

**ANALYTICAL MODELING OF A MULTISTAGE BUBBLE
COLUMN HUMIDIFIER**

BY

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Dedicated

To my beloved family for their infinite encouragement and support

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NOMENCLATURE

Symbols

a_s	Specific area (surface area per unit volume)	m^{-1}
A	Cross-sectional area of bubble column	m^2
C_p	Specific heat at constant pressure	$J/Kg.K$
D_{AB}	Diffusion coefficient	m^2/s
d_b	Bubble diameter	m
d_o	Perforated plate orifices diameter	m
g	Gravitational acceleration	m/s^2
h_t	Convective heat transfer coefficient	$W/m^2.K$
h_d	Convective mass transfer coefficient	$Kg/m^2.s$
H	Height	m
H_{c1}	static liquid height	m
H_{c2}	aerated liquid height	m
h	Specific enthalpy	J/Kg
h_{fg}	Enthalpy of vaporization	J/Kg
$\dot{m}$	Mass flow rate	Kg/s
P	Pressure	Pa
$\dot{q}$	Rate of heat transfer	W
R	Thermal resistance	K/W
T	Temperature	$^{\circ}C$
V	Volume	m^3
V_c	aerated volume	m^3
V_g	Superficial velocity	m/s
Q	Heat transfer rate	W
q	Heat flux	W/m^2
k	Thermal conductivity	$W/m^2.K$
E	Effectiveness	$\%$
t	Coil thickness	m

Greek Symbols

ϵ	Gas holdup	—
μ	Dynamic viscosity	$Kg/m.s$
ρ	Density	Kg/m^3
σ	Surface tension	N/m
ω	Humidity ratio	$Kg_{vapor}/Kg_{dry\ air}$
φ_i	Relative Humidity	%

Subscripts

a	Air
c	Liquid in bubble column
i	Inlet
o	Outlet
r	Ratio
sen	Sensible
lat	Latent
sc	Saturated air at column liquid temperature
Sat	Saturation
v	Water vapor
W	Water
da	Dry Air
b	Bubble
C	Column fluid
D	Coil
CD	Column to Coil
DW	Coil to Hot water inside the coil
$turn$	Coil turn
$ratio$	Density ratio of the air and water vapor

Dimensionless groups

GOR	Gain Output Ratio	$m_p h_{fg} / Q_{in}$
HCR	Heat Capacity Ratio	
Le_f	Lewis factor	$h_t / h_d C_{p,a}$
NTU	Number of Transfer Units	$h_d a V_c / \dot{m}_{da}$
Re	Reynolds number	$(\rho_a V_g d_b / \mu_a)$
Sc	Schmidt number	$\mu_a / \rho_a D_{AB}$
Sh	Sherwood number	$h_d d_b / \rho_v D_{AB}$
K	Dean number	
Nu	Nusselt number	$h_t D / k$
Pr	Prandtl number	$C_p \mu / k$
Z	Thickness Ratio	

ABSTRACT

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Lack of fresh water is one of the century issues that increase as the world population increases, whereas most of the world population undergoes clean water scarcity. In order to mitigate such shortage, a considerable effort has been offered in the desalination sector to facilitate and develop a cost-effective, reliable and efficient desalination methodology to process seawater in order to obtain fresh water. Humidification-dehumidification (HDH) is a technique which treated as a gas carrier desalination method that is ultimate to fit for a small-scale water desalination system.

This thesis will focus on a proposed analytical approach to identify the optimum operation and design of the humidifier of the HDH system by utilizing system enhancement such as multi-staging of the humidifier layers. Such enhancement will help in making the fresh water accessible for the world population. The outcomes from this study are of pivotal importance to understand the optimum operating conditions of the humidifier for its possible integration with the dehumidifier. Consequently, an improved HDH system can be obtained. The thesis will also study two different models and their influences on the overall bubble column humidifier performance. Operational and design parameters are studied in this thesis to set the limits of the optimum system parameters. A comparison of

both models is provided to comprehend the difference between these models. Both models were tested and validated with the literature and found to be quantitatively and qualitatively similar. For Multi-staging, different configurations were analyzed in this work; these are series and parallel Multi-staging. This analysis intended to shed light on different configurations effect on the overall system performance. A comprehensive parametric study was investigated to explore the effect of multi-staging. This parametric study includes but not limited to the effect of multi-staging on the total effectiveness, outlet humidity ratio, outlet air temperature, and each stage's effectiveness. The effect of operating factors was also investigated in conjunction with the multi-staging. These operational parameters are the effect of varying water inlet temperature, air Inlet temperature, mass ratio, and inlet relative humidity on exit water and air temperatures as well as the overall effectiveness of the system. Overall, multi-staging has proven its influence in boosting the maximum value of the effectiveness achieved from 33% for the single stage to 94% for the fourth stages of bubble column humidifier. Results have shown that multi-staging influences the outlet air humidity ratio positively, four stages are enough to almost triple the outlet humidity ratio, the humidifier at the fourth stage can produce air with a humidity ratio of almost $0.15 \frac{Kg_{vapor}}{Kg_{dry air}}$ compared with a $0.05 \frac{Kg_{vapor}}{Kg_{dry air}}$ at the first stage. The humidity ratio increment noticed in the multi-stages bubble column humidifier is a sign of a highly effective system since a higher value of humidity ratio leads to a higher amount of water produced of the HDH system.

ملخص الرسالة

الاسم الكامل: مشاري عبدالكريم الباهلي

عنوان الرسالة: دراسة تحليلية رياضية لمبخر هواء متعدد الطبقات

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تاريخ الدرجة العلمية: مايو 2020

مشكلة نقص المياه العذبة هي قضية القرن التي تزداد مع زيادة عدد سكان العالم حيث يعاني معظم سكان العالم من شح المياه النظيفة. لتخفيف هذا النقص، بُذلت جهود كبيرة في قطاع تحلية المياه لتسهيل وتطوير منهجية تحلية مياه فعالة وموثوقة لمعالجة مياه البحر من أجل الحصول على المياه العذبة. (الترطيب-إزالة الترطيب) (HDH) هي تقنية تعامل كوسيلة لتحلية الماء من خلال الغاز الناقل وهي في نهاية المطاف مناسبة لنظام تحلية المياه على نطاق صغير. ستركز هذه الأطروحة على مقارنة تحليلية مقترحة لتحديد التصميم الأمثل للمربط في نظام HDH من خلال الاستفادة من تحسين النظام مثل المربط متعدد المراحل. سيساعد هذا التحسين في توفير المياه العذبة لسكان العالم. هذه الدراسة لها أهمية محورية لفهم ظروف التشغيل المثالي للمربط لإدماجها المحتمل مع مزيل الرطوبة. وبالتالي ، يمكن الحصول على نظام HDH محسن. ستدرس الأطروحة أيضاً نموذجين مختلفين وتأثيرهما على أداء مربط عمود الفقاعات بشكل عام، وسيتم أيضاً دراسة لمختلف عوامل التشغيل والتصميم في هذه الأطروحة لتحديد حدود النظام المثلى. في هذه الأطروحة أيضاً سيتم توفير مقارنة بين كلا النموذجين لفهم الفرق بين هذه النماذج. تم اختبار كلا النموذجين والتحقق منهما ومقارنتهما بالأعمال السابقة المماثلة ووجد أنهما متشابهان من حيث الكم والنوع. بالنسبة إلى المربط ذو الطبقات المتعددة ، تم تحليل نوعين مختلفين في طريقة التركيب، وهي سلسلة ومتعددة المراحل للمربط، وكان الهدف من هذا التحليل هو إلقاء الضوء على تأثير التكوينات المختلفة على أداء النظام العام. لاستكشاف تأثير النظام المتعدد الطبقات على المربط ، تم التحقيق في دراسة شاملة لاستكشاف الأداء الأمثل للنظام. تشمل هذه الدراسة الشاملة على سبيل المثال لا الحصر تأثير المراحل المتعددة على الفعالية الكلية ، ونسبة رطوبة المخرج ، ودرجة حرارة الهواء الخارج ، وفعالية كل مرحلة. بالإضافة إلى ذلك، تم التحقيق في تأثير عوامل التشغيل بالتزامن مع تعدد

الطبقات. هذا التحقيق تجلى في تأثير درجات الحرارة المتغيرة لمدخل المياه ، ودرجة حرارة مدخل الهواء ، ونسبة الكتلة والرطوبة النسبية للمدخل على درجات حرارة خروج الماء والهواء بالإضافة إلى الفعالية الكلية للنظام. بشكل عام ، أثبتت المراحل المتعددة تأثيرها في زيادة القيمة القصوى للفعالية المحققة من 33٪ للمرحلة المفردة إلى 94٪ للمراحل الأربعة من المرطب. أوضحت النتائج أن المراحل المتعددة تؤثر على نسبة رطوبة الهواء بشكل إيجابي ، أربع مراحل كافية لتضاعف نسبة رطوبة المخرج ثلاث مرات تقريباً. إن زيادة نسبة الرطوبة التي لوحظت في المرطب متعدد المراحل هي علامة على نظام فعال للغاية لأن القيمة العالية لنسبة الرطوبة تؤدي إلى كمية أكبر من الماء المنتج من نظام HDH.

CHAPTER 1

INTRODUCTION

1.1 Motivation

The expansion in total populace development results in intense interest for consumable water that is anticipated to be around 7,000 billion m^3 by the year 2030. Currently, the supply of pure water is more than 4,100 billion m^3 , and that is lower than the anticipated consumable water request. The majority of the world population lacks access to drinkable water worldwide, especially those who live in low income communities. The issue is increasingly severe in developing nations where the populace development prediction is a lot higher when compared with developed nations. An example to highlight this issue in India, where 200,000 villages lack access to safe water, which indicates that there is a visible area of need to support creating a sustainable remedy to the rural water issues so that the global water crisis can be resolved. It is a difficult challenge to provide a maintainable remedy to adequately provide the drinkable water demand by a reasonable energy source. It is a fact that water is one of the most available resources present on earth. However, approximately 97.5 % of the earth's water is saline water, whereas the rest is around 2.5 % pure water. The majority of the drinkable water is difficult to extract as it is frozen in its most of the cases. Thus, a low quantity of drinkable water is accessible to help our lives. Knowing the fact that the population keeps on expanding and industrial development hints at no backing off, it is unavoidable that the conventional source of fresh water is not maintainable. To reduce this compromising threat and the fears of moving

toward an emergency, the response for sustainability requires a reliable, efficient and small scale water desalination system. Desalination systems, such as Humidification-dehumidification desalination system (HDH), which uses the same phenomenon of the rain in nature, is a potential technology to solve this challenge. Thus, this thesis will analytically study different performance aspects of the bubble column humidifier as part of the Humidification dehumidification desalination system (HDH).

1.2 Desalination Process

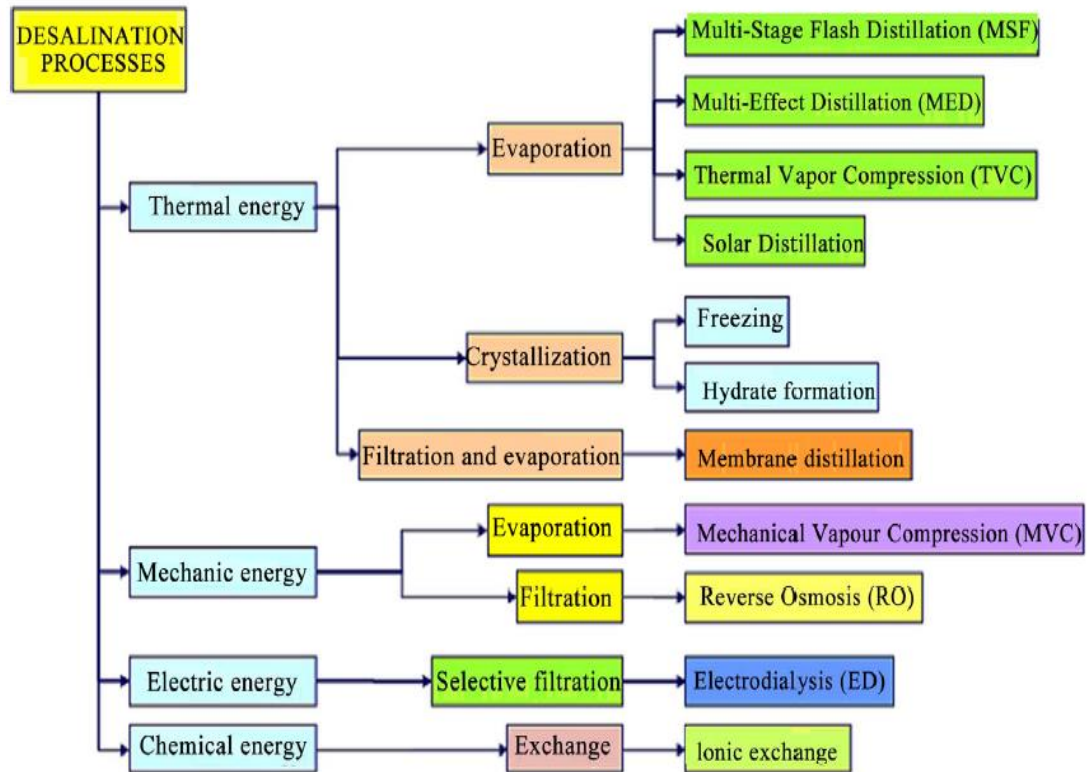


Figure 1-1. Desalination Process Types [32]

Many processes have been developed for water desalination. Figure 1-1 shows various desalination process types. Desalination processes can be categorized into two groups, these are single-phase (membrane process) and phase change (thermal process).

In the process of phase change, sources such as fossil fuels or solar energy are typically used for the evaporation of water, which eventually to be condensed to afford drinkable water. The phase change process includes solar distillers, Freezing distillation, vapor compression (VC) distillation and Multi-Stage Flash (MSF) distillation. On the other hand, Electro Dialysis (ED) distillation and Reverse Osmosis (RO) distillation are the most common types of single-phase membranes desalination process. In comparison, thermal driven desalination technology offers many advantages over membrane technologies. These advantages are the less prone to scaling and fouling as well as the ability to handle the feed water of a higher temperature.

1.3. HDH Process Principle

The process of HDH starts with an air that mixes with a certain amount of steam. The air capability to absorb water vapor is directly proportional to the increase of the temperature. For instance, 1 kg of dry air is capable of absorbing 0.5 kg of steam and about produce a total heat 2814 kJ when increasing the temperature from 30 C° to 80 C°. The phenomenon of extracting the salt from the brine is based on the fact that the airflow passes the saline water and carry a certain amount of steam with the association of the sensible heat that is provided by the brine. After that, fresh water can be produced by allowing the steam of humid air to get in contact with a cold surface, which will result in a condensation of the vapor, usually, condensation occurs in a different heat exchanger where the latent heat recovery heats the brine. Analogs to many other distillation processes, HDH process operates with the only stream of brine. It can also operate with the recirculation of water to make use of wasted energy to operate the system.

1.4 The Technology of Humidification Dehumidification Desalination

The most popular HDH desalination method is solar desalination. At the ambient pressure, seawater starts to evaporate, yielding the air to humidify. Eventually, the humid air will be condensed. The Solar desalination process is also known as the Multiple Effect humidification dehumidification process. It also has different names and types such as solar multistage evaporation condensation cycle (SMCEC) or multi-effect Humidification (MEH), which is a simulation method of the natural water cycle for a short period that will yield to a separation of the freshwater

from other substances by imposing evaporation and condensation. Thermal solar energy is the driving force for this type of HDH system. In advanced systems, the waste heat collected from the condensation is utilized for pre-heating the water source, which will result in an

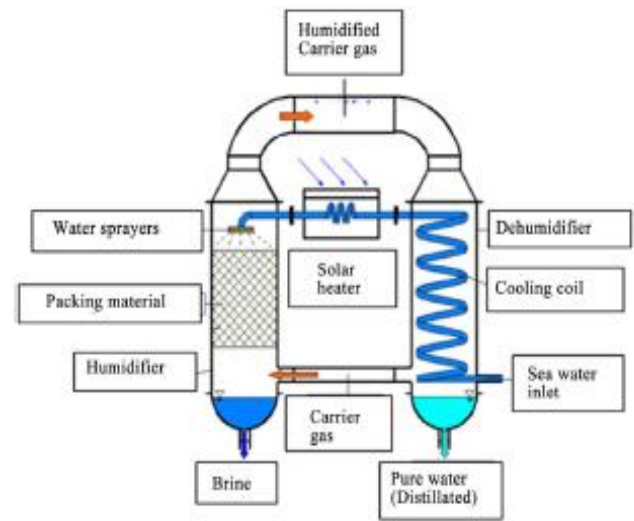


Figure 1-2. HDH process [32]

effective system. This system has proven its high effectiveness for many countries of arid and semi-arid with a small to medium scale desalination systems, and this is mainly due to the financial feasibility of this HDH type. There are several advantages of the HDH process shown in Figure 1-2, these are the simple set of equipment which can work under normal pressure and operation, assets costs are affordable and the flexible scaling of the system size. Because of the extensive use of solar desalination system using HDH over decades, a considerable number of both experimental and analytical types of research have been conducted with consideration of different aspects of the system performance.

1.5 Configuration Classifications of HDH Systems

Commonly, HDH systems are categorized into three main categories, these are heating system type, cycle configuration or type of energy used. Energy based categorization can be geothermal, thermal, solar or hybrid systems. On the other hand, HDH systems cycle configurations are classified as follows, open-air open-water systems, closed-water open-air systems and closed-air open-water systems. As shown in figure 1-3, the flow is driven by either natural or forced forces in all configurations. For heating based classifications, HDH systems can be classified by either air heated or water-heated systems.

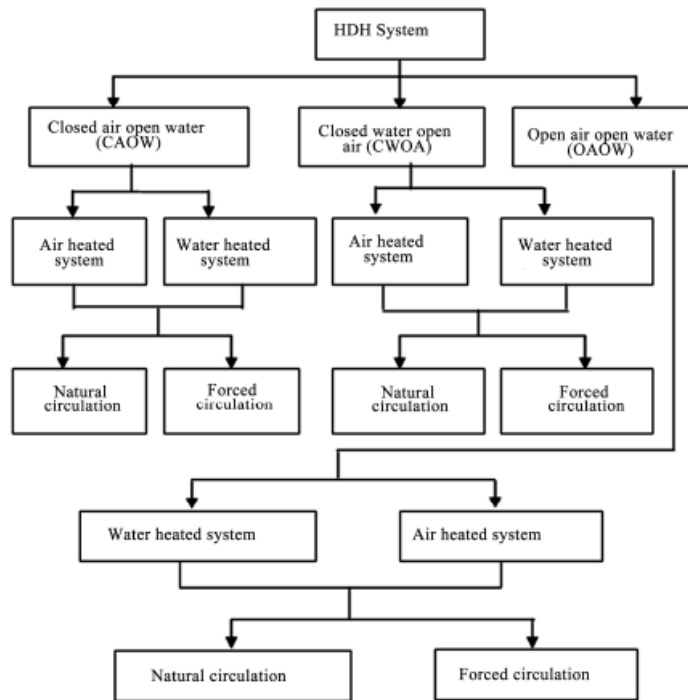


Figure 1-3. Configuration Classifications of HDH Systems [32]

1.6. Main Part of HDH System

1.6.1 Humidifier

In the humidifier of the HDH system, or what is known as an evaporator, water vapor is being held by the air holds when the streams come in contact with each other, it is always intended to moist the air as much as possible.

Spray towers, packed bed towers bubble columns, and wetted-wall towers are the typical air humidification devices used. They all share the same principle of operation, when air and hot water come into a contact where the air is in a dry form, water starts to diffuse into the air and inaugurate the air humidification. The concentration difference between the two streams is the driving force for the diffusion process. The difference in concentration relay on the partial pressure of water vapor in the air as well as the vapor pressure in the interface of gas-liquid streams.

Bubble column humidifiers have rarely been used in HDH desalination system. However, Abugderah and El-Agouz [1] have investigated a single stage bubble column humidifier performance experimentally by blowing air in a bubble form to pass through the seawater. The study has investigated the effect of different factors on humidification efficiency and the difference in moisture content. The results revealed that brine temperature and air velocity are significant factors. In addition, the air inlet temperature has a considerable effect on the difference in vapor content.

Moreover, packing has a significant effect on improving the efficiency of humidification, which is due to the increase of dispersion of water droplets, as well as increasing the contact

time and area. Devices comprising of packing material are so-called packed bed tower. The pack bed tower is just as a column that is set vertically and occupied with packing materials where the water flow falls from the top of the tower while the airflow passes in a counter flow mode. Packed bed tower has proven its influence on the total system effectiveness. Thus, many researchers use them in different HDH systems.

1.6.2. Dehumidifiers in HDH Systems

Researchers also name the dehumidifiers as condensers, since they both have the same operation. Literature has shown that there is a significant number of experimental researches that have been conducted to evaluate the mass and heat transfer that occurs in the dehumidifier.

1.7 Performance Parameters for HDH Systems

Literature had revealed several methods to evaluate the system performance of HDH. These reported methods are, but not limited to, specific water production, the ratio of the output gain (GOR), energy reuse factor, recovery ratio and specific entropy generation.

1.7.1 Gained Output Ratio (GOR)

Gained Output Ratio is also known as the ratio of performance; it is a dimensionless measure, which is the ratio of the amount of produced product to the supplied heat input.

$$GOR = \frac{m_p h_{fg}}{Q_{in}} \quad (1-1)$$

where h_{fg} is the vaporization heat that is calculated at the water inlet temperature. In the case of unity GOR, it entails that the system requires a sufficient supply of heat to vaporize

all the water produced instantly; also, it means that there is no effective energy recovery. One of the leading causes of the unity GOR is the loss associated with any basic solar still. Nevertheless, if the GOR is higher than one, it means that there is less heat supply required. In the case of using fossil fuel as a heating source, the high the GOR is, the less the fuel costs are.

1.7.2 Water production

Water production is an undeniable measure that stands for the amount of produced water per square meter of solar collector area per day. This parameter is an index of the efficiency of solar energy from HDH cycle. This measure is very vital for the solar driven HDH system, since the solar collector costs count for the majority of the capital cost.

1.7.3 Recovery Ratio

Recovery ratio (RR) stands for the ratio of the amount of produced water per kg of brine. It is also known as extraction efficiency. The parameter is generally found to be lower for the HDH system compared with conventional desalination systems.

1.7.4 Energy Reuse Factor

It is defined as the recovered energy ratio from the energy supplied to the water for heating purposes.

1.7.5 Entropy Generation

Entropy Generation is defined as the total entropy produced over the produced water mass flow rate.

1.8 Objectives and Outline of the Thesis

The main object of this thesis is to model a multi-stage bubble column humidifier as a high-performance unit for heating and humidifying the air in a water-heated system. This thesis will focus on a proposed analytical approach to identify the optimum design and operation parameters for the humidifier as part of the HDH system by utilizing system enhancement such as multi-staging the humidifier layers. The thesis will also study two different models and their influences on the overall bubble column humidifier performance. Operational and design parameters are studied in this thesis to set the limits of the optimum system parameters. A comparison of both models is provided to comprehend the difference between these models. Both models will be tested and validated with the literature to quantify the error percentage. Different Multi-staging configuration will be analyzed in this work, and these are series and parallel Multi-staging. This analysis intends to shed light on different configurations effect on the overall system performance. A comprehensive parametric study will be investigated to explore the effect of multi-staging. This parametric study includes but not limited to the effect of multi-staging on the total effectiveness, outlet humidity ratio, outlet air temperature and each stage's effectiveness. The effect of operating factors was also investigated in conjunction with the multi-staging. These operating parameters are the effect of varying water inlet temperature, air Inlet temperature, mass ratio and inlet relative humidity on exit water and air temperatures as well as the overall effectiveness of the system. Provided the boundary conditions used in this study, the obtained results are to be used for determining the required number of stages to get the targeted system performance of the bubble column humidifier.

Chapter 2

Literature Review

Humidification-Dehumidification (HDH) area has been extensively studied. There are many analytical types of research for the Dehumidifier part, whereas the Humidifier area was not given much attention in terms of modeling. This area has many potentials to add value. Below is a comprehensive review of the literature with regards to the humidifier in general and focusing on multi-staging effect on the bubble column humidifier performance.

