount of moisture condenses in the condenser in initial phase. Figure 6 stands for fresh water production in heating up process. This particular value is very similar as measured by Nawayseh et al.[5].

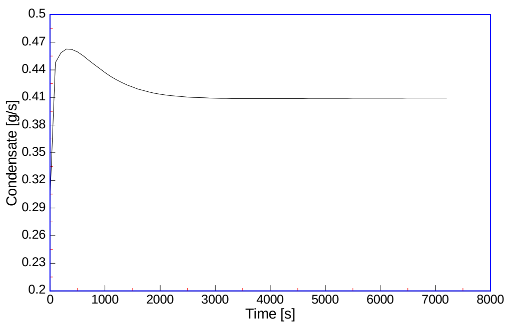

Fig 5. Instantaneous fresh water production in heating up process under constant power of 1 kW

In their work, the production of fresh water under 1kW constant power was 1.7 kg/h, whereas in current work it is 1.6 kg/h, which is less than 5% variation. It is notable that the amount of fresh water production in the current work is lower than that of Nawayseh et al. It is justified because in that paper the reading was taken after steady state condition prevailed, in this work heating up process is taken into consideration.

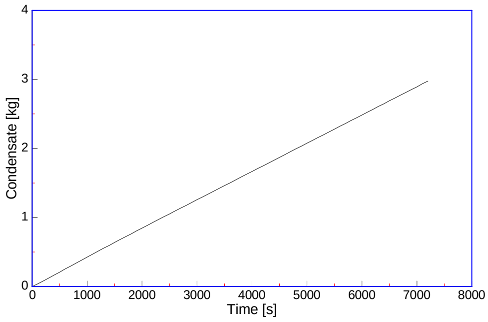

Fig 6. Fresh water production in heating up process under constant power of 1kW

4.4 HRF in the Heating up Process

Figure 7 shows the Heat Recovery Factor (HRF) in the condenser. HRF is defined as the ratio between heat recovered in the condenser to that of total heat used to heat up water before interring in to the humidifier. HRF is defined as follows:

$$HRF = \frac{\dot{m}_w \, cp_w (T_2 - T_1)}{\dot{m}_w \, cp_w (T_3 - T_1)} = \frac{T_2 - T_1}{T_3 - T_1} \qquad (39)$$

It is notable from the figure 7 that about 50 % applied heat is recovered in the condenser as heat of condensation under given condition.

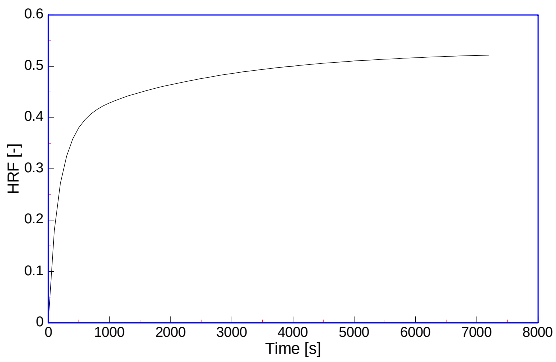

Fig 7. HRF in heating up process under constant power of 1 kW

5. CONCLUSIONS

Detailed models of MEH were presented in this paper that take into account most of the dynamic behavior with reasonable accuracy. Using these models simulation was carried out and results of simulation were discussed. It is evident from the results that current models are very competent to explain transient behavior of the MEH unit. This opens the door to further development of the MEH.

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7. NOMENCLATURE

Symbol	Meaning	Unit
A	Area	(m^2)
a	Area per unit volume	(m^2/m^3)
b	Width	(m)
C	Drag loss coefficient	(-)
cp	Specific heat at constant pressure	(kJ/kgK)
d	Diameter	(m)
F	Volumetric flow rate	(m^3/s)
Gr	Grashof number	(-)
Gz	Graetz number	(-)
h	Enthalpy	(kJ/kg)
h	Convective heat transfer coefficient	(kW/m^2K)
K	Mass transfer coefficient	(kg/m^2s)
k	Conductance	(kW/mK)
L	Length	(m)
L	Mass flow rate	(kg/s)
m	Mass	(kg)
$\dot{m}$	Mass flow rate	(kg/s)
Nu	Nusselt number	(-)
p	Pressure	(Pa)
Pr	Prandtl number	(-)
$\dot{Q}$	Heat flow	(kW)
Re	Reynold's number	(-)
T	Temperature	(°C)
t	Time	(s)
U	Overall heat transfer coefficient	(kW/m^2K)
u	Internal energy,	(kJ/kg)
V	Volume	(m^3)
v	Velocity	(m/s)
w	Moisture content of air (of dry air)	$(kg/kg$
Z	Condensation factor	(-)
Greek letters		
δ	Film thickness	(m)
ρ	Density,	(kg/m^3)
β	Inverse of film temperature,	$(1/K)$
μ	Dynamic viscosity	(kg/ms)
Subscript		
a	Air	
av	Average	
con	condensate	
cond	Condenser	
cyl	Cylinder	
f	Fin	
hum	humidifier	
t	Tube	
s	Steel	
w	Water	
1+x	Taking in to account moisture	
1-6	Various positions	