2.1 Overall literature Review

Munteshari [2] has worked in an enhanced novel cycle of HDH system experimentally in his work. To avoid using an external mechanical energy source, he heated the carrier air before it gets to the expander, which will yield to the required work to drive the compressor. His work demonstrated a thorough thermodynamic analysis to analyze the effects of different factors that are crucial to the performance. These are, but not limited to, mass flow rate ratio, air superficial velocity and water gate height. Moreover, he studied the multi-staging effect on the bubble column humidifier performance. In his work, he used a perforated plate to sparge the air and create the bubbles inside the bubble column humidifier. The perforated plate is located at the bottom of the bubble column humidifier. The bubble column he worked at was constructed of three stages, each one with 20 cm in height as shown in figure 2.1. The multi-stage bubble column humidifier was made of Plexiglas pipes that were set in series with an outer diameter of 0.277 m. The air is sparged from the ambient at a dry status to the system, whereas the water stream derives from the

dehumidifier. As an outcome of Munteshari's study, the system effectiveness was improved by more than 30% for the two stages and more than 45% for the three stage bubble column humidifier. The Maximum achievable effectiveness was 88.9%. On the other side, the pressure losses were found to be relatively low of 910 Pa. Moreover, the system was able to achieve a 100% outlet air relative humidity. Abd-ur-Rehman and Al-Sulaiman [3] had almost a similar scope with two significant differences, these are the source of heat and the multi-stage bubble column humidifier setup, for Abd-ur-Rehman, the source of heat was the solar heat whereas for Munteshari the water was heated using an instant water heater. As shown in figure 2.2, Abd-ur-Rehman experimental setup was put in a stepped shape; for Munteshari, the multistage bubble column humidifier was put in series form. So, Abd-ur-Rehman's work was a multi-stage stepped bubble column humidifier where the primary source of energy to supply his model was the solar thermal energy. His work also investigated the influence of some parameters on the humidifier performance, these are, but not limited to, the influence of the perforated plate geometry, superficial air velocity, inlet water temperature and inlet air relative humidity on

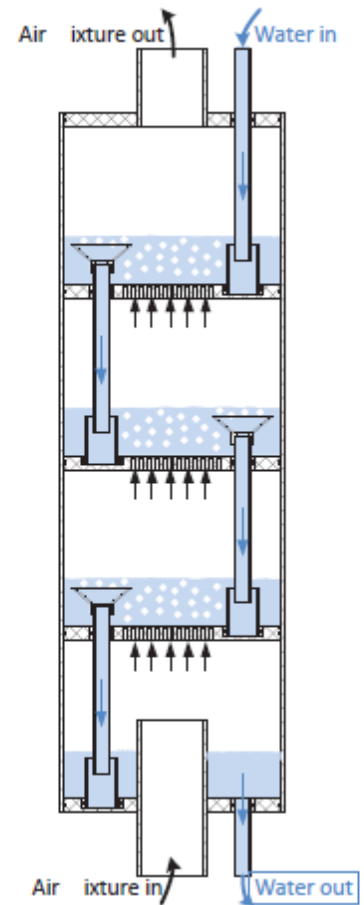


Figure 2-1 Munteshari Multi-Stage humidifier [2]

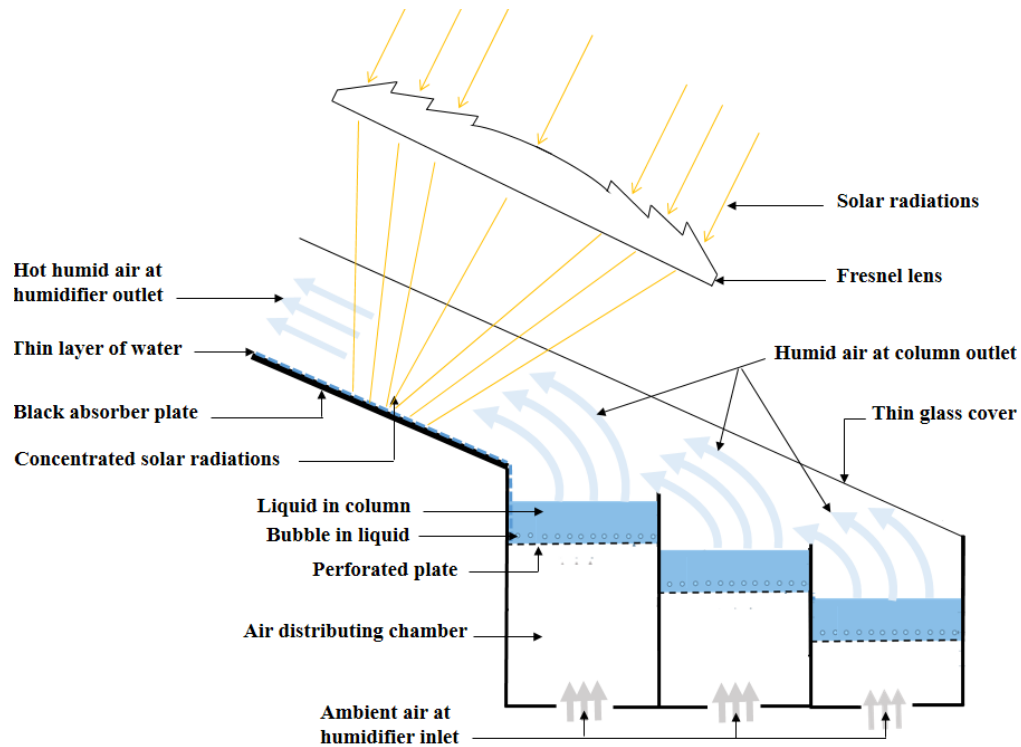


Figure 2-2 Abd-ur-Rehman Multi-Stage humidifier [3]

the performance of the humidifier. The study also covered the effect of the daylight throughout the day on the humidifier performance. Abd-ur-Rehman reported in his work that absolute humidity could increase by 9 % for the two stages configuration and 23 % for the three stages configuration, which shows the effect of multi-staging. The experiment work was executed throughout two different months of June and August to examine the influence of inlet air relative humidity variance. Abd-ur-Rehman concluded that in climatic conditions where the air relative humidity is low, the humidification efficiency of the humidifier shows a significant increase. Findings revealed that the higher the number of stages, the higher the total humidifier effectiveness. The maximum achieved effectiveness was reported to be 49 % for the single stage configuration, 60 % for the two stages, and 81 % for the three stages. research results indicated that the heat transfer coefficient affected greatly by superficial air velocity in the bubbler humidifier, and the higher the air

superficial velocity, the higher the heat transfer coefficient value. Moreover, their findings indicated that the mass transfer coefficient is directly proportional to air superficial velocity where the higher the value of the superficial velocity of air lead to a higher value of mass transfer coefficient, this is because of the link between heat and mass transfer coefficients through the Lewis factor. So, as the superficial velocity of air increases, heat and mass transfer coefficients increase. The outcomes of their research also indicated that the increase in the temperature difference between water and air stream increases the sensible heat flux significantly, whereas the heat transfer coefficient slightly decreases when increasing perforated plate hole diameter or water column height. Their main assumption was that the water in a bubble column is set at a uniform temperature. The results from this study are essential in understanding the ideal operating parameters of the humidifier to get integrated with HDH system. Narayan, G. P. [4] has added a significant contribution to the HDH systems sector, his research intended to develop a model for thermodynamic balancing by utilizing the mass extractions and injections methods, the study also covered the optimum design parameters for bubble column dehumidifier aiming for high rates of heat and mass transfer. He also investigated the optimum system operating parameters. The model has been validated experimentally with a small-scale HDH unit. Narayan has claimed a reduction of 55% in the energy consumed by HDH. The results of his work indicated that multi-staging can increase the effectiveness up to 90%. Furthermore, the study covered a novel factor which is defined as the enthalpy pinch, this parameter was a vital factor in thermal designing of HDH systems, and it is directly reflecting the system performance, the results revealed that the enthalpy pinch was found to be 14.8 kJ/kg dry air. Tow and Lienhard [5] have also added a significant contribution to HDH system

especially for the Dehumidifier part, they reported in their research that LMTD method assumed two main assumptions, these are constant heat capacity in both streams and it does not permit for the thermal energy produced in the vapor molecules stream diffusing a temperature gradient down in parallel with the concentration gradient. Another assumption was that the temperature difference in the bubble outside the boundary layer is considered insignificant when comparing it with the bubble internal temperature difference since air has a greater thermal diffusivity and smaller thermal conductivity than water. Their research revealed that the LMTD method does not apply to the stream with both mass and heat exchange because of the change in heat capacity flow rate. Tow and Lienhard research revealed that the temperature difference was the driver for the sensible heat transfer, whereas a log mean mass fraction difference was proposed to be the driving force for mass transfer. Ben-Amara et al. [7] and Orfi et al. [6] have used an evaporator of a spray tower type in their HDH setup. They studied the influence of water to dry air mass flow rate ratio variation on the system performance provided that the inlet temperature of the water and the absolute humidity are set to be constant. The inlet air temperature was set to be (80 °C), whereas the sprayed water temperature was set to be (60 °C). The authors reported that the higher the amount of water sprayed, the higher the achieved absolute humidity. Nevertheless, it was noticed that additional sprayed quantity would lead to a decrease in the absolute humidity. Hence, for air heated HDH systems, the mass flow rate ratio is found to have an optimum value, which provides maximum possible humidity. For the aforementioned fact, it is suggested to use multi-stage air heating to increase the production of fresh water. Amer et al. [8] have studied the HDH system experimentally with a humidifier 2 m high, 0.8 m long and 0.5 m wide with a packing material of about 6 m²

surface area. This packing is designed not to limit the airflow and keep it moist. By utilizing a hydraulic grid, water was sprayed on the packing material. An opening window was constructed to ease the packing materials change in order to achieve the maximum productivity of 5.8 liters/hr. Zhanian et al. [9] have studied the humidifier in the HDH using a small scale prototype of a cross-sectional area of 0.48 m^2 , and the height was 56 cm. A water distributor is located at the top of the humidifier, which supplies the pad with saline water. The liquid collector was fixed at the bottom of the humidifier to collect the remaining saline water. Therefore, the hot saline water and air will flow in a cross-flow direction. The used packing material was textiles of 14 m^2 cross-sectional area to expand the interface area between water and air. Liu [10] has studied experimentally and analytically the influence of pressure on the humidification-dehumidification processes for HDH system. The constructed humidifier is of 0.1 m diameter and 0.25 m height with an installed heating coil to maintain the thermal balance of the liquid inside the bubble column humidifier. The air enters the column as bubbles through perforated plates located at the bottom of the bubble column humidifier. Different parameters were studied in his work and evaluated against the performance of the system. These parameters were studied and validated in Chapter 3 of this thesis. Liu [10] has developed a model for the bubble column effectiveness (ϵ) correlation with the number of transfer units (NTU), which takes into consideration the effect of heat and mass transfer. Liu [10] concluded that liquid height in the bubble column does not affect the heat transfer provided the range of the heights investigated. Nevertheless, the results showed that the superficial air velocity has a vital influence on the humidifier thermal performance, which is attributed to the increase of turbulence.

2.2 HDH System review as per Configuration Classifications

This section will shed light at a glance at the researches that studied the HDH system. For a better organization, the literature review will be classified as per the configuration type as described in figure 1.3.

2.2.1 Open-air closed-water (OACW) water heated HDH systems

Ref.	Unit Summary	Summary of observation
El-Agouz and Abugderah [1]	- The airflow enters into the humidifier column through 32 holes of 0.01 m diameter each - Holes are being carried by PVC pipe with a 0.075 m diameter - The air is heated using two electric heaters of 1000 W powers each. - The humidifier column was made of steel of 9 cm thickness with 0.5-0.25 m square cross-section and 0.7 m height.	- At 75 °C for water and air streams, authors have claimed a maximum achieved vapor content difference of $222 \frac{g_{vapor}}{kg_{dry\ air}}$ which is considered high in comparison with to that obtained in literature for single stage and very similar for multi-stage - The author reported that humidification efficiency is profoundly affected by the column water temperature and air superficial velocity variation. The efficiency of humidification can reach up to 95%, with the increase of velocity to be 7.5 m/s.
Zhang et al. [11]	- The air is sparged to the humidifier through a sieve plate of 0.198 m diameter and 0.8 cm thickness - There are 91 holes. Each has a 1 mm diameter in the sieve plate, and they are distributed in the form of a regular triangle with a distance of 0.018 m.	- The authors have reported that the maximum obtained air relative humidity reached up 100%. - Throughout the process, the pressure drop was found to exceed 2 kPa
Khedr et al. [12]	- The performance was studied analytically. - The used system encompasses a seawater heater and a dehumidifier - it was recognized as the pioneer system to use contact condenser in HDH system.	Results reported that the GOR of the system is 0.8, heat recovery was reported low in that research as well. Nevertheless, economically, it was found that their HDH Process has a substantial potential for small-scaled desalination plants as small as $10 m^3/day$.

Al-Enzi et al. [13]	• The source of air heating was solar collectors • A blower was used to circulate the air. • The water was preheated to 35-45°C using an electrical heater.	• Authors have studied the effect of temperature variation of cooling water, hot water supply temperature variation, airflow rate and water flow rate variation on the production rate and heat and mass transfer coefficients. • The authors have reported that the maximum rate of production is obtained at a low hot water flow rate and a high airflow rate. • Authors have also concluded that the higher the hot water temperature, the higher the production rate. Whereas, the higher the cooling water temperature, the lower the production rate
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2.1.2 Open-water Closed-air (OWCA) water heated HDH systems

Ref.	Unit Summary	Summary of observation
Garg et al. [14]	• The system used was of a 5 liter capacity thermal storage • 2 m² Solar collector area that has been used for water heating. • The air has been supplied under natural convection mode. • A unique structure was used as the packing bed in the humidifier.	• performance of the cycle is profoundly affected by the inlet water temperature of the humidifier • The authors have concluded that the humidifier column heat loss is a vital factor in measuring the performance precisely.
Nafey al. et [15]	• A dual heating scheme used in this system • Separate heaters for water and air. • An air-cooled dehumidifier was used, thus, there is no latent heat recovery reported in this system. • Canvas type was the used packed bed.	• The maximum achievable reported production was 1.2 liters/hour • The authors concluded that the higher the air mass flow rate, the less the system productivity
Nawayseh et al. [16]	• The solar collector was used to heat the water to 80 °C. • The air was moving in both forced and natural modes. • Wooden slates were used as packing bed for the humidifier	• The achieved reported water production was 5 kg/hr. • The results have shown that the flow rate of water is significantly affecting the packing bed wetting area. • Authors concluded that the air natural driven mode would lead to better results in comparison with forced circulation.

Ben Bacha et al. [17]	• 6 m^2 Solar collector used as the primary source of water heating. • The system compromise of a storage tank that operates with a minimum temperature limitation. • Saline water from a well was provided a cooling water • Thorn trees packed bed type was used • The dehumidifier was constructed of polypropylene plates.	• A considerable amount of water production was reported at 19 liters per day. • The authors have concluded that in order to provide the same amount of fresh water, provided that the thermal storage was not used, a need of 16% more solar collector area is required • Authors have also concluded that the water inlet temperature, the water and airflow rate along with the humidifier packing material are significant factors in the performance of the system.
Al-Hallaj et al. [18]	• The system used a 2 m^2 solar collector as the primary source to heat the water to 50-70°C and • The system was analyzed by the air circulation modes using both natural and forced convection • Humidifier was constructed of a wooden surface at the top • Bench unit had a specific area of 87 $1/m$ • Pilot unit had a specific area of 14 $1/m$ • Condenser had a cross-sectional area of 0.6 m^2 for bench unit whereas the pilot unit had 8 m^2	• The authors reported that there is an optimum value for the flow rate of water where the system performance will reach the maximum • At low temperatures, results revealed that the forced air circulation is positively affecting the system performance, on the other hand, at high temperatures, natural circulation becomes advantageous to the system performance.
Klausner et al. [19]	• Utilizing the waste heat was used to heat the water to 60°C • A direct contact packed bed was used as the system dehumidifier. • The system also utilized the water produced in the dehumidifier as a coolant.	• The authors reported that this process could produce a fresh water efficiency of 8% with an energy consumption of 0.05 kWh/Kg of fresh water production at a feedwater temperature of 60°C
Farid et al [20]	• solar collector of a 1.9 m^2 area was used for heating the water. • The air was under forced circulation mode. • The system packing bed used is of Wooden shaving type	• Reported achieved production was 12 Liter/m^2 • The authors were not able to reach to a conclusion with the effect of air superficial velocity on the production • The reported results presented an optimum value of the water flow rate

2.1.1.2 Open-water closed-air (OWCA) air heated HDH systems

Ref.	Unit Summary	Summary of observation
Orfi et al. [6]	• 2 m^2 solar heaters were used to heat both air and water • packing bed that consists of spongy material was used for this system. • The used evaporator was designed to move the heated water in the horizontal surface and the capillaries in the vertical plates and air pass in different directions	• The authors revealed that an optimum mass flow ratio between air and water could be found which will yield to the maximum humidification
Yamali al. et [21]	• The solar collector of the flat plate type was used to heat the water. • The used humidifier was a pad humidifier. • A heat exchanger of the tube-fin type was used as a dehumidifier. • Storage tank with a $0.5 m^3$ water volume was sued in this system	• the authors claimed that the system could produce up to 4 kg/day. • Results indicated that there is no effect of increasing the air flow rate to the system performance. Whereas, the increase in the mass flow rate of water increases the system productivity.
Houcine et al. [22]	• This system consists of five stages for humidification and heating purposes. • FFWP type with $4.98 m^2$ area was used to construct the first two stages. The other collectors are all of classical commercial type • The air was under forced circulation mode.	• The authors claimed a maximum production of water to be 516 L/day • Corrosion was observed in the plant due to the corrosive seawater. • The air temperature reached 90°C. • The cost of water production of this system was high
E. Chafik [9]	• $2.08 m^2$ Solar collectors of four fold web plate (FFWP) type was used to heat the air to 80°C • Multi-stage was used for humidification and heating in this system • Forced circulation of air.	• The constructed system was too expensive where the air heaters counted for almost 40% of the cost • Authors have concluded that by minimizing the pressure drop, the HDH system can be enhanced.
Ben-Amara al. et [22]	• FFWP collectors were used to heat the air to a temperature of 90°C • Solar collectors were made of Polycarbonate plates and aluminum strips • The system used an insulation of Aluminum foil and polyurethane	• Different factors were studied in this work and their effect on the system performance, these are but not limited to air mass flow rate variation, humidity and temperature of inlet air and wind velocity variation. • As a result, the less the wind velocity, the higher the collector efficiency.

CHAPTER 3

ANALYTICAL MODELING OF THE HUMIDIFIER

In this chapter, a mathematical model will be developed and analyzed of the bubble column humidifier under two different scenarios.

- 1- Stagnant water with a heating coil that maintain the required energy (base model)
- 2- Moving air and water streams

The first model will be considered as the base model for the other models that will be studied throughout this thesis. The base model was validated with Liu [10] and found to be accurate with a with an insignificant error percentage. For both models, the operational and design parameter on the system performance will be investigated to capture all enhancements that can be attributed to the operation and design factor variation. A comparison of both models will be provided to comprehend the difference between these models. Both models will be tested and validated with the literature to raise the confidence level of the results reliability.

3.1 Stagnant Water Model

This model represented in Figure 3-1 is a bubble column humidifier with stagnant water that is associated with a

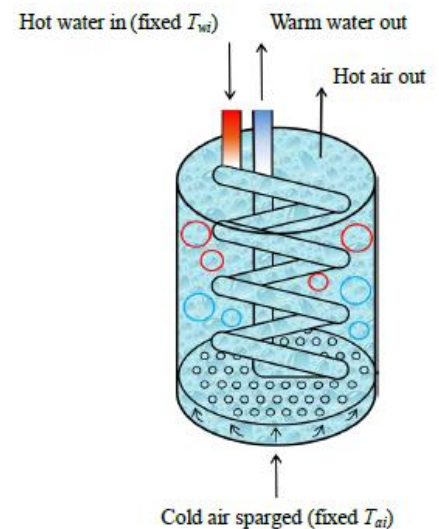


Figure 3-1. Schematic of Bubble Column Humidifier [10]

heating coil to maintain the desired temperature; the fluid inside the coil is water. To

examine this model analytically, we need to define the geometry and configuration of the system. This will be illustrated in the next section

3.1.1 Geometry

A schematic of the bubble column humidifier is presented in figure 3-2. The humidifier has an inside diameter of 100 mm, 250 mm in height and 10 mm thickness. The air enters the bubble column humidifier in a bubble form by utilizing a perforated plate fixed at the bottom of the column. The bubble column is occupied with water to a certain level. In addition, the bubble column is associated with a heating coil to control the temperature of the water.

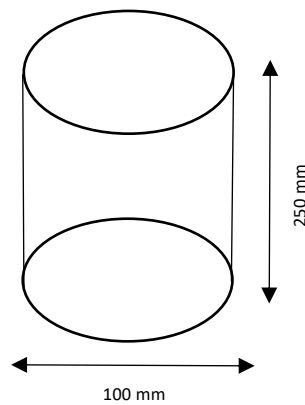


Figure 3-2. Bubble Column Humidifier Geometry Schematic

The immersed heating coil in the column provides the needed heating load. The copper coil has 0.95 cm outside diameter, a wall thickness of 0.15 cm, and is spiraled to a length of 100 cm with a turn diameter of 17 cm.

The perforated plate is a cylindrical box that has a 98 mm outside diameter and has 83 holes, each one of them is of 1 mm diameter. Holes are to be considered uniformly distributed over the plate.

3.1.2 Mathematical Modeling

In the bubble column humidifier, the humidification process takes place when the air at a cold temperature enters the bubble column humidifier in a bubble form sparged through a perforated plate located at the bottom of the column. Consequently, uniformed bubbles are being generated in the bubble column humidifier. The inertia and buoyancy forces are the rise drivers of the bubbles. Thus, these bubbles get into direct contact with the stagnant warm water in the bubble column humidifier. The water temperature in the column is controlled by a heating coil, as described in the previous section. Furthermore, air inlet temperature and the water in coil inlet temperature are to be considered constant. As a result of the direct contact between the air bubbles and the stagnant water in the bubble column humidifier, Heat and mass transfer take place due to temperature and concentration differences. Thus, water starts to lose sensible heat to the air; that heat is to be supplied by the heating coil. In addition, latent heat, which governs the mass transfer between the water molecules to the air bubbles, contributes to the majority of heat transfer between the air and bubble. Figure 3-3 shows the thermal resistance network between the air bubbles and the water in the bubble column humidifier.

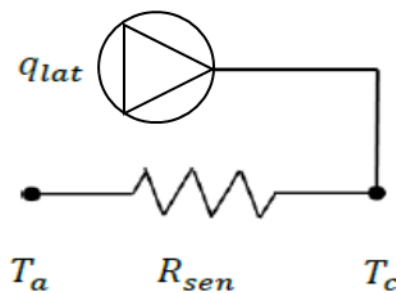


Figure 3-3. Thermal Resistance Network

The bubble column humidifier can be treated as a cooling tower since they have a similar function where the flow in the coil and the air that sparged into the column are a representation of a counter-flow heat exchanger. In the following analytical investigation, some assumptions are considered in this model, these are

- The system operates in a steady-state condition.
- Bubbles assumed to not impinge with the heating coil.
- Heat losses from the system to the ambient are to be neglected.
- The water temperature in the bubble column humidifier is set to be uniform

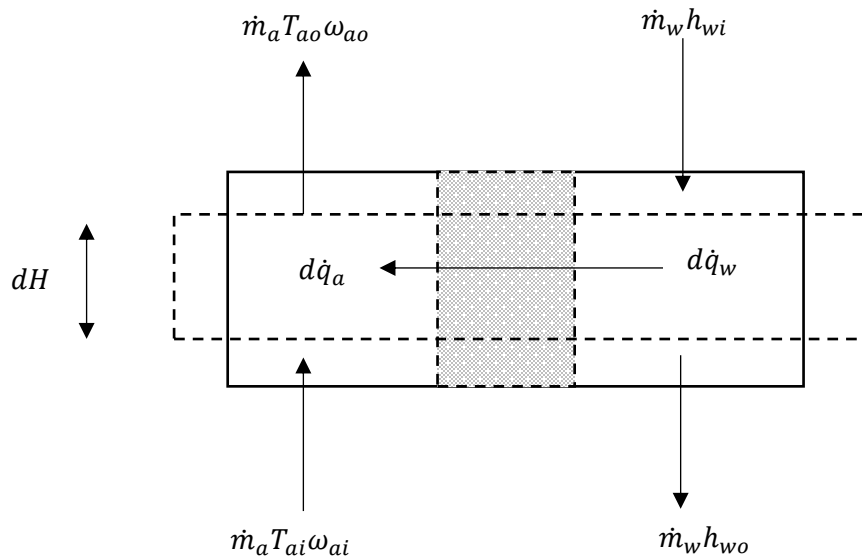


Figure 3-4. finite volume element in a bubble column humidifier

Figure 3-4 shows a segment of a control volume of the humidification process that occurs between the air and the water in the humidifier. The bubble column humidifier can be represented analytically to calculate the performance and investigate the influence of different parameters on the system performance by understanding the process of heat and mass transfer represented in figure 3-4. The energy balance for the total system on the water and air streams can be written as follow

$$d\dot{q}_a = d\dot{q}_w \quad (3-1)$$

Equation 3-1 can be represented in enthalpy form for each stream to be

$$d\dot{q}_a = \dot{m}_a h_a \quad (3-2)$$

$$d\dot{q}_w = -\dot{m}_w h_w \quad (3-3)$$

The negative sign that appears in equation (3-3) represents the loose of heat from the water to the air.

The heat transfers from the water to the air can be represented by sensible and latent heat whereas latent heat represent most of the transferred heat, both heat types can be analytically represented as follows

$$Q_{Sensible} = h_t \times a_s \times (T_c - T_{ai}) \times A \times H \quad (3-4)$$

$$Q_{Latent} = h_d \times a_s \times (\omega_{sc} - \omega_i) \times h_{fg} \times H \quad (3-5)$$

$$Q = Q_{Sensible} + Q_{Latent} \quad (3-6)$$

To relate the mass transfer coefficient with the heat transfer coefficient, we will introduce the Lewis factor

$$Le_f = \frac{h_t}{h_d c_{p,a}} \quad (3-7)$$

From literature, Le_f is usually ranges between 1 ~ 0.7; hence, in our model, we will set the value to be unity.

Mass balance for the system can be applied similarly to the cooling tower [10], so over a finite element, differential mass balance equation is as follows

$$d\omega_a = \frac{h_d \times a_s \times A (\omega_{sc} - \omega_{a,i}) \times dH}{\dot{m}_a} \quad (3-8)$$

Integrating both sides, equation (3-6) will result in the following equation

$$\omega_{a,o} = \omega_{a,i} + \frac{h_d \times a_s \times A (\omega_{sc} - \omega_i) \times H}{\dot{m}_a} \quad (3-9)$$

The inlet humidity ratio can be found knowing saturation pressure and the relative inlet humidity as follows

$$\omega_{a,i} = \frac{(0.622 \times \phi_i \times P_{sat_a})}{P - (\phi_i \times P_{sat_a})} \quad (3-10)$$

Where P is the total pressure in Pascal

Humidity ratio at saturation is the state where the humidity ratio found at the column water temperature as follows,

$$\omega_{sc} = \frac{(0.622 \times 1 \times P_{sat_w})}{P - (1 \times P_{sat_w})} \quad (3-11)$$

h_d in equation (3-7) is known as the convective mass transfer coefficient, in order to find the value of h_d we need to define the specific interfacial area of the bubble (a), which is as follows

$$a_s = \frac{6\epsilon}{d_b} \quad (3-12)$$

The gas hold-up mentioned in equation (3-12) can be found by either following the Joshi and Sharma definition [23],

$$\epsilon = \frac{V_g}{0.3 + 2V_g} \quad (3-13)$$

Or, by following Akita and Yoshi [24] definition of the gas hold up to be

$$\epsilon = \frac{H_{c2} - H_{c1}}{H_{c2}} \quad (3-14)$$

Where H_{c1} represent the static liquid height and H_{c2} stands for the aerated liquid height in the bubble column humidifier.

The aerated liquid height is the lead to find the aerated volume which will be used to find the Number of Transfer Unit (NTU) by multiplying it with the bubble column humidifier cross-sectional area as follows,

$$V_c = H_{c2} \times A \quad (3-15)$$

In regards to the bubble diameter, provided the hole diameter d_o , Miller [25] and Narayan [4] have determined the bubble diameter to be as the following

$$d_b = \left\{ \frac{6\sigma d_o}{(\rho_l - \rho_g) \cdot g} \right\}^{\frac{1}{3}} \quad (3-16)$$

aforementioned equations discussed the heat transfer side of the system. Furthermore, it is not easy to figure out the mass transfer mechanism between the bubble and the liquid in the bubble column humidifier. Miller [25] has investigated the convective mass transfer coefficient, his model was a vapor that diffuses into a vapor-air mixture. The model has been subject to further modification by Liu [10] to capture the pressure effect. In the present work, we will adopt the same methodology to investigate the mass transfer between the bubble and liquid in the bubble column. The modified correlation is simply relating dimensionless number together, and these numbers are Sherwood number, density ratio of the air and water vapor, Schmidt number and Reynolds number as follows

$$Sh = 0.32 \times Re^{0.43} \times Sc^{0.21} \times \rho_{ratio}^{1.3} \quad (3-17)$$

Whereas Sherwood number can be represented in another form to be as follows

$$Sh = \frac{h_d d_b}{\rho_v D_{AB}} \quad (3-18)$$

Where D_{AB} is the diffusion coefficient and it is a property of a state. ρ_v is the water vapor density.

Reynolds number is simply the ratio of inertial forces to viscous forces in the fluid which is defined as follows

$$Re = (\rho_a V_g d_b) / \mu_a \quad (3-19)$$

Whereas μ_a is the viscosity of the air. Schmidt number is a dimensionless number which relates the air viscosity to the air density and diffusion coefficient as follows

$$Sc = \mu_a / (\rho_a D_{AB}) \quad (3-20)$$

ρ_{ratio} mentioned in (3-17) is the density ratio of the air to the water vapor in the bubble column humidifier.

Furthermore, Braun et al. [26] have developed an ε -NTU model for a cooling tower that runs in a counter flow mode. This model is mainly intended to identify the mass transfer coefficient. Since the cooling tower and bubble column have an identical process, the model can be used to identify the mass transfer coefficient. The surface area is the only difference between these two models, the surface area of the bubble column is the surface area of the bubbles, however, in cooling towers, the packing surface area is the considered area. This model has also been adopted by Liu [10]. The correlation is as follows

$$NTU = \frac{h_a a V_c}{\dot{m}_{da}} \quad (3-21)$$

Where a is the specific interfacial area and V_c is the aerated volume inside the column, which can be found in (3-15).

In order to measure the performance of the system, there are multiple ways aforementioned in the introduction chapter, the most critical factor is the effectiveness which has multiple definitions, the most representative one is the enthalpy-based effectiveness which is to

relate the actual performance with the ideal performance. Since the air stream has the minimum heat capacity, the effectiveness correlation will be as follows,

$$E_H = \frac{h_{a,o} - h_{a,i}}{h_{a,o}^{ideal} - h_{a,i}} \quad (3-22)$$

Where $h_{a,o}^{ideal}$ is the air enthalpy that is measured at the inlet water temperature, which reflects the maximum enthalpy that the air can obtain.

Another vital performance measure is the Heat Capacity Ratio (HCR), which stands for the ratio of maximum possible total enthalpy change of the cold stream to the hot stream. Narayan [4] has introduced a correlation of the heat capacity ratio (HCR) for bubble column humidifier to be as follows

$$HCR = \min \left\langle \frac{h_{sa,wi} - h_{ai}}{h_{wi} - h_{w,ai}}, \frac{h_{wi} - h_{w,ai}}{h_{sa,wi} - h_{ai}} \right\rangle \quad (3-23)$$

In this thesis, the air stream found to has the minimum heat capacity.

In order to relate the effectiveness with the Heat Capacity ratio, Braun et al. [26] have introduced the following correlation.

$$E = \frac{1 - \exp[-(1 - HCR).NTU]}{1 - HCR.\exp[-(1 - HCR).NTU]} \quad (3-24)$$

With the aforementioned set of equations, the system is fully defined and can be modeled for further investigation

3.1.3 Coil Geometry

The aforementioned model described is set for stagnant water with a moving air as shown in Figure 3-10. From a design perspective, the required coil length and outer diameter is crucial to the system operation and demanded temperature, in addition to that, the coil geometry plays a significant role in the bubble movement as there will portion of the bubble that will impinge with the coil causing a real issue to the uniformity of the air bubble temperature. This section will investigate the coil side from a heat transfer perspective to reach the optimum coil geometry provided the aforementioned assumptions and heat load.

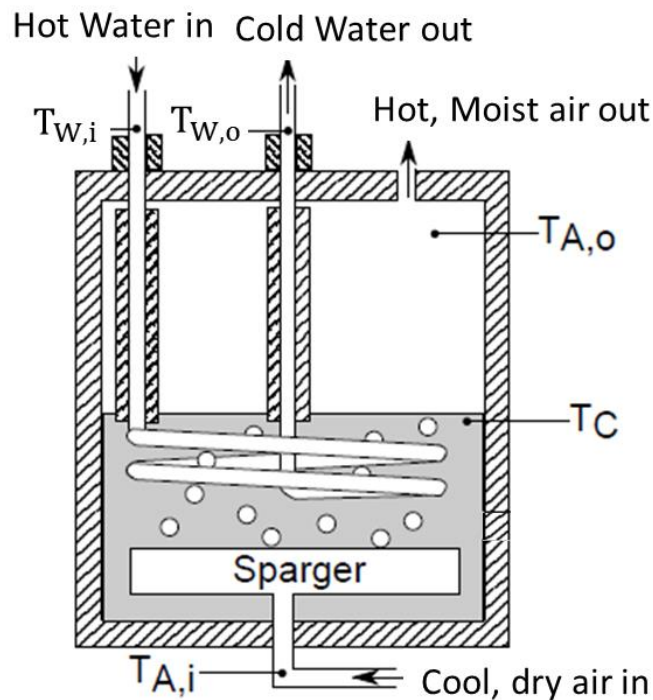


Figure 3-5 Schematic of the Bubble column humidifier and the heating coil

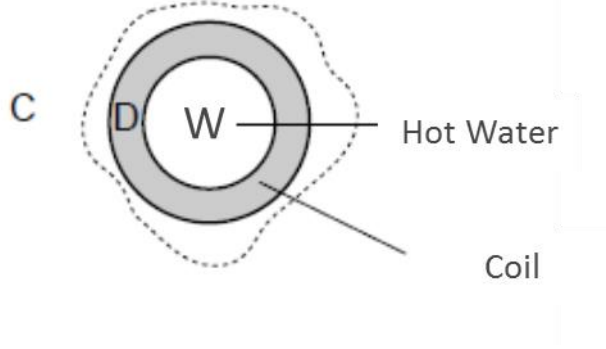


Figure 3-6 Schematic of the resistances between heating coil and the water in the column

It is a fact that the amount of heat gained by the air (which is described in the previous section) will be equal to the amount of heat lost by the heating coil, which can be described as follows

$$Q_{air} = Q_{coil} \quad (3-25)$$

Tow and Lienhard [5] has studied the coil heat transfer for a dehumidifier which can be applied for the humidifier since both have the same structure, so the heat transfer through the coil can be represented in the following form

$$Q_{coil} = \frac{\Delta T_{LMTD, CW}}{R_{DW} + R_{CD}} \quad (3-26)$$

Where R_{DW} and R_{CD} are the resistance of the coil to the water inside the coil and the resistance of the coil to the water in the bubble column respectively. To model the driving thermal force across the coil log-mean temperature (LMTD) can be used as follows,

$$\Delta T_{LMTD, CW} = \frac{T_{W,o} - T_{W,i}}{\ln\left(\frac{T_C - T_{W,i}}{T_C - T_{W,o}}\right)} \quad (3-27)$$

Mori and Nakayama [27] have developed correlations for the heat transfer coefficient inside curved tubes, which can be used to formulate the Resistances inside and outside the coil mentioned in equation (3-26).

Hence, the resistance between the column water to the coil is

$$R_{CD} = \frac{1}{h_{t,CD} A_{D,o}} \quad (3-28)$$

The Nusselt number of the coil is

$$Nu_{CD} = \frac{h_{t,CD} x_{D,o,coil}}{k_C} \quad (3-29)$$

Where k_C is the thermal conductivity of the water inside the bubble column which is a property of the water at a specific temperature.

Deckwer's model [28] introduced an empirical correlation that can predict heat transfer coefficients over a wide range of operating conditions, his model is based on the indication that the bubbles' flow acts like small eddies which interact with the heat transfer surface.

The empirical correlation is as follows,

$$h_{t,CD} = 0.1 k_C^{1/2} \rho_C^{3/4} C_{P,w}^{1/2} \mu_C^{1/4} g^{1/4} V_g^{1/4} \quad (3-30)$$

In the other hand, the resistance between the coil and the water inside the coil has been analytically studied by Mori and Nakayama [27], the analytical correlation is

$$R_{DW} = \frac{1}{h_{t,DW} A_{D,i}} \quad (3-31)$$

Nusselt number formulation in a curved tube can be found using the dimensionless numbers of the Dean Number K and the thickness parameter Z

$$Nu_{D,DW} = \frac{h_{t,DW} D_{D,i}}{k_W} = 0.8636 \left(\frac{K^{1/2}}{Z} \right) \quad (3-32)$$

Provided the Prandtl number of the water inside the tube, Thickness parameter Z can be found using below correlation

$$Z = \frac{2}{11} \left(1 + \sqrt{1 + \frac{77}{4} Pr_W^{-2}} \right) \quad (3-33)$$

Dean number is a dimensionless number which can be found using the below formula

$$K = Re_W \left(\frac{D_{D,i}}{D_{turn}} \right)^{1/2} \quad (3-34)$$

Where D_{turn} is the coil turn diameter.

3.2 Results and Discussion

Equations 3-1 to 3-34 are modeled and solved numerically utilizing Engineering Equation solver (EES) software. Moist air and liquid water physical properties are available in the software. Results investigated the influence of operational and design parameters. The operational parameters are the effect of superficial velocity on outlet air and water temperature, effectiveness and mass transfer coefficient as well as the effect of water height on the system effectiveness. On the other side, different design parameters were investigated to gauge the influence of different parameters on the outlet temperature of the air with the consideration of the optimum coil geometry. Theses design parameters are the effect of coil outlet diameter and length variation on the coil outlet water temperature and the achieved Reynolds number.

Table 3-1 Assumed Variables for the stagnant water model

Symbol	Definition	Value	Unit
d_o	Hole Diameter	0.001	m
H	Liquid Column Height	0.05	m
Le_f	Lewis Factor	1	-
$\dot{m}_a$	Inlet Air Mass Flow Rate	0.0015	Kg/s
$T_{a,i}$	Inlet Temperature of Air	15	$^{\circ}C$
T_c	Bulk Temperature of The Liquid Column	45	$^{\circ}C$
V_g	Superficial Gas Velocity	0.05	m/s
A	Surface Area	0.0785	m^2
P	Inlet and Outlet Pressure of Air	101325	Pa

Table 3-2 Assumed Values for the coil

Symbol	Definition	Value	Unit
$D_{D,o}$	Coil Immersed Outer Diameter	0.0095	m
t	Coil Thickness	0.0015	m
L	Coil Length	1	m
D_{turn}	Turn Diameter	0.17	m
$T_{w,i}$	Inlet Temperature of Hot Water in The Coil	60	$^{\circ}C$

3.2.1 Influence of Operational Parameters

Effect of Air Superficial Velocity on Air Outlet Temperature

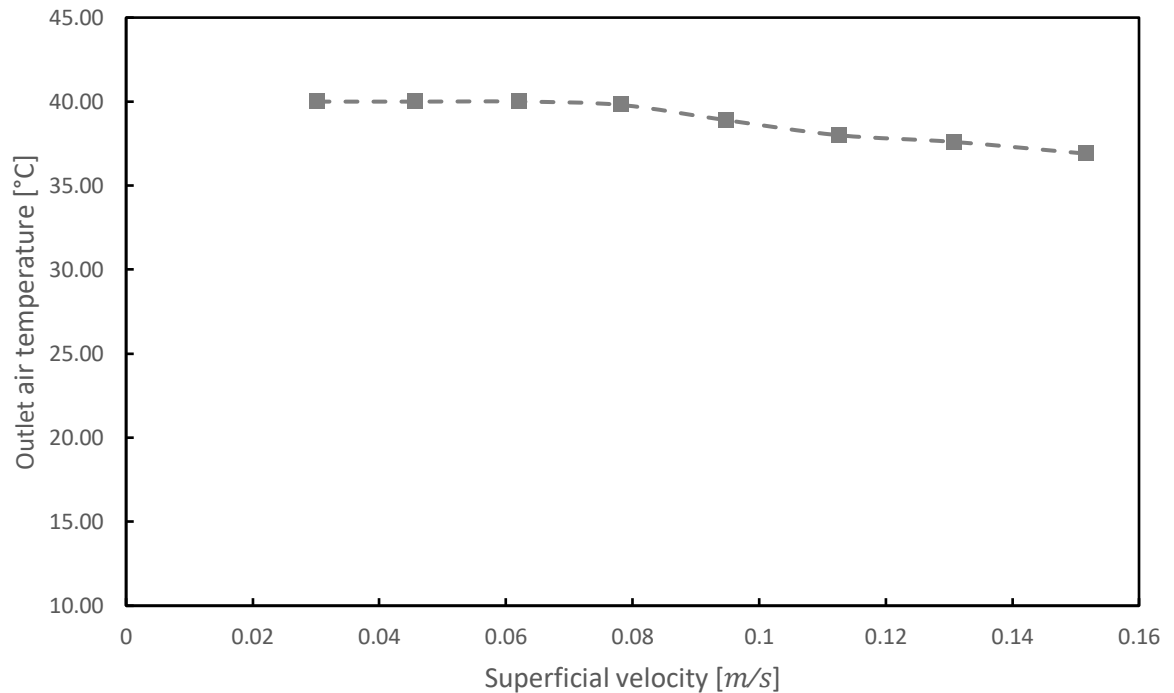


Figure 3-7 Effect of superficial velocity on outlet air temperature

In figure 3-7, the results show the effect of varying air superficial velocities on the air outlet temperatures. The air outlet temperature decreases with the increase of the superficial velocity. The increase of the heat transfer coefficient due to the increase of air superficial velocity is the leading attribution to the decrease of the exit air temperature. From another perspective, NTU decreases with the increase of airflow; subsequently, the air effectiveness decreases, which will lead to a decrease in the exit air temperature.

Effect of air Superficial Velocity on Water Outlet Temperature

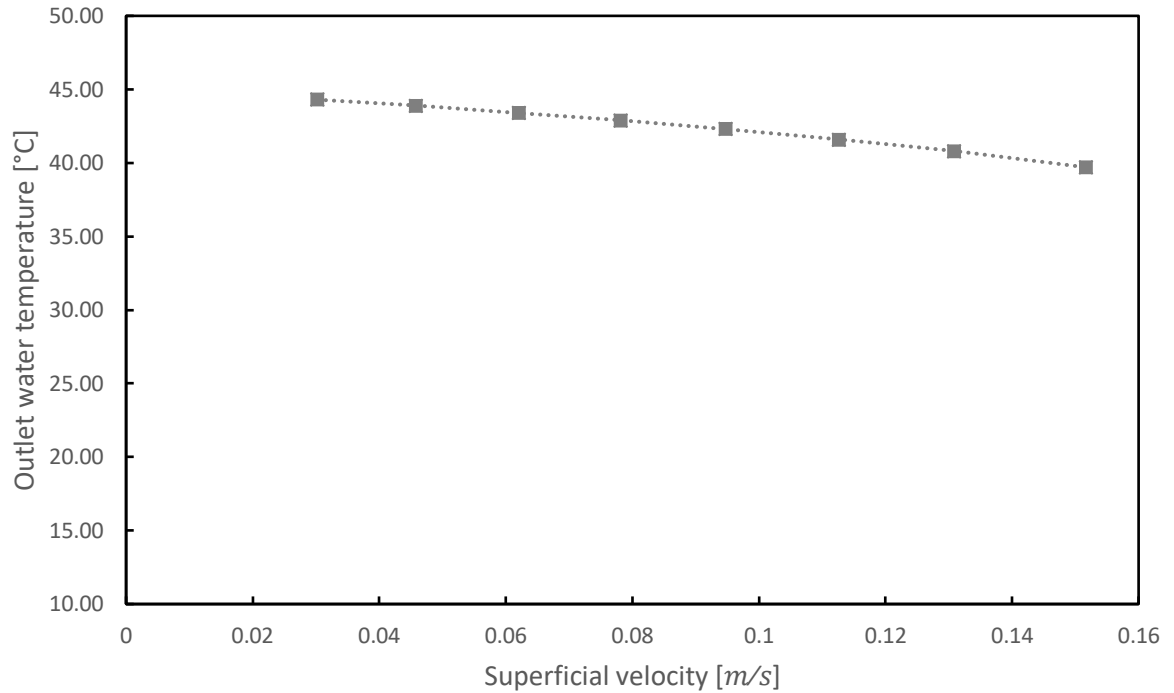


Figure 3-8 Effect of air superficial velocity on water outlet temperature

Figure 3-8 shows the effect of superficial velocity on the water outlet temperature. Result indicated that the higher the superficial air velocity is, the lesser the water outlet temperature. Knowing that the inlet temperature and flow rate of water are fixed, the decrease in the exit water temperature is attributed to the increase of the airflow rate, which directly increases the rate of sensible heat transfer that leads to a decrease in the water outlet temperature.

Effect of Air Superficial Velocity on Total Rate of Heat Transfer

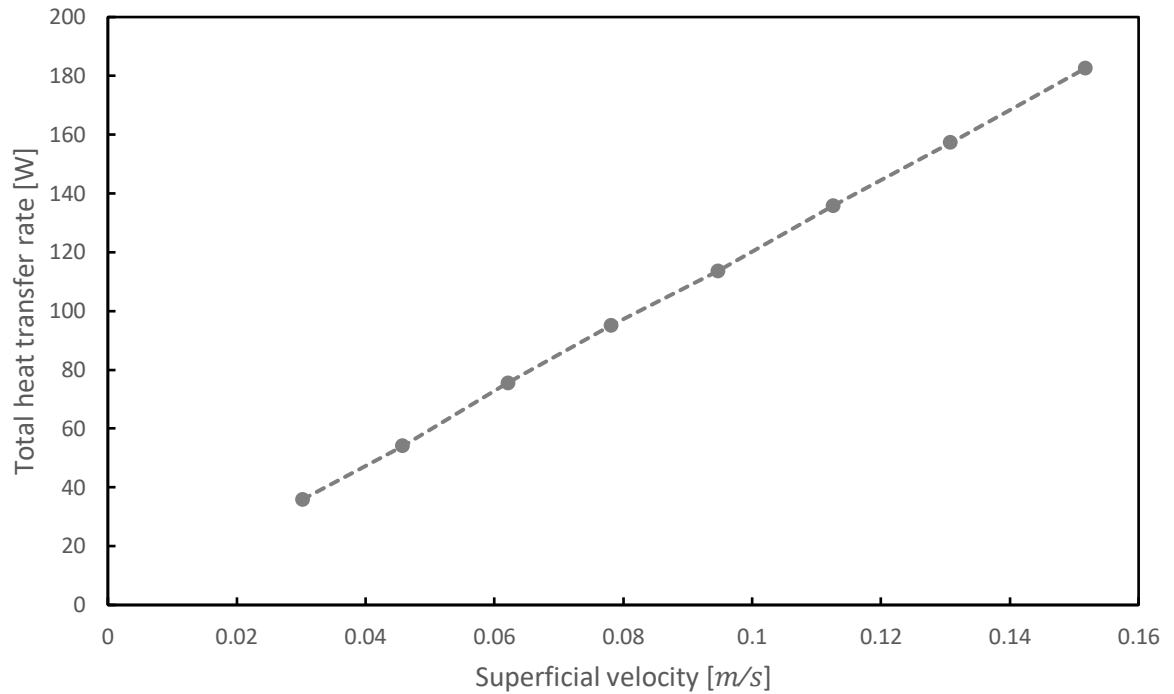


Figure 3-9 Effect of air superficial velocity on total heat transfer rate

Figure 3-9 illustrates the influence of varying superficial air velocity on the total rate of heat transfer. Findings indicated that the higher the superficial air velocity, the higher the total rate of heat transfer, this is primarily attributed to the mass flow rate increase that yields to a higher sensible heat transfer rate.

Effect of Air Superficial Velocity on Effectiveness

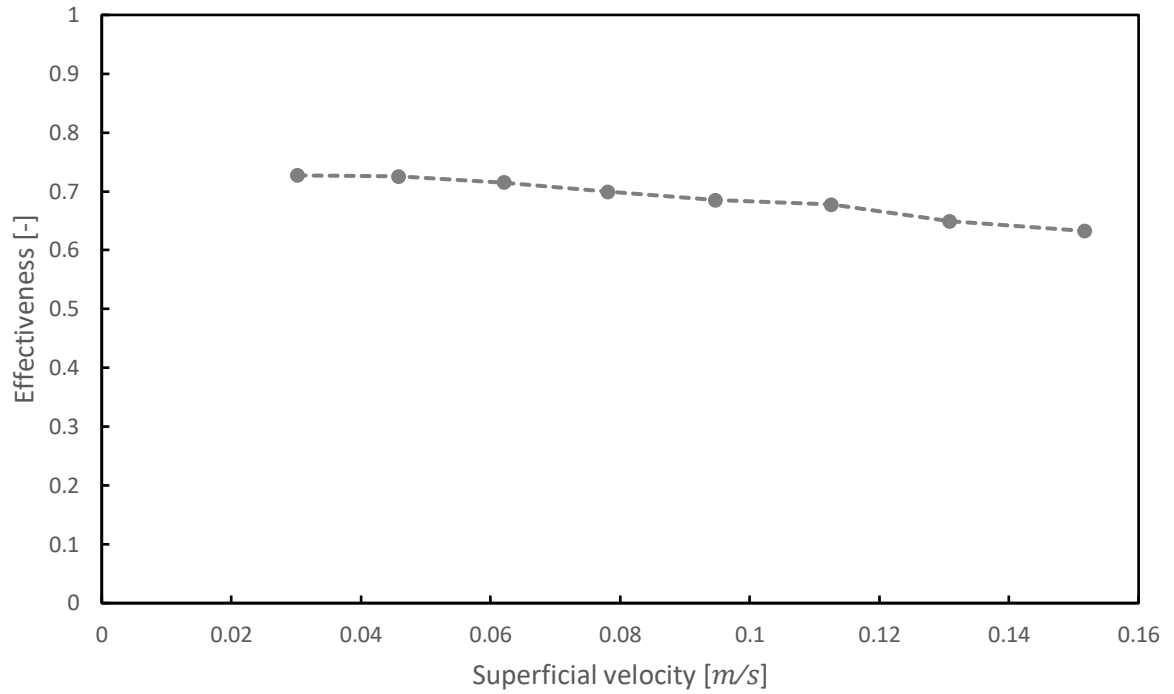


Figure 3-10 Effect of air superficial velocity on Effectiveness

Figure 3-10 illustrates the effect of the superficial velocity to the Effectiveness, where it is evidently revealed that the humidifier's effectiveness considerably decreases with the increase of superficial air velocity. From the definition of the effectiveness shown in Equation (3-22), the air outlet temperature decreases in the humidifier with the increase of superficial air velocity that yields to a decrease in the total humidifier effectiveness.

Effect of Air Superficial Velocity on Mass Transfer Coefficient

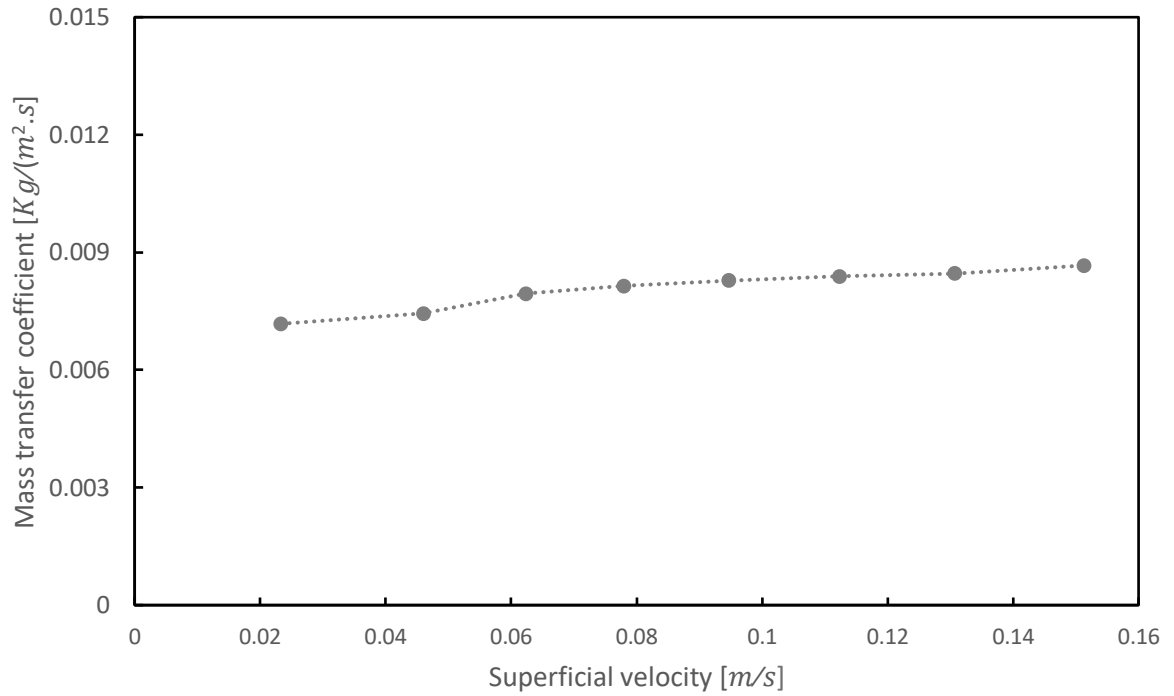


Figure 3-11 Effect of air superficial velocity on Mass transfer coefficient

Figure 3-11 shows that the higher the air superficial velocity, the higher mass transfer in the humidifier which is attributed to the air mass flow rate increase that will yield to an increase in the turbulent dissipation rate in the column; this can also be explained by the NTU correlation with the mass transfer coefficient shown in Equations (3-21) where it shows the more mass flow rate to the humidifier, the higher the mass transfer coefficient.

Effect of Column Liquid Height

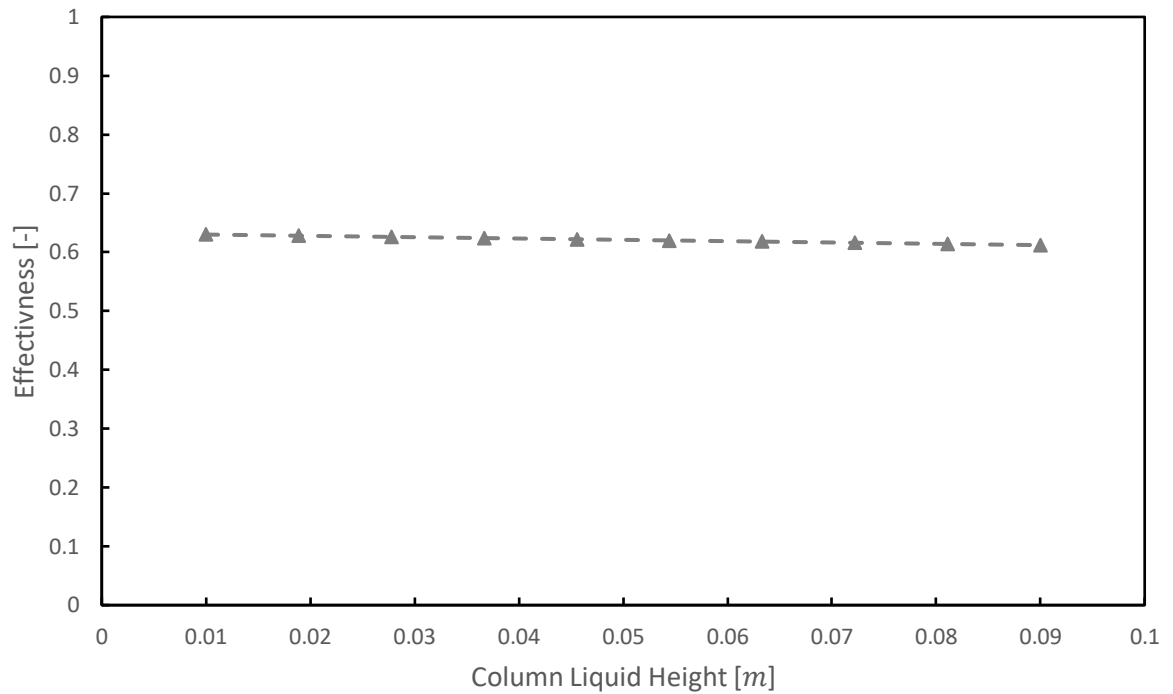


Figure 3-12 Effect of Column Liquid Height on total System Effectiveness

Figure 3-12 demonstrates the effect of varying the column liquid height on the total system effectiveness. The results revealed that there is an insignificant reduction of the effectiveness as the column liquid increases in a low scale of variance (few millimeters), and this can be described by understanding the length scale of the liquid circulation, which is an integral length of turbulence. Therefore, unless the column liquid height is reduced to a larger scale of variance, it will not show a significant effect on the bubble column Effectiveness. The liquid height is a very substantial factor when designing bubble column humidifiers in the humidification dehumidification desalination (HDH) system. The results agree with the reported result in the (3,10,2)

3.2.2 Influence of Design Parameters

Effect of Coil Length on Water Outlet Temperature in the Coil

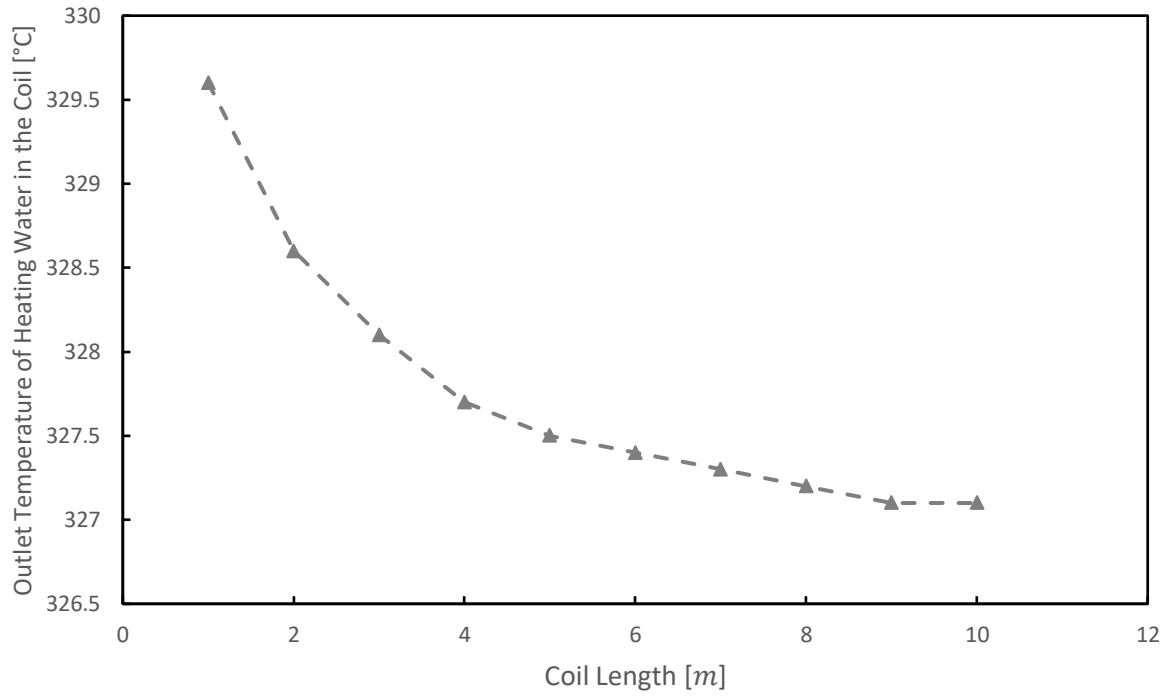


Figure 3-13 Effect of coil length on Water Outlet Temperature in the Coil

Coil Length plays a significant role in setting the heat transfer area. Figure 3-13 shows the effect of the length of the coil to the Hot water stream inside the coil. It is manifestly that the more length will offer more heat transfer area, which will yield to a more heat transfer between the coil running water and the water in the column. Consequently, the outlet temperature of the hot water in the coil will decrease.

Effect of Coil Outer Diameter on Water Outlet Temperature in the Coil

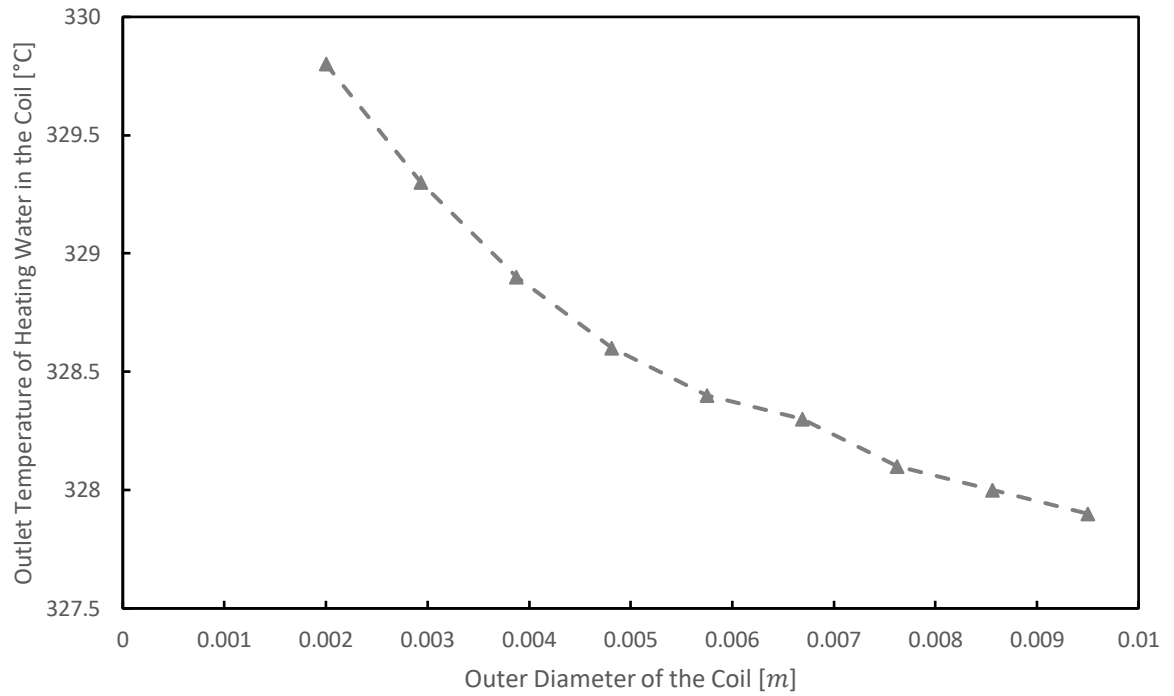


Figure 3-14 Effect of coil outer diameter on Water Outlet Temperature in the Coil

Figure 3-14 illustrates the influence of varying the outer diameter of the coil to the outlet temperature of the hot water stream in the coil. The outlet temperature of the hot water in the coil decreases considerably with larger outer coil diameter; this is attributed to the larger heat transfer area as a consequence of larger diameter.

Effect of Coil Outer Diameter on Reynolds Number

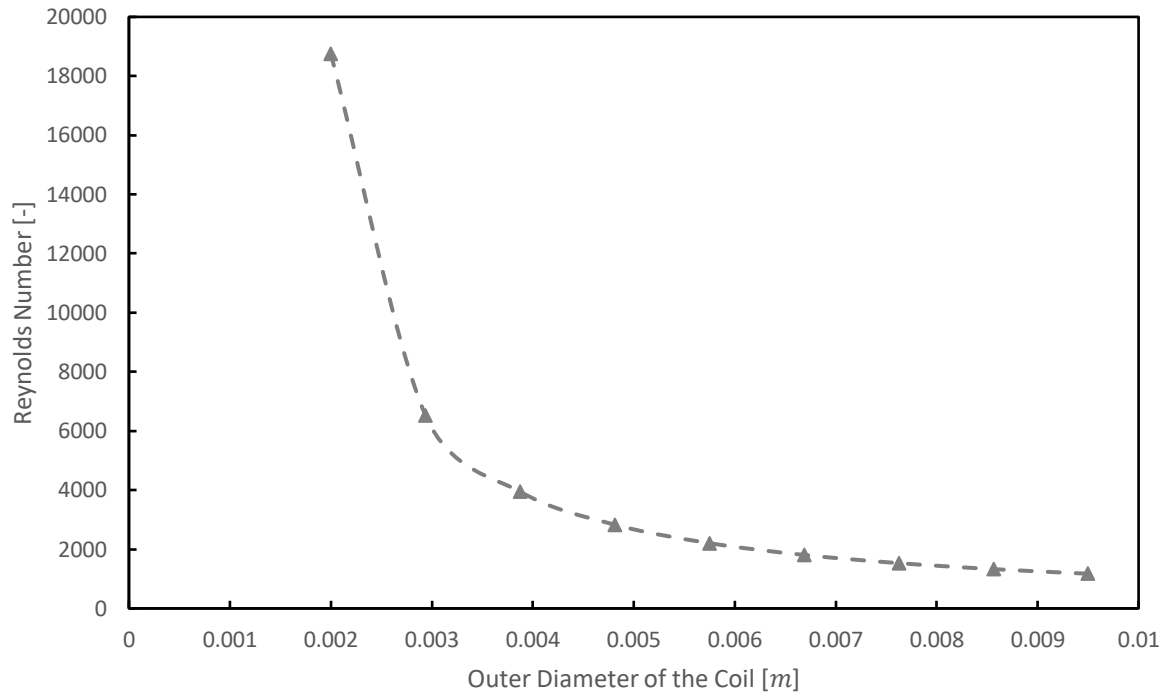


Figure 3-15 Effect of coil outer diameter on Reynolds number

Figure 13-15 illustrates the effect of the outer diameter on the Reynolds number, and it is evidently proven that the higher the outer diameter, the lower the velocity, provided that the water flow is constant, which can be described through the continuity theory. Consequently, Reynolds number will dramatically drop as the diameter increases.

As a conclusion, results revealed that in order to eliminate the issue of bubble adhesion to the hot water coil and avoid the interruption of the thermal balancing of the air bubbles in the bubble column humidifier, it is suggested to minimize the outer diameter of the heating coil while elongating the length of the coil to maintain the desired heat and mass transfer rates inside the bubble column humidifier. Hence, it is essential to balance between these design parameters to maintain the desired heat supplied by the coil

3.2.3 Validation of the Results

Results were validated and tested with Liu [10] model and found out to be quantitatively similar with a max error of 2.8%. Furthermore, the reported uncertainties for Liu [10] results were as follows, ± 0.00075 for mass transfer coefficient, $\pm 1.65\%$ for the effectiveness and ± 3.46 for the total heat, this will bridge the gaps of error percentage to a maximum error of 0.5% in the three factors mentioned above. Below figures illustrate the benchmark with Liu's results.

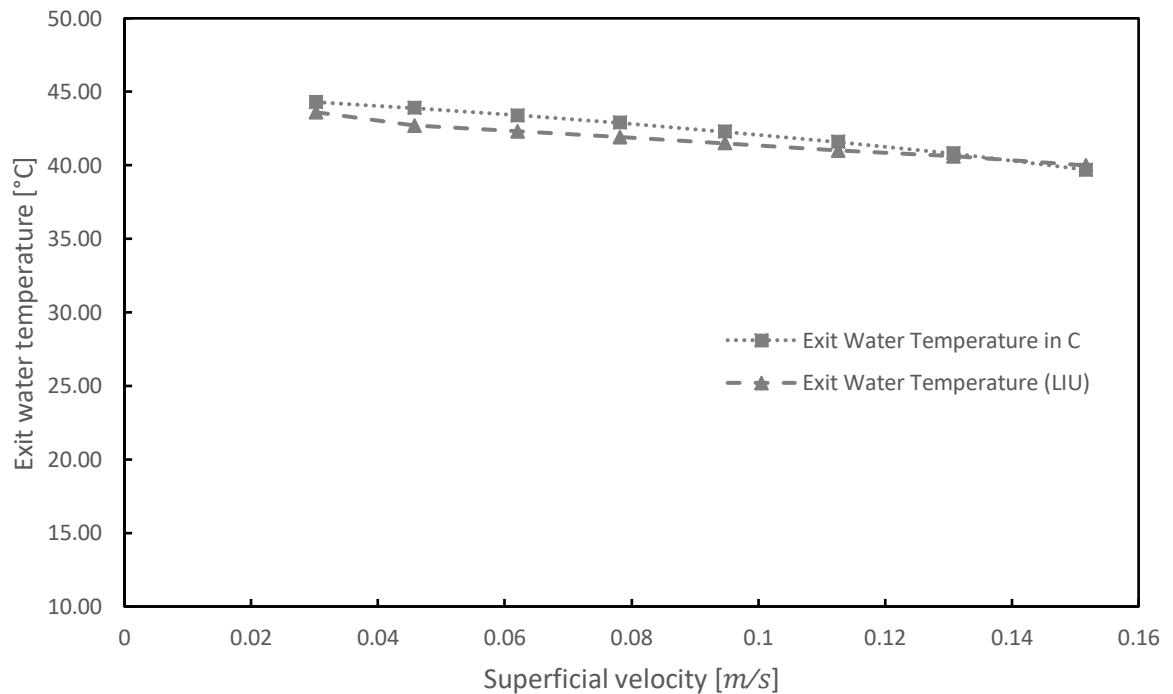


Figure 3-16 Validation of Effect of Superficial velocity on Exit Water Temperature

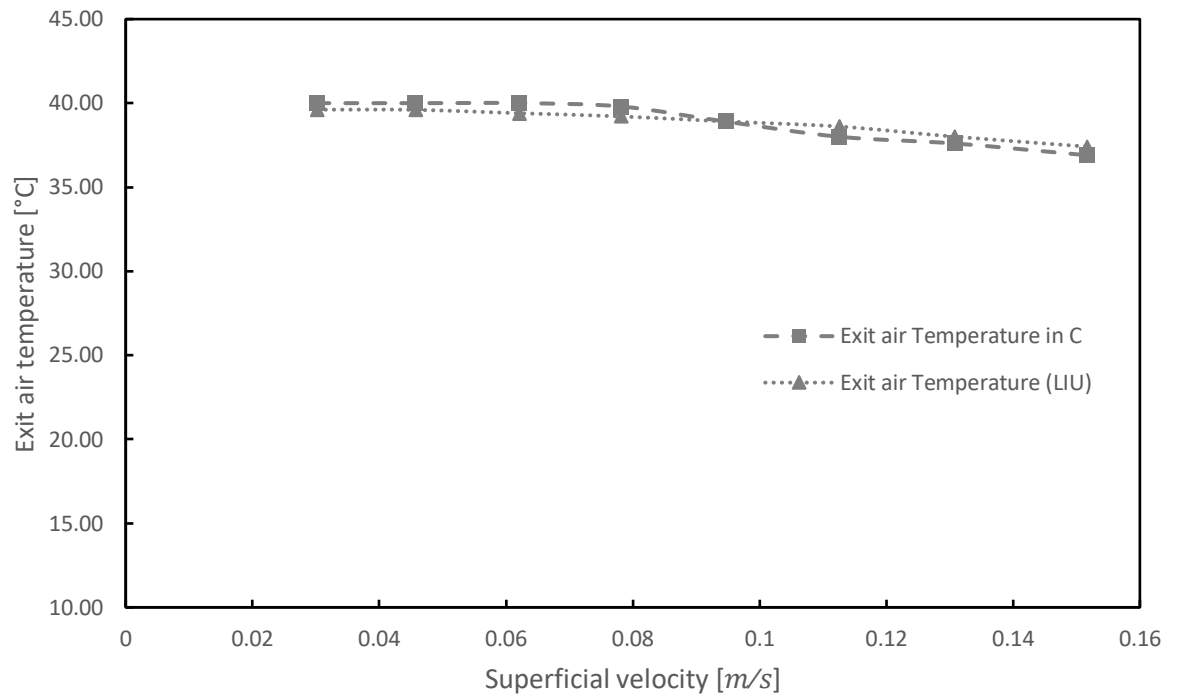


Figure 3-17 Validation of Effect of Superficial velocity on Exit Air Temperature

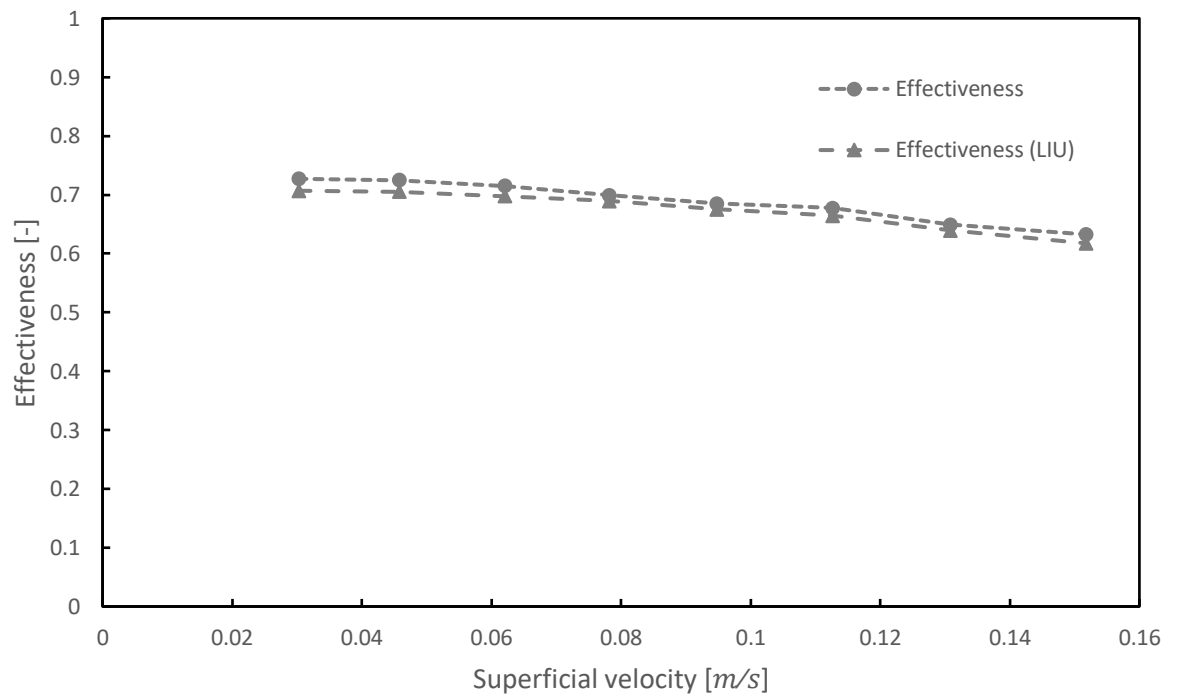


Figure 3-18 Validation of Effect of Superficial velocity on Effectiveness

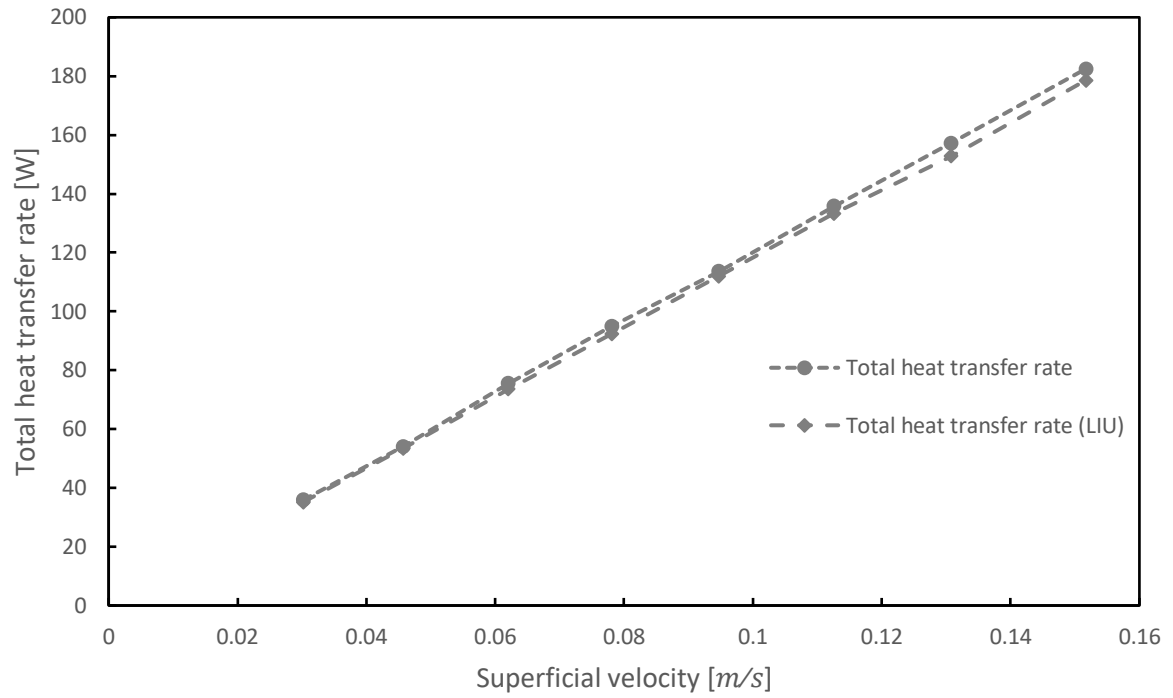


Figure 3-19 Validation of Effect of Superficial velocity on Total heat transfer rate

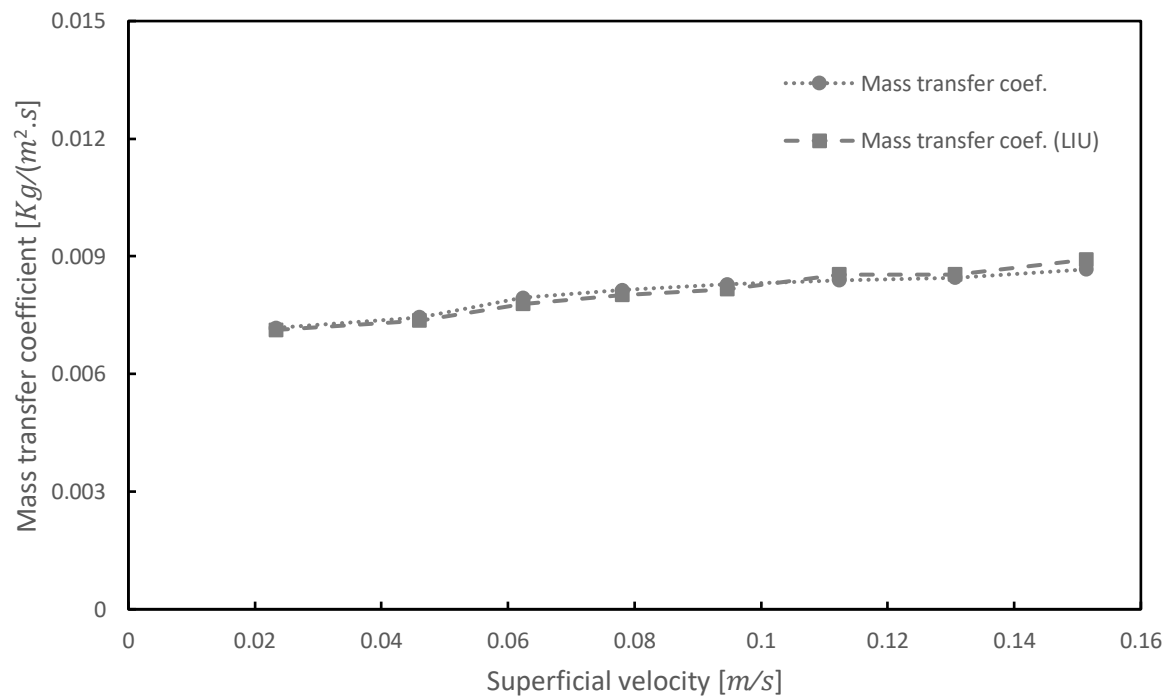


Figure 3-20 Validation of Effect of Superficial velocity on Mass transfer coefficient

3.3 Moving Streams

The section will shed light on the other model, which considers both streams in a moving form, as shown in figure 3-21, the heat and mass transfer will occur while both streams in direct contact with each other. Both latent and sensible heat will take place as described in the thermal resistance shown in figure 3-22

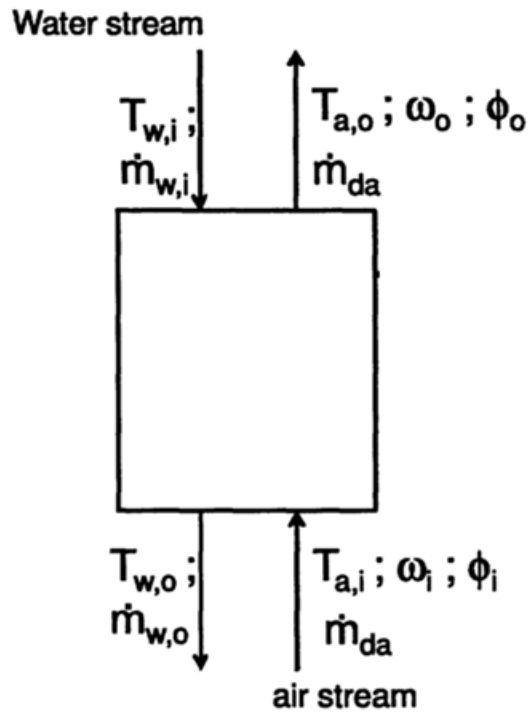


Figure 3-21 Schematic of the moving streams bubble column humidifier [4]

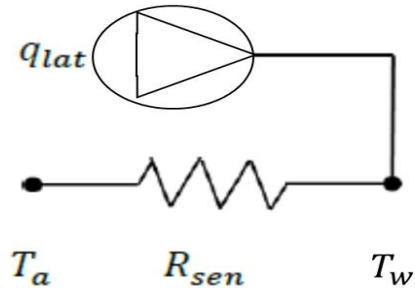


Figure 3-22 Thermal Resistance of the moving Stream model [4]

Adopting the same assumptions aforementioned in the previous model, the analytical model will be almost the same except for the mass and energy balance since we have moving streams in this model, hence, applying the mass balance conservation theory, it is a fact that mass of the air will remain the same throughout the process. Therefore,

$$\dot{m}_{a,i} = \dot{m}_{a,o} = \dot{m}_a \quad (3-35)$$

So, the mass balance conservation is as follows,

$$\dot{m}_{w,i} = \dot{m}_{w,o} + \dot{m}_a(\omega_{a,o} - \omega_{a,i}) \quad (3-36)$$

Applying energy balance for the bubble humidifier, in terms of enthalpies, the correlation will be as follows,

$$\dot{m}_{w,i}h_{w,i} = \dot{m}_{w,o}h_{w,o} - \dot{m}_a(h_{a,i} - h_{a,o}) \quad (3-37)$$

By merging equations (3-35) and (3-36) with the previous model analytical formulations, the model considered fully defined and can be modeled for further analysis.

3.3.1 Results and Discussion

The aforementioned equations are modeled and solved numerically by utilizing Engineering Equation solver (EES) software, moist air and liquid water physical properties are available in the software. The below section shows a comparison between the stagnant water model and the moving streams model for a single stage bubble column humidifier to highlight the differences between both models. The comparison will include the effectiveness of the system, the achieved humidity ratio and the air temperature.

Comparison Between the Two Models on the Total Effectiveness

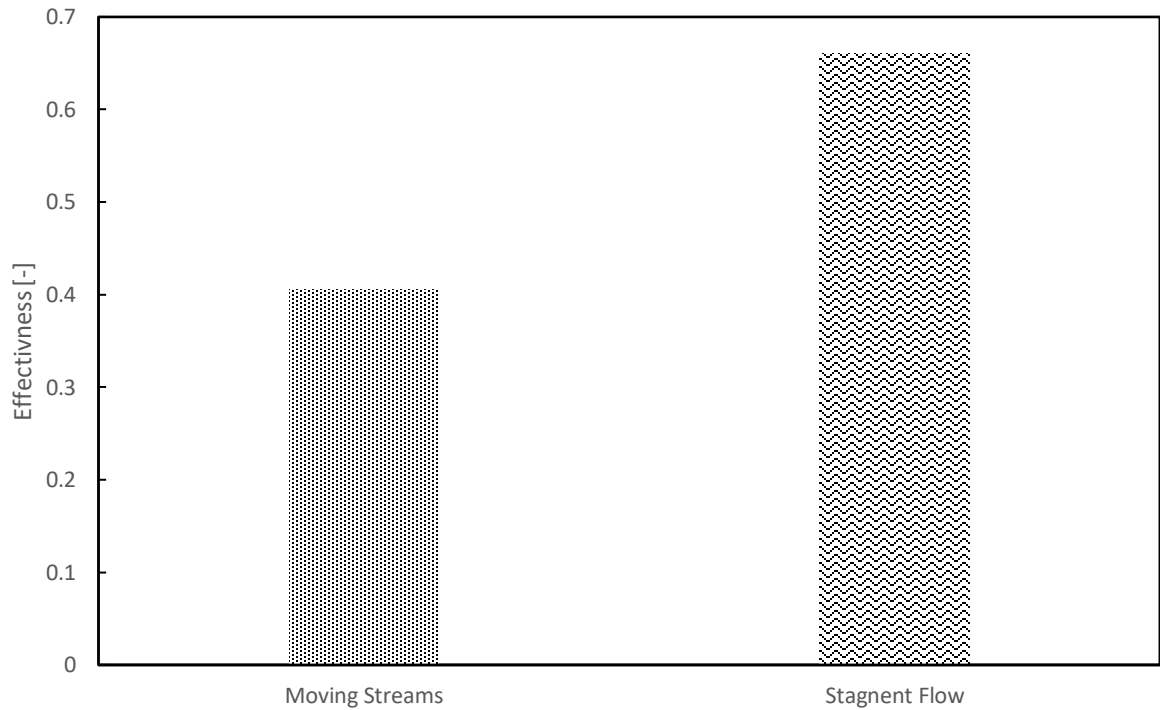


Figure 3-23 Comparison between the effectiveness of both models. Boundary conditions: $T_{ai} = 15\text{ }^{\circ}\text{C}$;

$$P = 101.3\text{ kPa}; T_{wi} = 45\text{ }^{\circ}\text{C}; H = 0.02$$

Figure 3-23 shows the comparison between the two models on the total effectiveness; obviously, the stagnant flow outweighs the effectiveness of the moving streams model, and this can be attributed to the coil inserted inside the bubble column humidifier that maintains the thermal balance across the water in the column which provide stability to the air to absorb more heat. In other words, if the water inside the stagnant water model has been divided into small segments, provided the coil is dissipating all the heat to water in a uniform manner, water segments will have a fixed temperature, in contrast, the moving streams segments will have a different temperature. Therefore, system effectiveness will be higher in the stagnant water model.

Comparison Between the Two Models on the Outlet Air Temperature

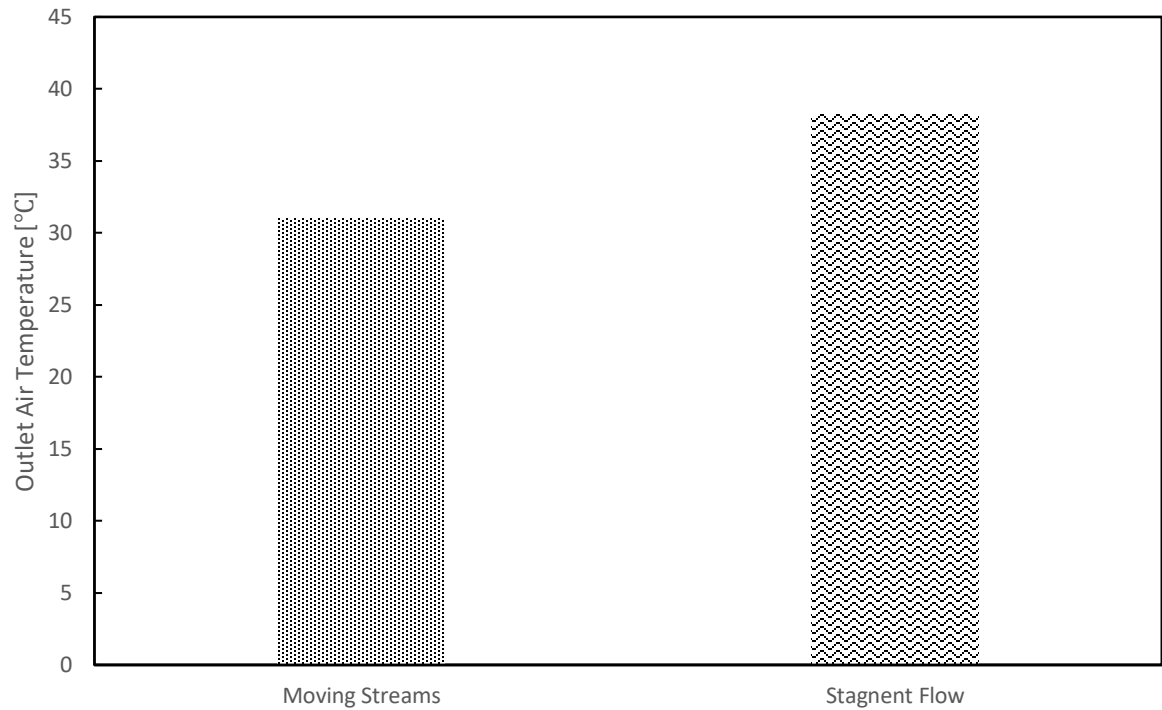


Figure 3-24 comparison of the Outlet Air Temperature for both models. Boundary conditions:

$$T_{ai} = 15\text{ }^{\circ}\text{C}; P = 101.3\text{ kPa}; T_{wi} = 45\text{ }^{\circ}\text{C}; H = 0.02$$

Figure 3-24 illustrates a comparison of the air outlet temperature for both models. Results show that the outlet air temperature achieved through the stagnant water model is higher than the one achieved in the moving streams. As described in the previous figure, the stagnant water model will have almost a uniform water temperature across the bubble column humidifier, which will lead to a higher heat transfer rate that will lead to a higher outlet air temperature for the stagnant water model.

Comparison Between the Two Models on the Outlet Humidity Ratio

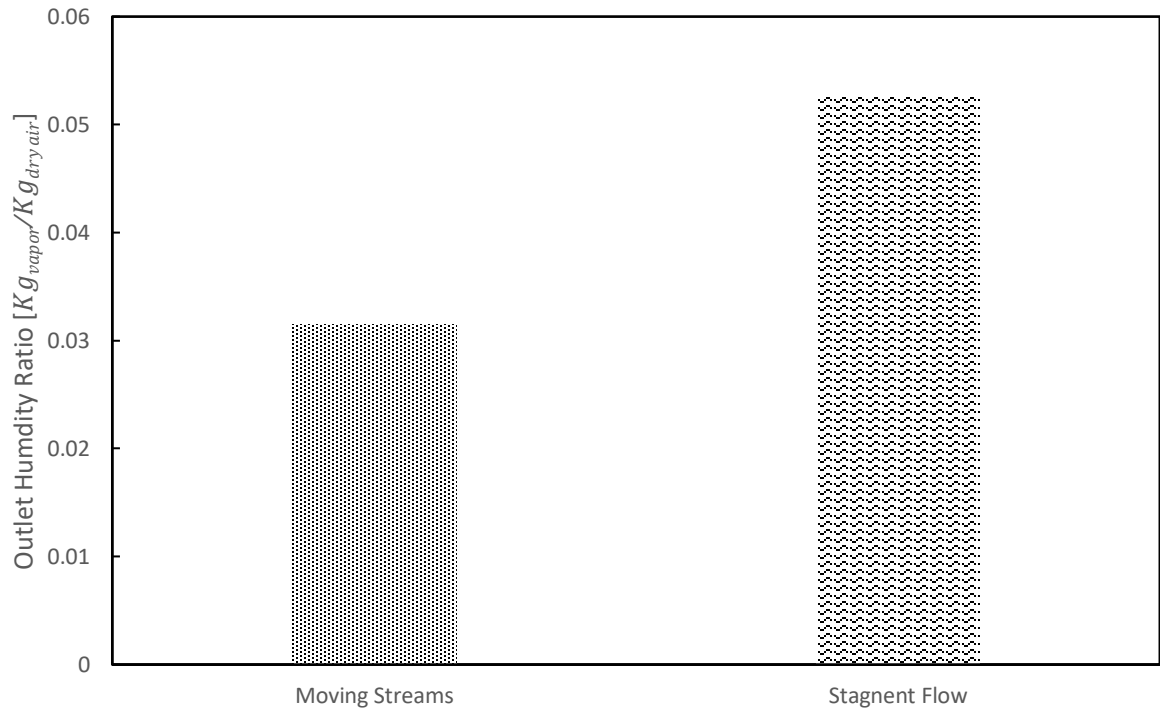


Figure 3-25 comparison of the Outlet Humidity Ratio for both models. Boundary conditions:
 $T_{ai} = 15\text{ }^{\circ}\text{C}$; $P = 101.3\text{ kPa}$; $T_{wi} = 45\text{ }^{\circ}\text{C}$; $H = 0.02$

Figure 3-25 shows the accomplished outlet air humidity ratio for both models. Results illustrate that the stagnant water model output air absorbs more humidity which can be dehumidified in the upstream of the HDH system to extract the fresh water, this can be described by the attribution of the high heat transfer rate in the stagnant water model which is directly proportional to the mass transfer coefficient as described in Lewis factor equation mentioned in eq. (3-7). Thus, a higher mass transfer rate will be accomplished, which will lead to a higher humidity absorbed in the stagnant water model.

3.4 Chapter Conclusion

In this chapter, a mathematical model has been developed for the bubble column humidifier for two different models. These models are the bubble column humidifier model that has stagnant water with a heating coil to maintain the required energy and the other model where the air and water streams are moving. The first model was considered as the base model for the other models that will be studied throughout this thesis. The base model was validated with Liu [10] and found to be accurate with a maximum error percentage of 2.8%; moreover, with consideration of Liu's [10] reported uncertainty, the variation will be eliminated for some of the results. The mathematical models covered the heat and mass transfer and were solved through Engineering Equation Solver (EES).

The preliminary results for the base model investigated the influence of operational and design parameters. The operational parameters are the effect of superficial velocity variance on outlet air and water temperature, effectiveness and mass transfer coefficient. The effect of water height on the system effectiveness was investigated too. The results showed that the superficial velocity increment has a positive impact on the total heat transfer and the mass transfer coefficient. However, the increase of the air superficial velocity influences the bubble column humidifier effectiveness and the outlet temperature of air and water negatively. Moreover, the results also revealed that height has an insignificant effect on total system effectiveness. For the design parameter, different design factors were investigated to gauge the influence of different parameters on the outlet temperature of the air with the consideration of the optimum coil geometry. These design parameters are the effect of coil outlet diameter and length variation on the coil outlet water temperature and the achieved Reynolds number. Results showed that in order to eliminate

the bubble adhesion to the hot water coil and avoid the interruption of the thermal balancing of the air bubbles in the bubble column humidifier, it is suggested to minimize the outer diameter of the heating coil while elongating the length of the coil to maintain the desired heat and mass transfer rates inside the bubble column humidifier. Hence, it is essential to balance between these design parameters to maintain the desired heat supplied by the coil. The moving streams model was also studied and compared with the base model to highlight the areas of strength for both models in addition to setting the optimum model. The comparison covered the accomplished effectiveness of the system, the achieved humidity ratio and the air temperature. Overall, the results showed that for a single stage bubble column humidifier, the stagnant water model supersedes the moving streams model and this is mainly due to the coil inserted in the bubble column humidifier that maintains the thermal balance across the water in the column which provide stability to the air to absorb more heat, hence the stagnant water model is the optimum system. Nevertheless, due to the complexity of the design and practicality, the moving streams model will be considered as the default design for the purpose of multi-staging that will be discussed in the next chapter.

Chapter 4

Multi Staging Influence

In this chapter, an analytical investigation is performed in order to evaluate the performance of a multi-stage bubble column humidifier for the moving streams model. Series and Parallel configurations will be investigated to assess the effect of different configurations on the bubble column humidifier performance. The effect of multi-staging on different performance parameters will be explored, such as humidity ratio, total humidifier effectiveness, the effectiveness of each stage and outlet air temperature with different configurations. Furthermore, the influence of different operation parameters on the humidifier performance will be investigated, and this includes the effect of varying inlet water temperature, inlet air temperature, the mass ratio (m_r), and inlet relative humidity on the outlet condition of both air and water stream as well as the total system effectiveness. The mass ratio aforementioned is the mass ratio of the water to the air in the bubble column humidifier. The ultimate goal of this section is to shed light on the influence of adding more layers of bubble column humidifier on the total system effectiveness. This chapter will address the required number of stages to achieve an effective bubble column humidifier provided that the system operates in a preset initial boundary condition. To further explore this subject, in this chapter, a comprehensive parametric study is performed to discover the required number of stages to achieve a highly effective system with the variation of the initial boundary conditions. This chapter also discussed and provided a comparison for the operation and maintenance of both series and parallel multi-staged bubble column humidifiers.

4.1 Multi-Staging Effect with Parallel and series Configuration

This Section will present the effect of multi-staging on the performance of the moving streams model mentioned earlier in chapter 3, as well as investigating the bubble humidifier configuration effect (Parallel or Series). The main difference between the series and parallel configurations are the mass flow rate for the series configured humidifier which is set to be four times the one for the parallel humidifier. In addition, at each stage, the inlet water temperature is maintained constant for the parallel multi-staged columns while it is changing in the series one. In this section, we will investigate the influence of multi-staging on humidity ratio, effectiveness, effectiveness of each stage and outlet air temperature.

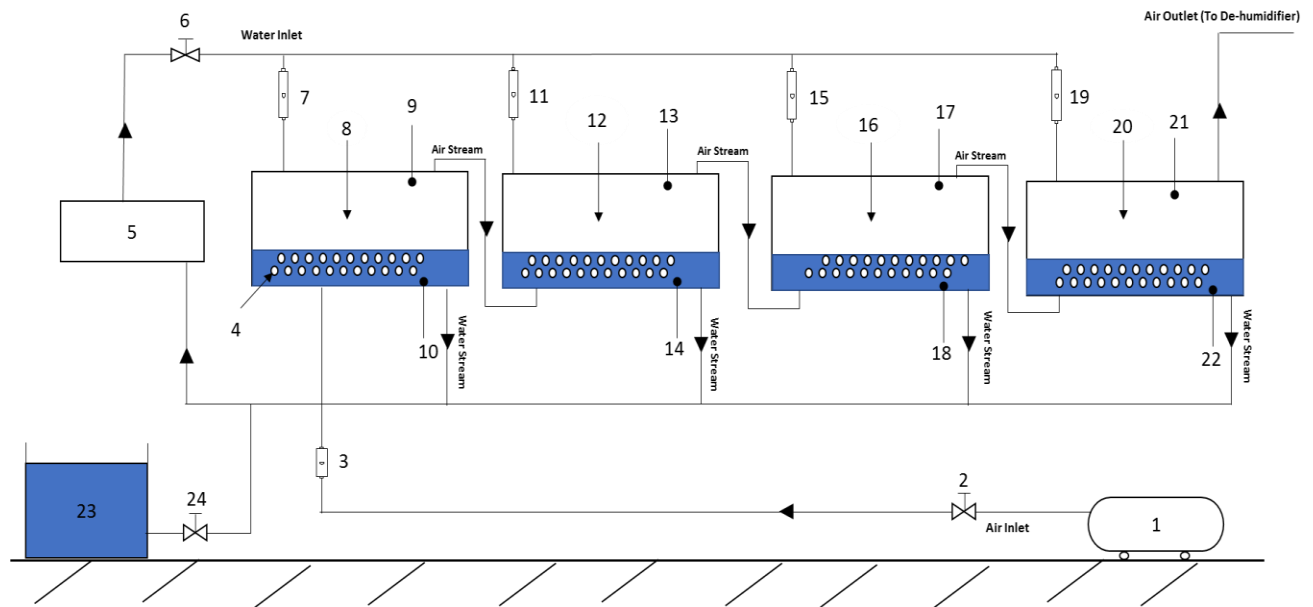


Figure 4-1 Schematic of a Multi-Stage Humidifier in a Parallel Configuration

- (1) Air Compressor, (2,6,24) Valve, (3,7,11,15,19) Rotameter, (4) Air bubbles, (5) water Heater, (8) first Stage of the bubble column humidifier, (9,10,13,14,17,19,21,22) Thermocouples, (12) second Stage of the bubble column humidifier, (16) Third Stage the bubble column humidifier, (21) fourth stage of the bubble column humidifier, (23) Make-up tanks

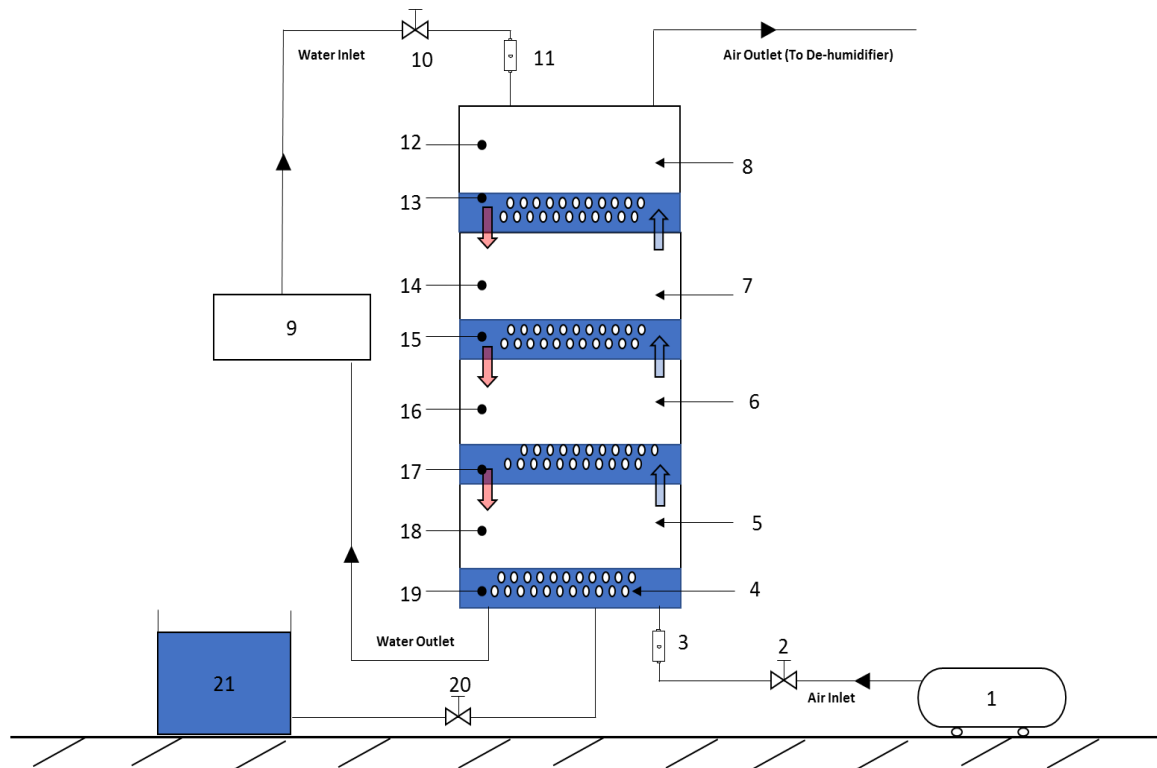


Figure 4-2 Schematic of a Multi-Stage Humidifier in a Series Configuration

- (1) Air Compressor, (2,10,20) Valve, (3,11) Rotameter, (4) Air bubbles, (5) first Stage of the bubble column humidifier, (6) second Stage of the bubble column humidifier, (7) Third Stage the bubble column humidifier, (8) fourth stage of the bubble column humidifier, (9) water Heater, (12,13,14,15,16,17,18,19) Thermocouples, (21) Make-up tanks

Effect of Number of Stages on the Total Effectiveness

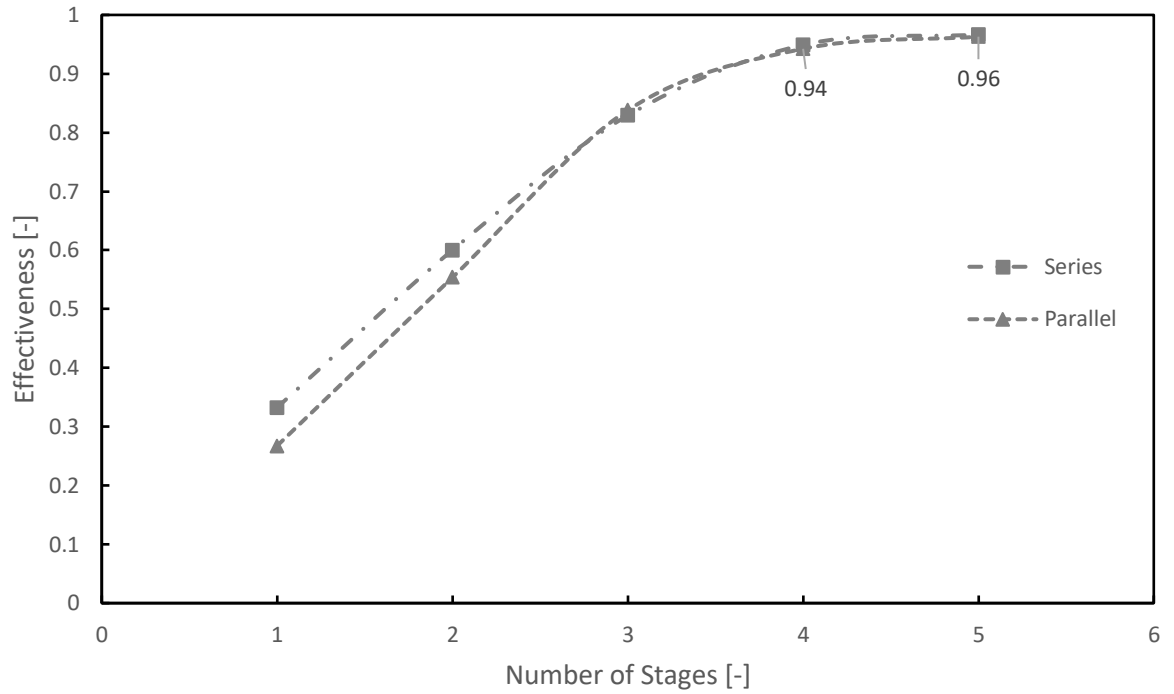


Figure 4-3 Effect of Number of Stages on The Total Effectiveness. Boundary Conditions:

$$T_{ai} = 33 \text{ }^{\circ}\text{C}; P = 101.3 \text{ kPa}; T_{wi} = 60 \text{ }^{\circ}\text{C}; H = 0.02; m_{r,parallel} = 2.8; \phi_i = 40\%$$

Figure 4-3 answers the central question of this thesis, which is the number of stages suggested for optimum effectiveness with consideration of the financial feasibility. Results in figure 4-3 show that the effectiveness is highly affected by adding more stages to the HDH system. The results illustrated that the single stage of HDH system following the moving streams model could accomplish 27% and 33% effectiveness for parallel and series configuration respectively, with the addition of two more stages of bubble humidifiers in any configuration, the effectiveness can be tripled to reach to almost 82%. At the fourth stage, the bubble column humidifier will reach an effectiveness of 94%. Furthermore, the results show that it not recommended to add the fifth stage since the effectiveness will slightly increase to be 96%, which is not financially feasible. So, it is suggested to maintain

only four stages since the effectiveness will not change considerably after the fourth stage. Both series and parallel configuration have almost the same effectiveness with a slightly higher percent for the series configuration.

Effect of Number of Stages on Outlet Humidity Ratio

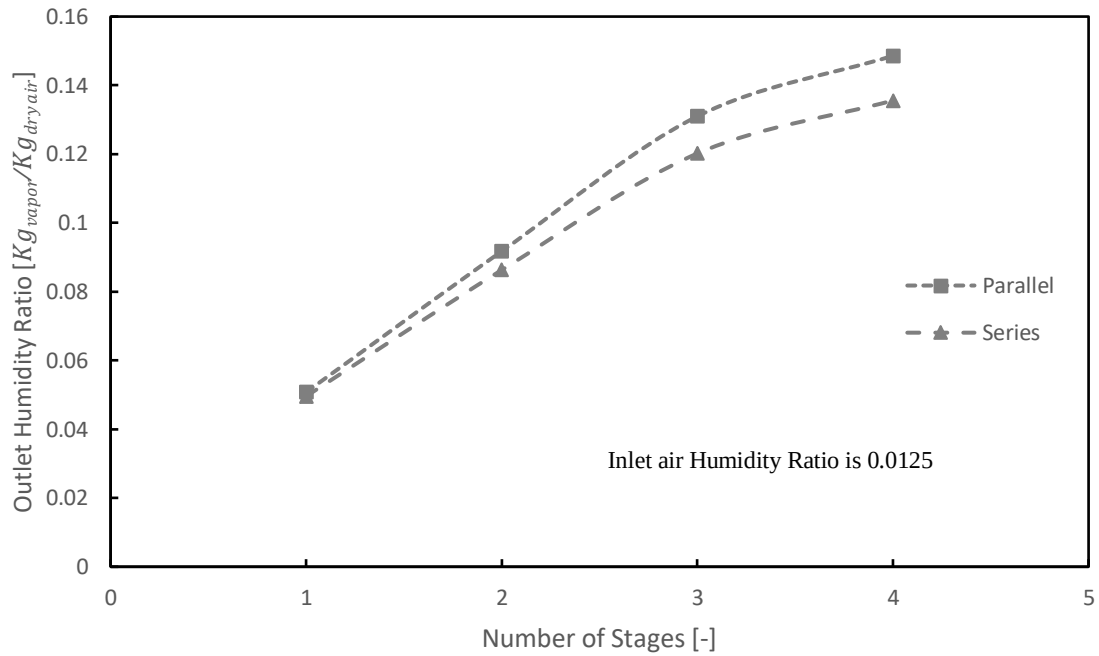


Figure 4-4 effect of number of stages on the outlet humidity ratio. Boundary conditions

$$T_{ai} = 33\text{ }^{\circ}\text{C}; P = 101.3\text{ kPa}; T_{wi} = 60\text{ }^{\circ}\text{C}; H = 0.02; m_{r,parallel} = 2.8; \phi_i = 40\%$$

Results in figure 4-4 show that influence of multi-staging to the outlet air humidity ratio, the results show that both series and parallel configurations have almost the same results for all the four stages. Results revealed that four stages are enough to almost triple the results of the outlet humidity ratio for both configurations, the humidifier at the fourth stage can produce air with a humidity ratio of almost 0.15 compared with a 0.05 at the first stage. The humidity ratio increment noticed in the multi-stages bubble column humidifier is a sign of a highly effective system since a higher value of humidity ratio leads to a higher amount of water produced of the HDH system.

Effect of Number of Stages on the Outlet Air Temperature

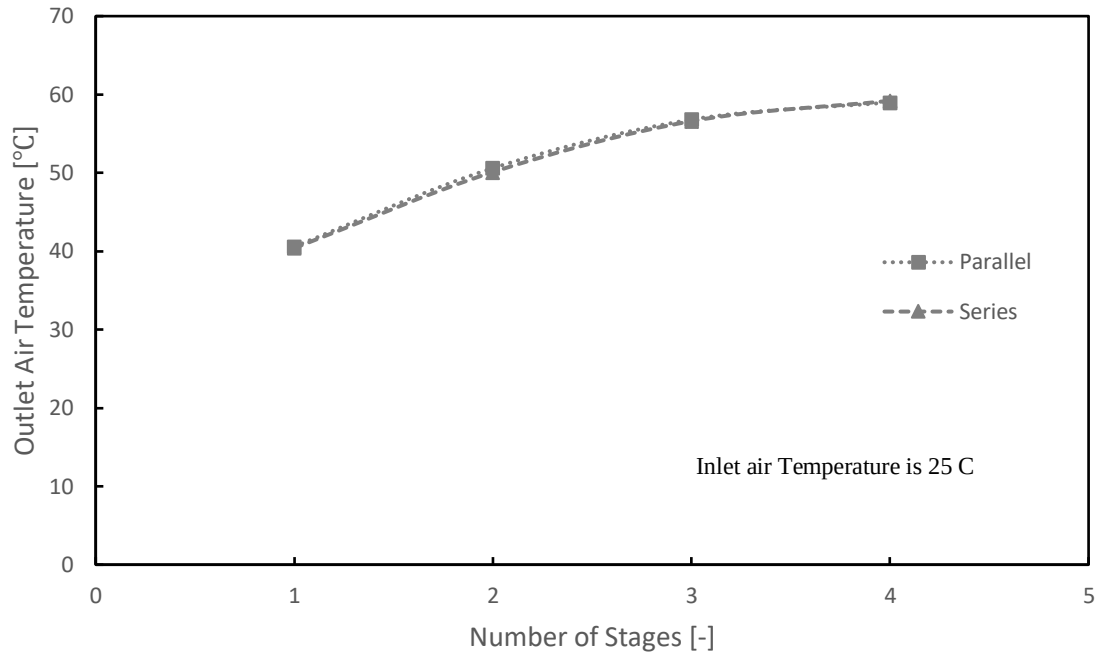


Figure 4-5 Effect of multi-staging on the outlet air temperature. Boundary conditions:

$$T_{ai} = 33\text{ }^{\circ}\text{C}; P = 101.3\text{ kPa}; T_{wi} = 60\text{ }^{\circ}\text{C}; H = 0.02; m_{r,parallel} = 2.8; \varphi_i = 40\%$$

Results in figure 4-5 show that the influence of multi-staging to the outlet air temperature, the results show that both series and parallel configurations have almost the same results for all the four stages. Multi-staging influences the air temperature significantly, the results show that the achieved outlet air temperature in the first stage is 40 C, whereas the temperature has significantly increased at the fourth stage to be almost 59 C.

Effect of Number of Stages on Each Stage Effectiveness

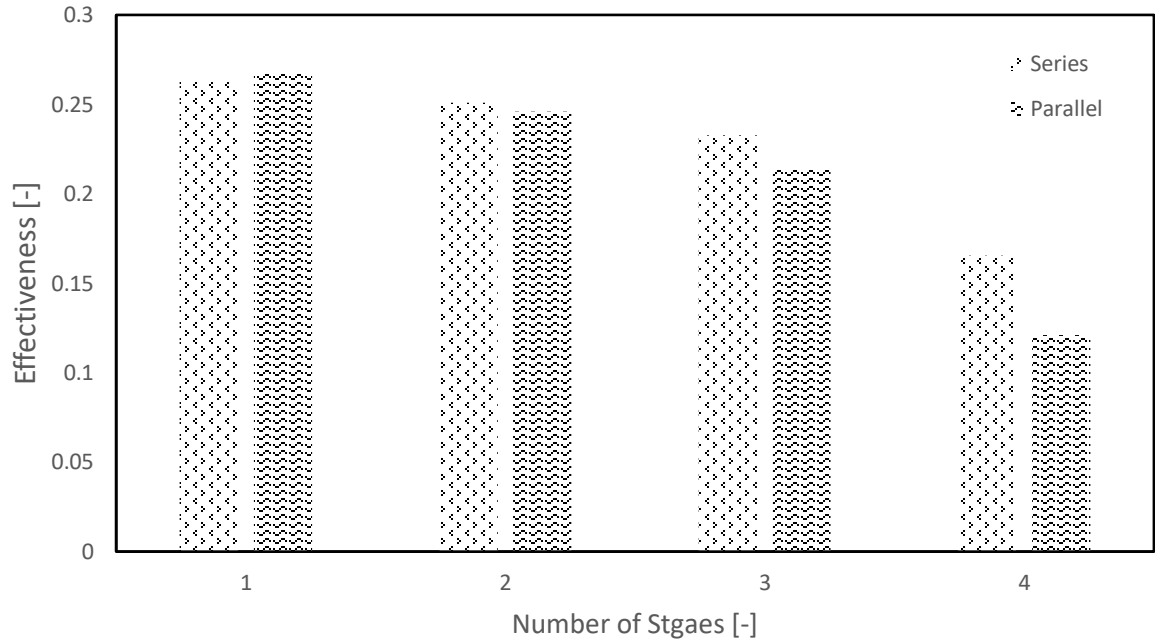


Figure 4-6. Effect of multi-staging on each stage effectiveness. $T_{ai} = 33\text{ }^{\circ}\text{C}$; $P = 101.3\text{ kPa}$; $T_{wi} = 60\text{ }^{\circ}\text{C}$; $H = 0.02$; $m_{r,parallel} = 2.8$; $\phi_i = 40\%$

Figure 4-6 reveals the influence of the multi-staging on the effectiveness of each stage. The results represent a comparison of each stage achieved enthalpy with the maximum enthalpy of the system to illustrates the effectiveness behavior as we increase the number of stages. The smooth reduction of both parallel and series configured bubble column humidifiers can be described by understanding the potential energy of each stage. For the first stage, the air is at its lowest temperature, whereas the water will be at its peak temperature; thus, the potential heat transfer is higher compared to the other stages where the air temperature starts to increase the water temperature starts to decrease. Stated differently, the availability of heat and mass transfer decreases with the increase of stages.

4.1.1 Validation of the Results

The data has been tested and validated against a similar in nature research., This research was done by Munteshari [2] where he investigated a multi-stages bubble column humidifier experimentally, his work was done considering three stages for the humidifier setup. Below is a comparison between this thesis results that was investigated analytically with the experimental results obtained by Munteshari [2]

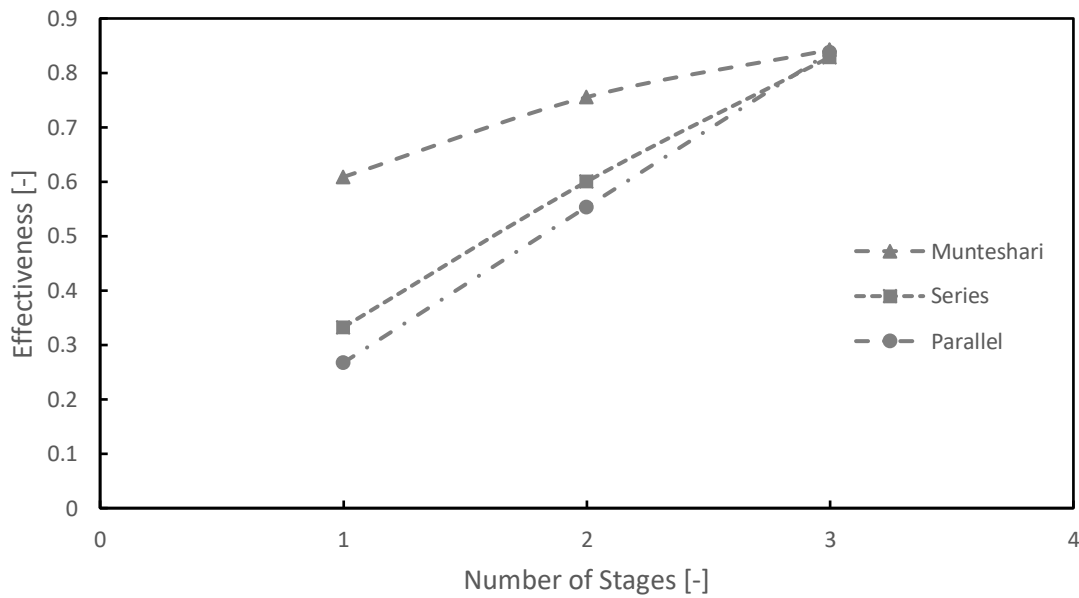


Figure 4-7 The effect of multi-staging on the total system effectiveness. Boundary conditions:
 $T_{ai} = 33\text{ }^{\circ}\text{C}$; $P = 101.3\text{ kPa}$; $T_{wi} = 60\text{ }^{\circ}\text{C}$; $m_r = 2.8$ $\phi = 40\%$ $H = 0.02\text{ m}$

Figure 4-7 illustrates a qualitative similarity between the results obtained by Munteshari [2] and the results analyzed by this thesis. The results show that at the third stage, the result is found to be almost quantitatively similar with an accomplished three stages system effectiveness of almost 84 %, this validation raises the confidence level of the obtained results by this thesis since it can be compared with the results found experimentally by Munteshari. The difference between the results found in the first and second stages can be

attributed to some assumed parameters. Furthermore, the uncertainty reported by the Munteshari was $\pm 2.52\%$. Thus, the gap of the error percentage will be reduced to an acceptable level of confidence.

4.2 Operational Parameters Variation Influence

This section will investigate the influence of different parameters on the performance of the modeled bubble column humidifier that consists of four stages configured in both series and parallel configurations. These parameters are the effect of the inlet air and water temperatures variation, the effect of the mass ratio variation and the influence of the inlet relative humidity variation. These parameters were studied to explore the effect on the outlet condition of both air and water, the overall effectiveness.

Effect of Water Inlet Temperature on Exit Water and Air Temperatures

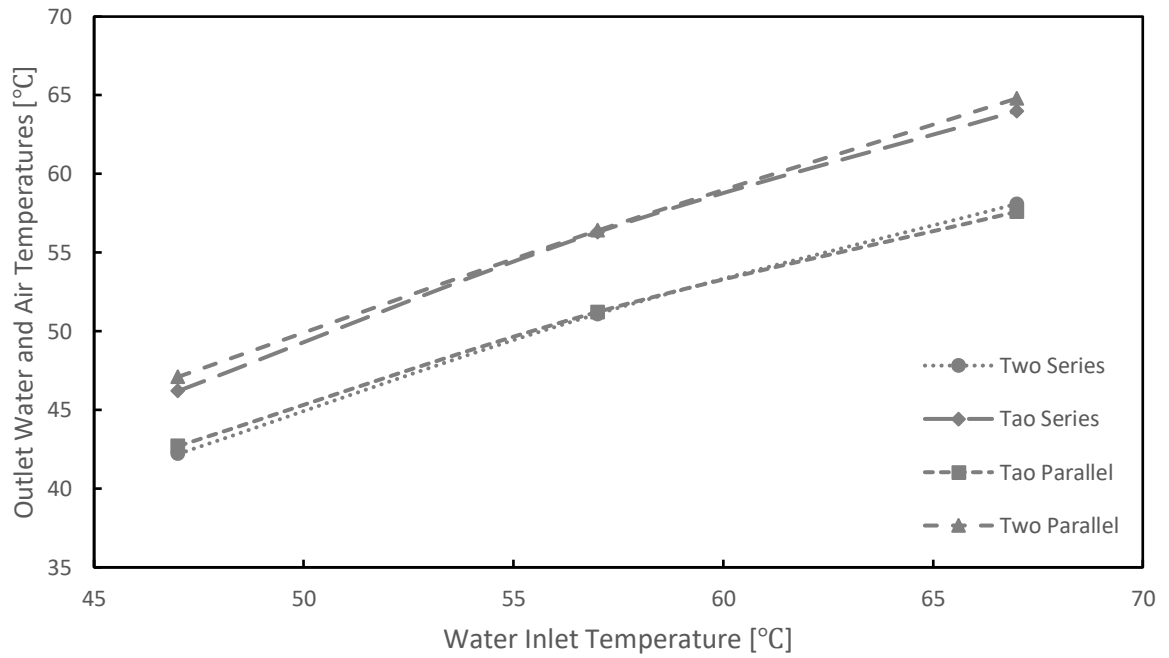


Figure 4-8. Effect of Water Inlet Temperature on Exit Water and Air Temperatures. Boundary conditions: $T_{ai} = 33\text{ }^{\circ}\text{C}$; $P = 101.3\text{ kPa}$; $H = 0.02$; $m_{r,parallel} = 2.8$; $\varphi_i = 40\%$

Figure 4-8 shows the influence of varying the water inlet temperature on the outlet conditions of the bubble column humidifier for both configurations. It is evidently proven that the higher the inlet water temperature, the higher the outlet water and air temperatures. Overall, for both configurations, the inlet water temperature variance has a considerable effect on the exit condition of water and air. As described earlier, provided that the air inlet temperature is kept constant, the main attribution to this increase is the increase of the heat and mass transfer availability due to water inlet temperature increase.

Effect of Air Inlet Temperature on Exit Water and Air Temperatures

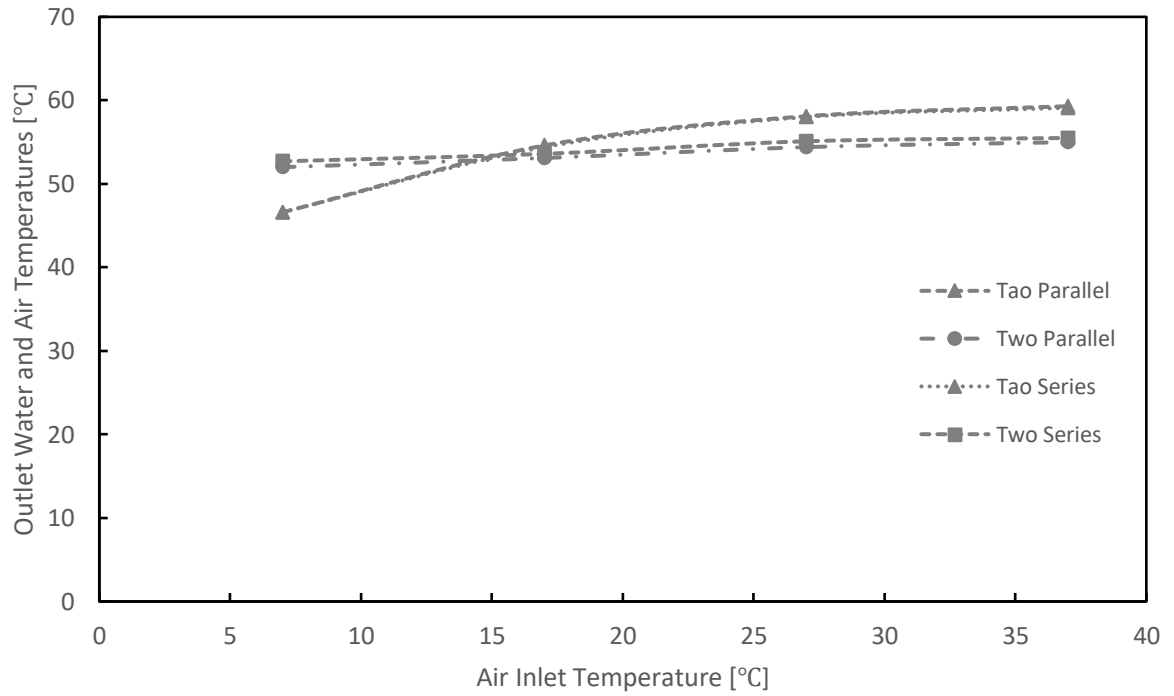


Figure 4-9. Effect of air Inlet Temperature on Exit Water and Air Temperatures. Boundary conditions:
 $P = 101.3kPa$; $T_{wi} = 60\text{ }^{\circ}\text{C}$; $H = 0.02$; $m_{r,parallel} = 2.8$; $\varphi_i = 40\%$

Figure 4-9 shows the influence of varying the air inlet temperature to the outlet conditions of the bubble column humidifier for both configurations. Evidently, the higher the inlet air temperature, the higher the outlet air temperature as shown in figure 4-8, Both parallel and series configurations will have almost the same results. For the water outlet temperature, it is shown that there is a considerable increase which is directly proportional with the increase of the air inlet temperature since the increase of the air inlet temperature will minimize the opportunity for the air to extract heat from the water provided that the water inlet temperature is fixed.

Effect of Mass Ratio on Exit Water and Air Temperatures

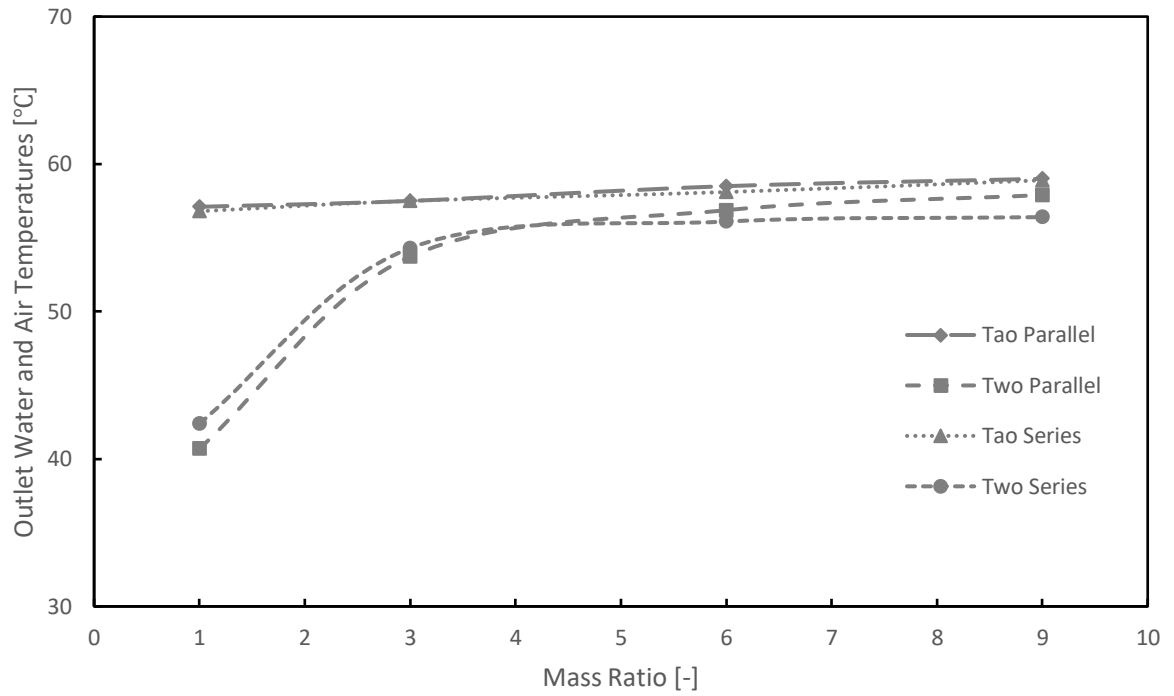


Figure 4-10. Effect of mass ratio on Exit Water and Air Temperatures. Boundary conditions:

$$T_{ai} = 33 \text{ }^{\circ}\text{C}; P = 101.3 \text{ kPa}; T_{wi} = 60 \text{ }^{\circ}\text{C}; H = 0.02; \varphi_i = 40\%$$

Figure 4-10 demonstrates the influence of varying the mass ratio on exit water and air temperatures. Provided that the mass flow rate of the air is kept constant and the inlet temperatures of air and water are constant, the variation of the mass flow will have a significant effect on the increment of the water outlet temperature for both configurations, particularly at a mass ratio between 1 and 4. The results show also a minimal effect of the mass ratio to the water temperature at a mass ratio number higher than 4. However, the effect is found to be positively minimal to the air outlet temperature.

Effect of Inlet Relative Humidity on Exit Water and Air Temperatures

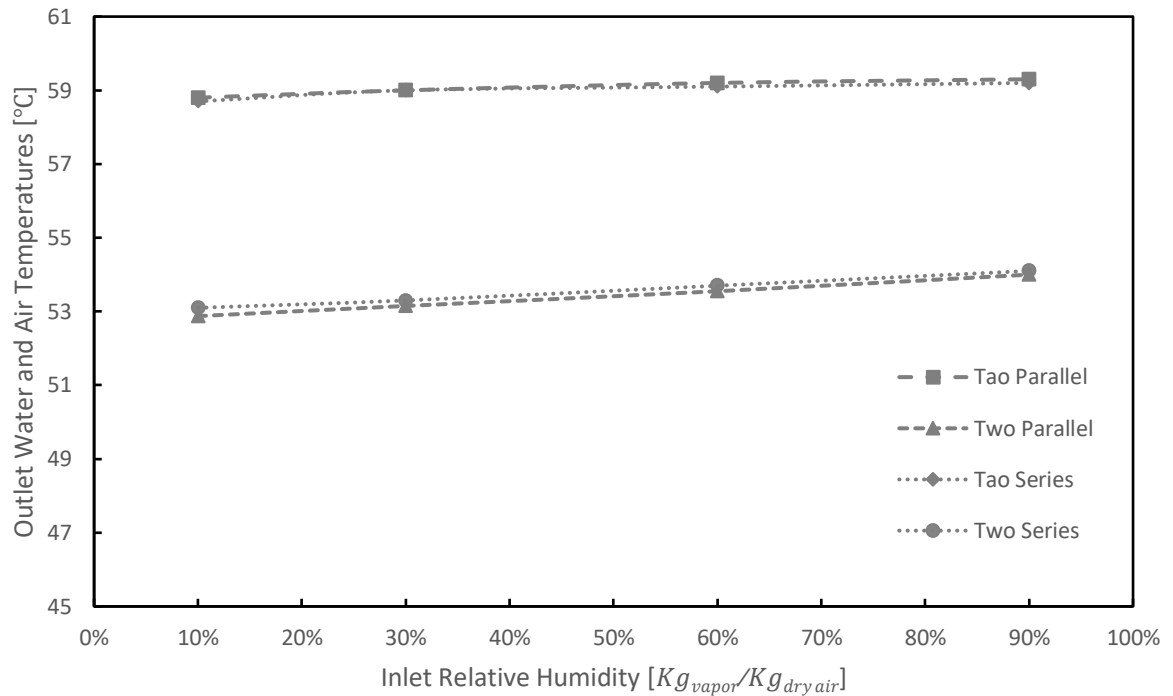


Figure 4-11. Effect of Inlet Relative Humidity on Exit Water and Air Temperatures. Boundary conditions: $T_{ai} = 33\text{ }^{\circ}\text{C}$; $P = 101.3\text{ kPa}$; $T_{wi} = 60\text{ }^{\circ}\text{C}$; $H = 0.02$; $m_{r,parallel} = 2.8$

Figure 4-11 indicates the influence of the Inlet Relative Humidity to the outlet water and air temperatures. The results show that the higher the relative inlet humidity of the air, the higher the exit air and water temperatures. The variation of the relative inlet humidity is enough to raise the air temperature to 3 degrees from the initial temperature. On the other side, varying the relative inlet humidity has an insignificant effect on the increment of the water outlet temperature with 0.4 degrees for both series and parallel configured bubble column humidifier. Overall, the variation of relative inlet humidifier has a slightly considerable effect on the exit condition of the air and water.

Effect of Water Inlet Temperature on Total Humidifier Effectiveness

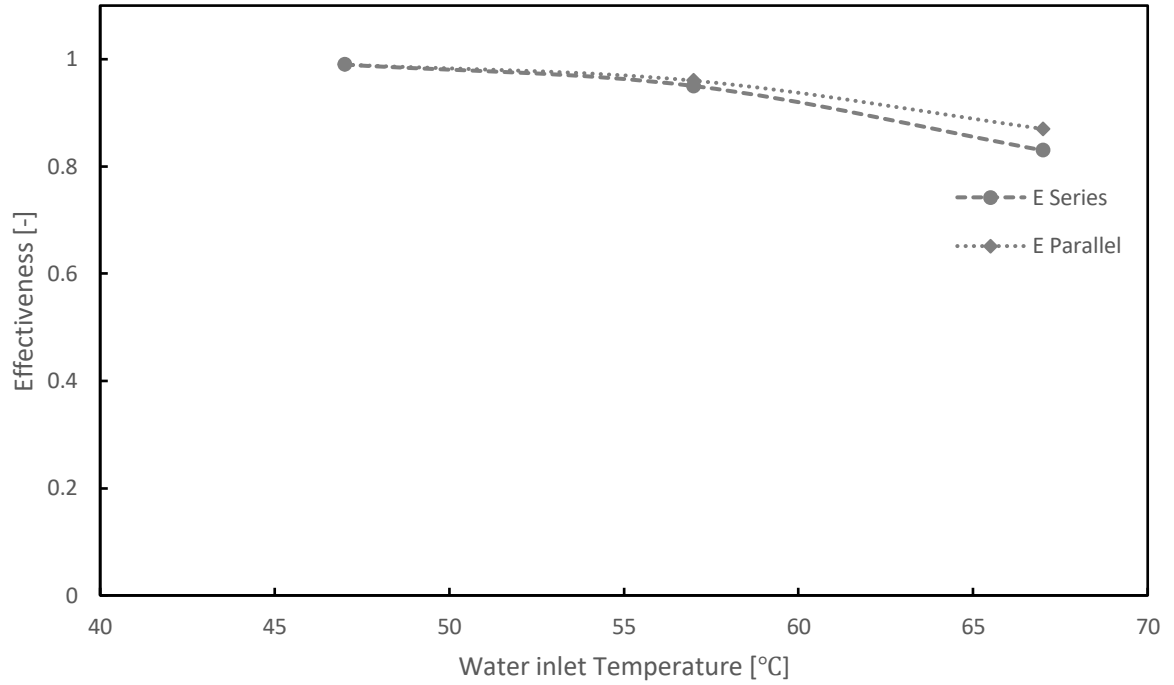


Figure 4-12. Effect of water inlet temperature on total humidifier effectiveness. Boundary conditions:
 $T_{ai} = 33\text{ }^{\circ}\text{C}$; $P = 101.3\text{ kPa}$; $H = 0.02$; $m_{r,parallel} = 2.8$; $\varphi_i = 40\%$

The results shown in Figure 4-12 illustrate the influence of water inlet temperature variation on total humidifier effectiveness. It is evidently shown that the effectiveness in both configurations is impacted negatively with the increase of the inlet water temperature. This can be attributed to the definition of the effectiveness, in this work, effectiveness has been defined in equation (3-22) to be the actual enthalpy achieved compared to the maximum achievable enthalpy that is calculated at the inlet water temperature, accordingly, as the inlet water temperature increases the effectiveness decreases.

Effect of Air Inlet Temperature on Total Humidifier Effectiveness

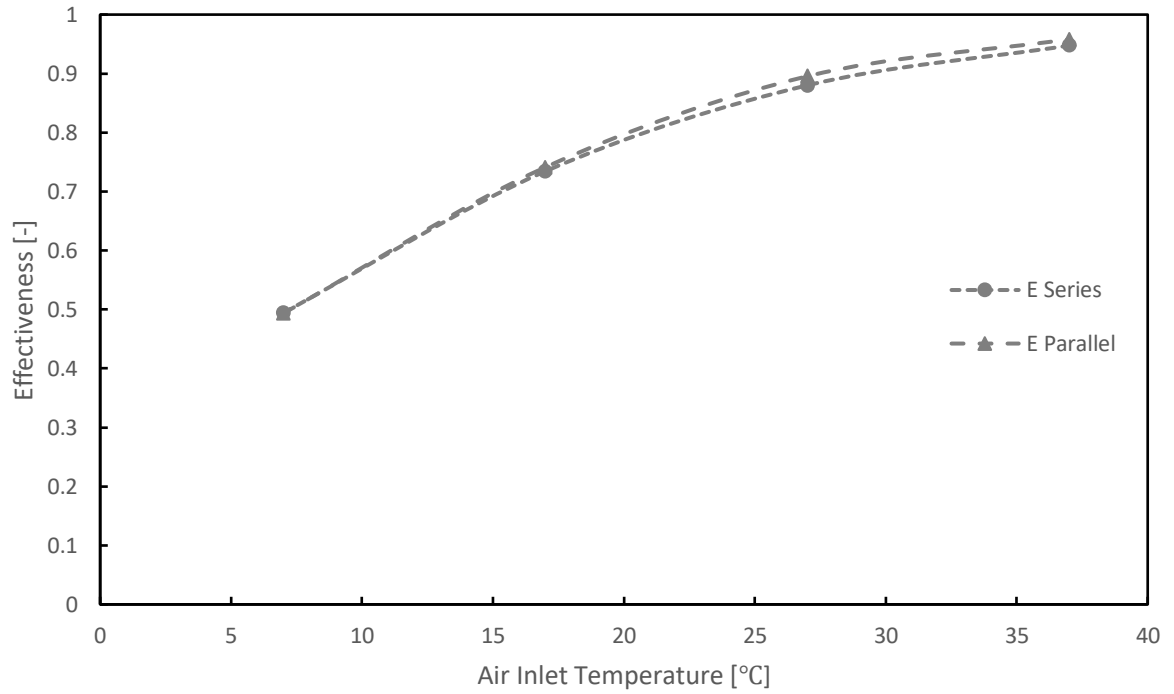


Figure 4-13 Effect of Air Inlet Temperature on Total Humidifier Effectiveness. Boundary conditions:
 $P = 101.3kPa$; $T_{wi} = 60\text{ }^{\circ}\text{C}$; $H = 0.02$; $m_{r,parallel} = 2.8$; $\varphi_i = 40\%$

Results shown in Figure 4-13 indicate the effect of air inlet temperature variation on total humidifier effectiveness. In contrast to the results shown in figure 4-12, the bubble column humidifier total effectiveness is enhanced with the increase of the air inlet temperature, the is due to the same reason aforementioned in the previous results of figure 4-12, where the air inlet temperature increment increases the enthalpy achieved, consequently, that the effectiveness in both configurations increases. The parallel configuration has a more considerable improvement compared to the series configuration.

Effect of Mass Ratio on Total Humidifier Effectiveness

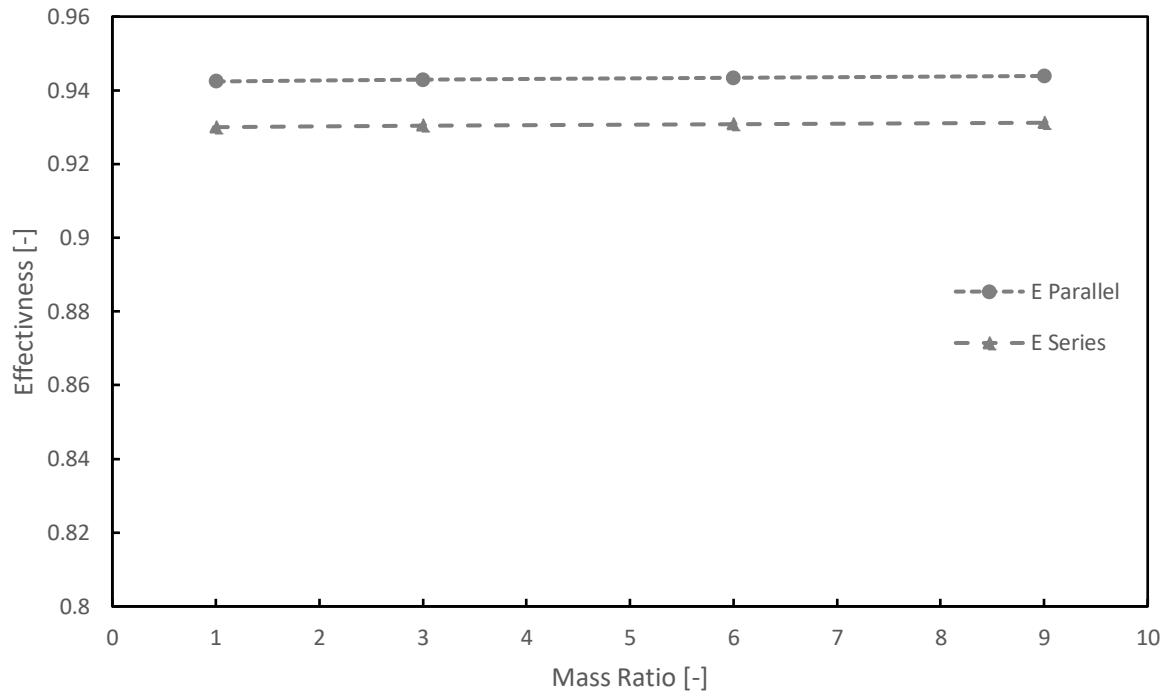


Figure 4-14. Effect of Mass Ratio on Total Humidifier Effectiveness. Boundary conditions:

$$T_{ai} = 33\text{ }^{\circ}\text{C}; P = 101.3\text{ kPa}; T_{wi} = 60\text{ }^{\circ}\text{C}; H = 0.02; \varphi_i = 40\%$$

Figure 4-14 reveals the influence of mass ratio on total humidifier effectiveness. As described in the results of figure 4-10, provided that the mass flow rate of the air is kept constant and the inlet temperatures of air and water are constant. The mass ratio will insignificantly influence the air outlet temperature in a positive manner, which accordingly will lead to a minimal impact on the air achieved enthalpy; thus, the total system effectiveness will increase insignificantly.

Effect of Inlet Relative Humidity on Total Humidifier Effectiveness

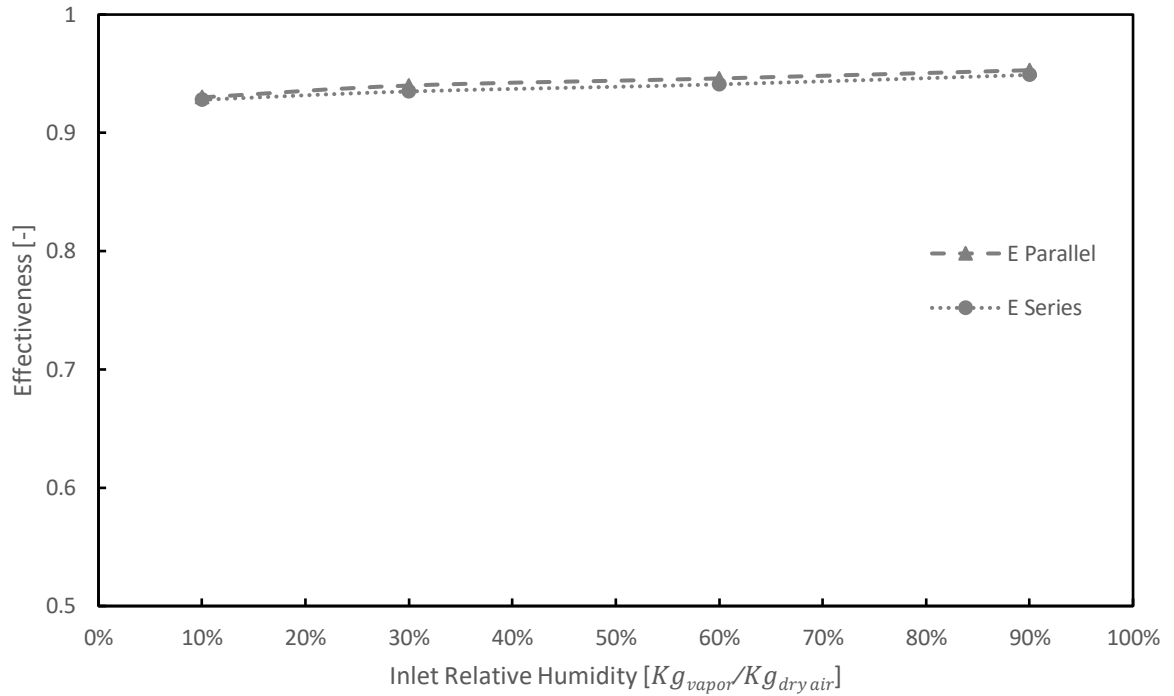


Figure 4-15. Effect of inlet relative humidity on Total Humidifier Effectiveness. Boundary conditions:
 $T_{ai} = 33\text{ }^{\circ}\text{C}$; $P = 101.3\text{ kPa}$; $T_{wi} = 60\text{ }^{\circ}\text{C}$; $H = 0.02$; $m_{r,parallel} = 2.8$

The results shown in Figure 4-15 illustrates the effect of inlet relative humidity on total humidifier effectiveness. Results reveal that the air inlet relative humidity has a very minimal effect on the effectiveness by approximately 0.02 increment both series and parallel configured humidifier. This can be described as shown in figure 4-11, where the maximum temperature increment due to varying the inlet air relative humidity is almost 3 degrees, which counts for a considerably low change in the enthalpy that will lead to a very minimal change in the effectiveness. The influence of the inlet air relative humidity on the performance of humidifier is an essential aspect as it signposts the optimum performance operating conditions for its possible integration with a dehumidifier. Additionally, the inlet air

to the system is ambient air, which might vary throughout the year. Therefore, the performance of the humidifier was analyzed at a different relative humidity of the inlet air.

4.2.1 Required Number of stages for each Parameter

In the previous sections, the required number of stages to achieve high performance with consideration of financial feasibility were investigated and found to be four (4) stages that can achieve 94% effectiveness provided the initial boundary conditions used in the model. Consequently, this parametric study will focus on the required number of stages to achieve the 94% effectiveness with the variation of the initial boundary conditions

Figure 4-16 shows that the required number of stages increases with the decrease of the air inlet temperature; evidently, this can be described by understanding the potential heat and mass transfer that increases with the decrease of the air inlet. Consequently, more stages are needed to raise the temperature to the desired temperature that will lead the system anticipated effectiveness. In the same concept, if the inlet water temperature increase, which is the hot stream, the potential heat and mass transfer increases, which means that there is a need for more stages to recover that potential heat transfer that leads to the desired effectiveness as shown in figure 4-17. In figures 4-18 and 4-19, the results revealed that the mass ratio and air inlet relative humidity increment would have a minimal effect to the required number of stages.

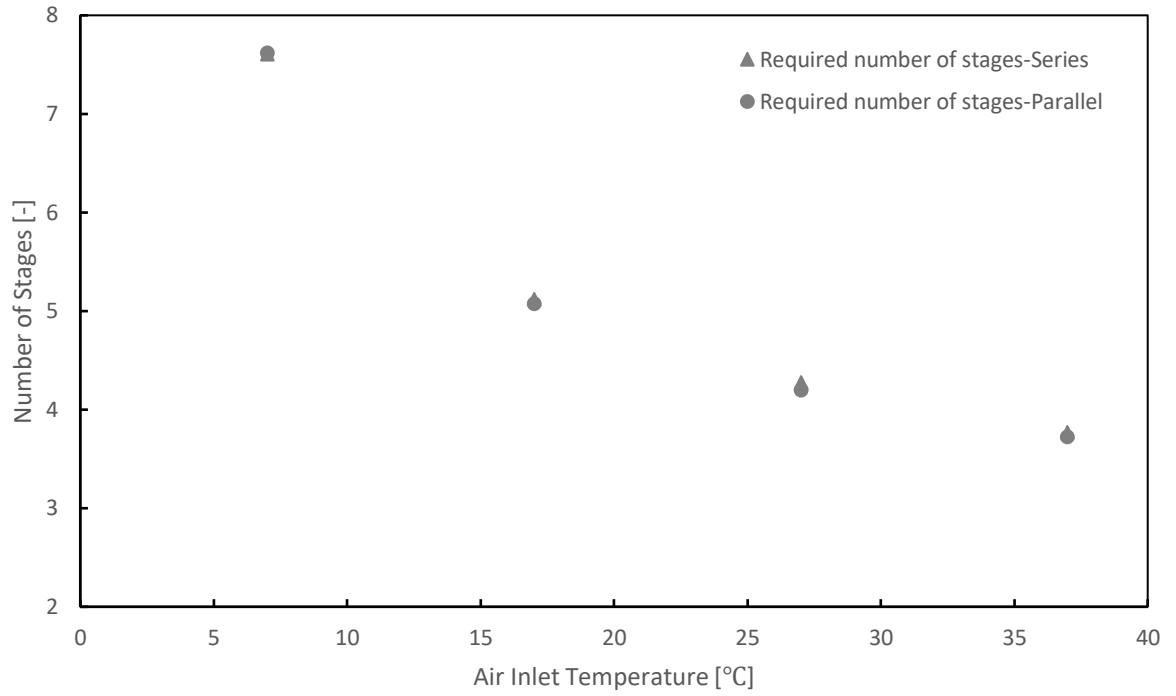


Figure 4-16. Effect of Air Inlet Temperature on number of stages. Boundary conditions:

$$P = 101.3kPa; T_{wi} = 60\text{ }^{\circ}C; H = 0.02; m_{r,parallel} = 2.8; \varphi_i = 40\%$$

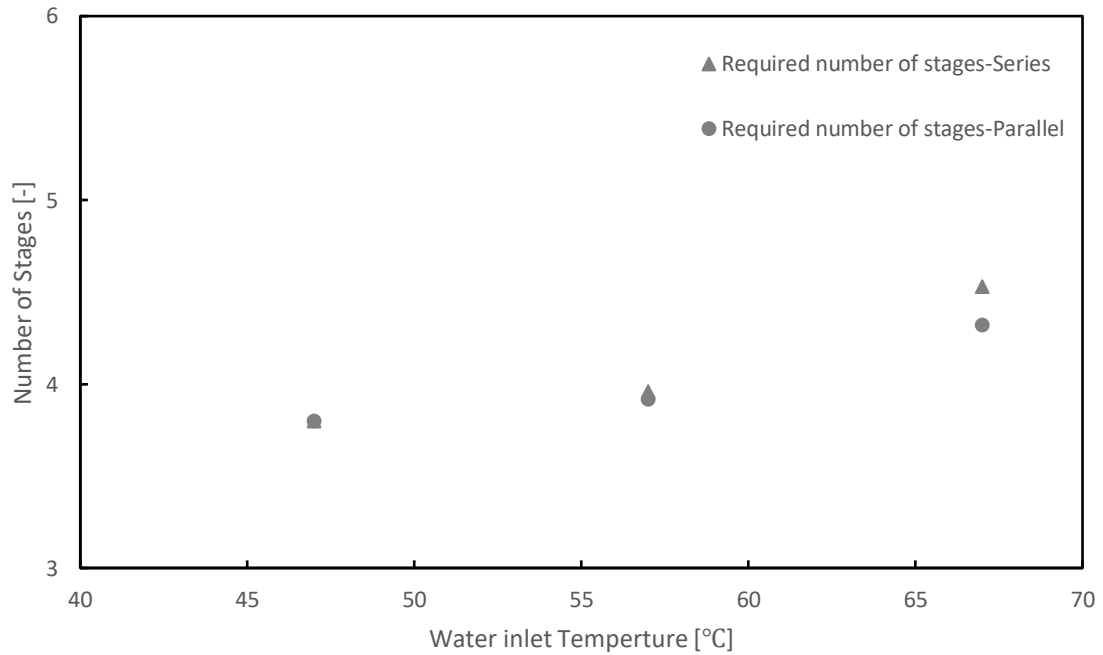


Figure 4-17. Effect of water Inlet Temperature on number of stages. Boundary conditions:

$$T_{ai} = 33\text{ }^{\circ}C; P = 101.3kPa; H = 0.02; m_{r,parallel} = 2.8; \varphi_i = 40\%$$

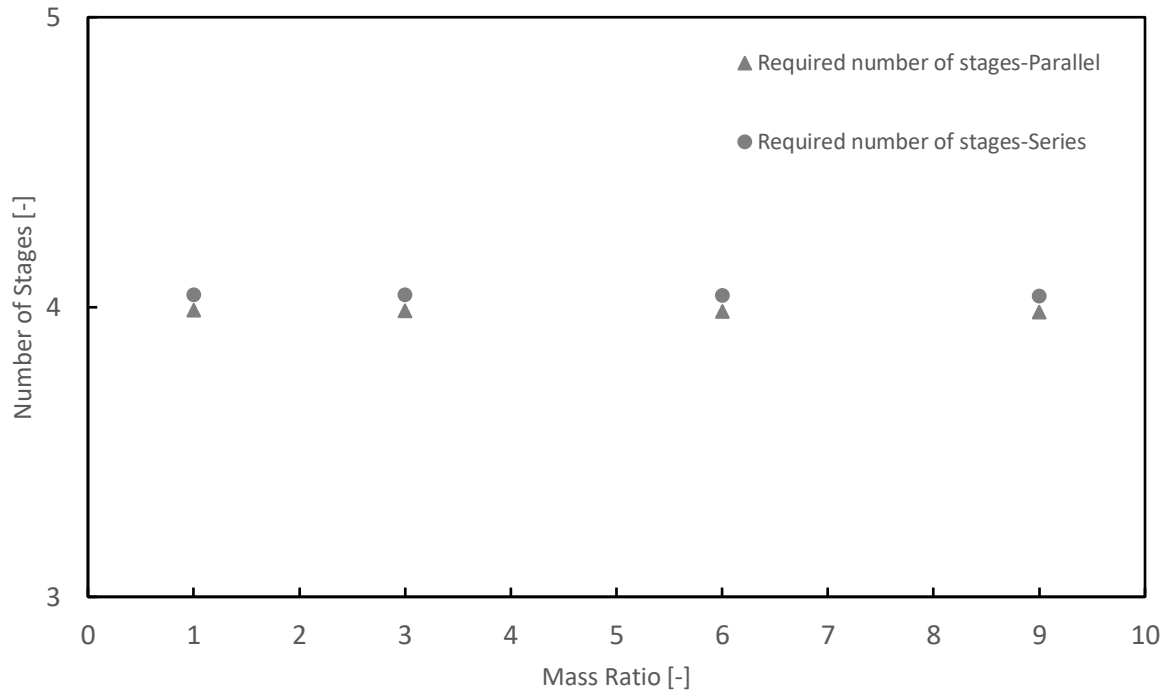


Figure 4-18. Effect of mass ratio Temperature on number of stages. Boundary conditions:

$$T_{ai} = 33\text{ }^{\circ}\text{C}; P = 101.3\text{ kPa}; T_{wi} = 60\text{ }^{\circ}\text{C}; H = 0.02; \varphi_i = 40\%$$

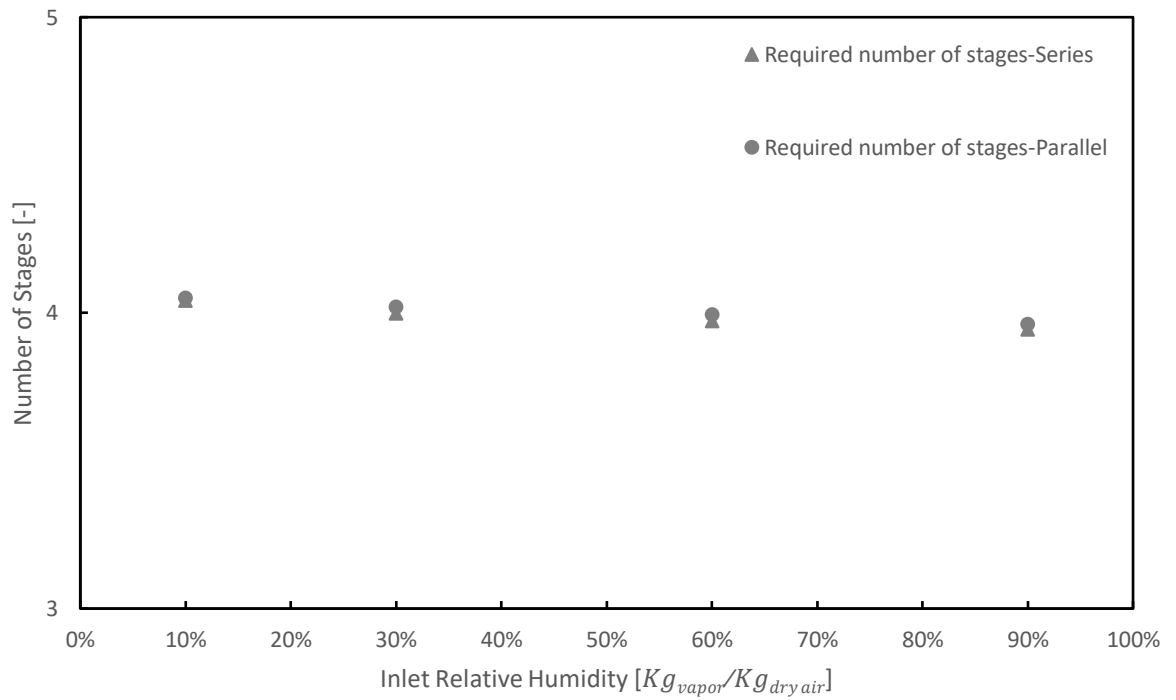


Figure 4-19. Effect of Inlet relative humidity on number of stages. Boundary conditions:

$$T_{ai} = 33\text{ }^{\circ}\text{C}; P = 101.3\text{ kPa}; T_{wi} = 60\text{ }^{\circ}\text{C}; H = 0.02; m_{r,parallel} = 2.8$$

4.3 Operation and Maintenance

Ultimately, this section will provide a comparison of parallel and series multi-staged bubble column humidifiers for the operation and maintenance aspects. This will provide a full picture of the practicality of both configurations. Generally, HDH desalination systems are inexpensive to build and maintain.

4.3.1 Operation

In the bubble column humidifier, heat and mass are transferred simultaneously from saline water to the air stream. For a series multi-staged bubble column humidifier, the hot water stream enters the bubble column from the top of the bubble column humidifier and through the water entrance gate, which is located in the fourth stage as indicated in Figure 4-2, after which the water will flow over the sparger until reaching the exit water gate that will allow the water to flow through to the other stage. The same operation will be repeated in the next stages until it reaches the lower plate, eventually, the water will be drained to the water heater for other iterations. For the air streams, the air is sparged through the perforated plate that is located at first stage bottom plate to form bubbles that will pass through a pool of hot water. Thus, due to the direct contact process, heat and mass are transferred simultaneously from the hot water to the air stream. Then, the humid air will go through to the next stages in the same operation until reaching the air exit gate. For the parallel multi-staged bubble column humidifier, the process of heat and mass transfer between the water stream and air bubbles are the same as the process in the series multi-staged. For the air stream, the same operation will take place as the series multi-staged bubble column humidifier. In contrast, as shown in figure 4-1, the water stream will be split into four (4)

parallel streams that will enter each stage separately. The main difference between the series and parallel configurations are the mass flow rate for the series configured humidifier which is set to be four times the one for the parallel humidifier. In addition, at each stage, the inlet water temperature is maintained constant for the parallel multi-staged columns while it is changing in the series one. Hence, it is clear that the series multi-staged bubble column humidifier has a more straightforward operation compared with the parallel multi-staged one.

4.3.2 Maintenance

Generally, adding trays will boost up the system performance; however, it adds additional complexity to the maintenance of a multi-staged bubble column humidifier. A parallel multi-staged bubble column humidifier will require less complicated maintenance compared to the series multistage one due to the separation of each stage.

4.4 Enthalpy Pinch

Enthalpy pinch is a performance measure for heat and mass exchanger. It can be considered as the minimum loss of enthalpy between the two streams in the bubble column humidifier. In the figures, it represents the minimum distance between the water and air streams achieved enthalpy. The lower the enthalpy pinch, the better the bubble column humidifier performance.

Parallel Configured Bubble Column Humidifier

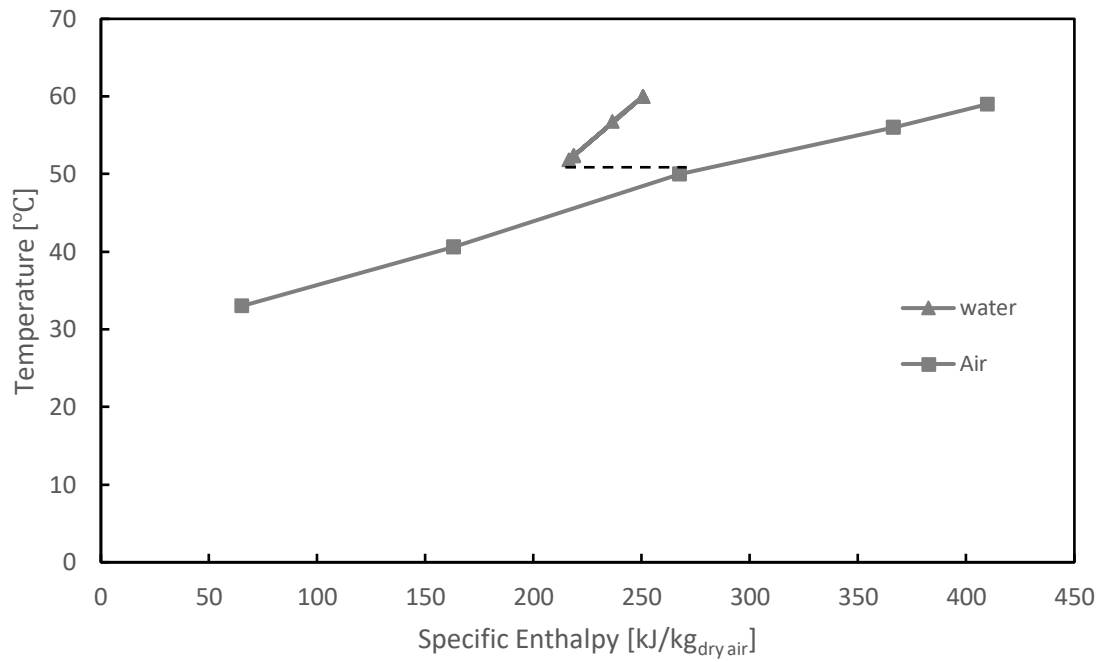


Figure 4-20 Enthalpy Pinch for Parallel Configured Bubble Column Humidifier

For the Parallel configured bubble column humidifier, the results shown in figure 4-20 indicated that the system has an enthalpy pinch of 47 kJ/kg dry air which considered to be slightly higher in comparison with the enthalpy pinch found in Karim M. Chehayeb [29] work, his study revealed an enthalpy pinch of 20 kJ/kg dry air.

Series Configured Bubble Column Humidifier

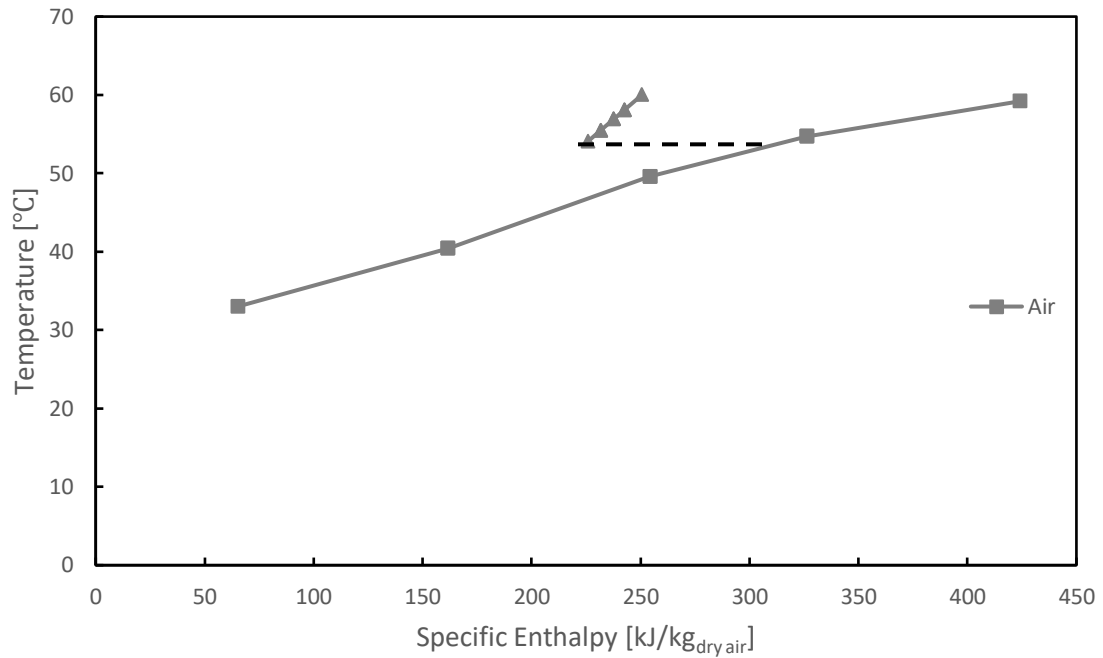


Figure 4-21 Enthalpy Pinch for Series Configured Bubble Column Humidifier

Results in figure 4-21 indicate that the series configured bubble column humidifier has a higher enthalpy pinch than the parallel configured bubble column humidifier. The results revealed that an enthalpy pinch of 70 kJ/kg dry air is the accomplished series configured bubble column humidifier, which is higher than the one obtained in the parallel humidifier. Thus, parallel configured humidifier is more efficient.

4.5 Chapter Conclusion

The ultimate goal of this chapter is to shed light on the influence of adding more layers to the bubble column humidifier on the total system performance. In this chapter, an analytical investigation was performed in order to evaluate the performance of a multi-stage bubble column humidifier for the moving streams model. Different configurations were studied to assess the effect of different configurations on the bubble column humidifier performance, these are Series and Parallel multi-staging. The effect of multi-staging on different performance parameters was also investigated, these are the outlet humidity ratio, total humidifier effectiveness, outlet air temperature and each stage's effectiveness for both series and parallel configurations. The results showed that the effectiveness, humidity ratio and outlet air temperature is profoundly affected by adding more stages to the HDH system, the results illustrated that the single stage of HDH system following the moving streams bubble column humidifier model could accomplish 27% and 33% effectiveness for parallel and series configuration respectively, with the addition of three more stages of bubble humidifiers in any configuration, the effectiveness can be tripled to reach to almost 94%. Both series and parallel configuration have almost the same results with a slightly higher percent for the series configuration. The data has been validated with Muntashri's [2] work, where he investigated multi-stages bubble column humidifier experimentally, his work was done considering three stages for the humidifier setup. In the third stage, the result is found to be quantitatively similar with a maximum effectiveness difference of 0.09%. This validation raises the confidence level of the obtained results. This chapter addressed the required number of stages to achieve an effective bubble column humidifier provided that the system operates in a preset initial boundary condition. Moreover, a comprehensive

parametric study is performed to discover the required number of stages to achieve a highly effective system with the variation of the initial boundary conditions. The results are showing that the required number of stages increases with the decrease of the air inlet temperature. In contrast, if the inlet water temperature increase, the required number of stages increase. Furthermore, mass ratio and air inlet relative humidity increment will have a minimal effect on the required number of stages. This chapter also covered a comprehensive parametric study that shows the influence of different operational parameters on the outlet conditions of both air and water streams as well as the total system effectiveness for both configurations. These parameters are the inlet water temperature, inlet air temperature, mass ratio, and relative humidity. Overall, the results illustrated that the increment of these parameters would increase the outlet air and water temperature. On the other side, the overall effectiveness is negatively affected by varying the water inlet temperature. Nevertheless, the other parameters increase will have a positive impact on the total effectiveness. This chapter also discussed the operational and maintenance differences between the parallel and series multi-staged bubble column humidifier. The discussion revealed that the series multi-staged bubble column humidifier has a more straightforward operation compared with the parallel multi-staged one. However, a Parallel multi-staged bubble column humidifier will require less complicated maintenance compared to the series multi-staged one. Moreover, Enthalpy pinch was investigated to assess the performance of the bubble column humidifier from another angle. Results revealed that both parallel and series configured humidifiers are having an enthalpy pinch of 47 kJ/kg dry and 70 kJ/kg dry air respectively.

Chapter 5

Conclusion and Future Work

5.1 Conclusion

In this thesis, a mathematical model has been developed to study the multi-stage bubble column humidifier as part of the Humidification dehumidification desalination system (HDH) for two different models. These models are the bubble column humidifier model that has stagnant water with a heating coil to maintain the required energy and the other model where the air and water streams are moving. The first model was considered as the base model for the other models that will be studied throughout this thesis. The mathematical models covered the heat and mass transfer and were solved through Engineering Equation Solver (EES).

The main obtained results are summarized as follows:

- The base model is validated with Liu [10] and found to be accurate with a maximum error percentage of 2.8%; moreover, with consideration of the Liu [10] reported uncertainty, this variation will be eliminated for some of the results.
- The results show that the superficial velocity increment has a positive impact on the total heat transfer and the mass transfer coefficient
- The increase of the air superficial velocity influences the bubble column humidifier effectiveness and the outlet temperature of air and water negatively.
- Moreover, the results also reveal that the column liquid height has an insignificant effect on total system effectiveness.

- Results show that in order to eliminate the bubble adhesion to the hot water coil and avoid the interruption of thermal balancing of air bubbles in the bubble column humidifier, it is suggested to balance between the outer diameter and length of the heating coil to maintain the desired heat and mass transfer rates inside the bubble column humidifier.
- Results show that the stagnant water model supersedes the moving streams model, and this is mainly due to the coil inserted in the bubble column humidifier that maintains the thermal balance across the water in the column.
- The stagnant water model is the optimum system. Nevertheless, due to the complexity of the design and practicality, the moving streams model is considered as the default design for the purpose of multi-staging.
- Effectiveness, humidity ratio and outlet air temperature are profoundly affected by adding more stages to the HDH system
- A single stage of bubble column humidifier can accomplish 27% and 33% effectiveness for parallel and series configuration, respectively, with the addition of three more stages of bubble humidifiers in any configuration, the effectiveness can be tripled to reach to almost 94%.
- Both series and parallel configuration have almost the same results with a slightly higher percent for the series configuration.
- The data has been validated with Muntashri's [2] and the results are found to be quantitatively similar with a maximum effectiveness difference of 0.09% in the third stage.

- Provided the preset initial boundary conditions, the required number of stages to achieve an effective bubble column humidifier is found to be four (4) stages.
- Results show that the required number of stages increases with the decrease of the air inlet temperature.
- In contrast, if the inlet water temperature increase, the required number of stages increase.
- Mass ratio and air inlet relative humidity increment have a minimal effect on the required number of stages.
- Results illustrate that the increment in the inlet water temperature, inlet air temperature, mass ratio and relative humidity would increase the outlet air and water temperature.
- The overall effectiveness is negatively affected by the increase of the water inlet temperature.
- The inlet air temperature, mass ratio and relative humidity increase have a positive impact on the total effectiveness.
- Series multi-staged bubble column humidifier has a more straightforward operation compared with the parallel multi-staged one.
- Parallel multi-staged bubble column humidifier requires less complicated maintenance compared to the series multi-staged arrangement.
- Results reveal that both parallel and series configured humidifiers are having an enthalpy pinch of 47 kJ/kg dry and 70 kJ/kg dry air respectively.

5.2 Future Work

- Investigate this study experimentally; this will raise the confidence of the credibility of the results. The experimental study would provide a physical expression of different aspects of the study, which might illuminate us to more factors that may affect the overall system performance.
- Investigate the feasibility of using different type of water with different contaminations. The main intention of this recommendation is to study the behavior of different types of water on the outlet air humidity ratio that will be influence the amount of the produced fresh water.
- Integrate the effect of the heat sources to the analytical equations as it is a vital part of the HDH system. This will ensure capturing all HDH parts' contribution to the overall system performance.
- Study the feasibility of running the system using solar energy as a primary source of heat. This addition will create a platform for integration between the desalination system and renewable energy.
- Consider of the study different designs of a dehumidifier to analyze the performance of the overall HDH system.

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- Meshary A. Al-Bahli, Mohammed A. Antar, 2020 “Behavioral Study of Two Bubble Column Humidifier Models – Analytical Study” (To Be Submitted)
- Meshary A. Al-Bahli, Mohammed A. Antar, 2020 “Multi Stages Effect on Enhancing the Bubble Column Humidifier System Performance” (To Be Submitted)
- Meshary A. Al-Bahli, Mohammed A. Antar, 2020 “Parallel and Series Multi-Staged Bubble Column Humidifier – Comprehensive Study” (To Be Submitted)