

**HEAT AND MASS TRANSFER FOR A SHALLOW
BUBBLE COLUMN DEHUMIDIFIER**

BY

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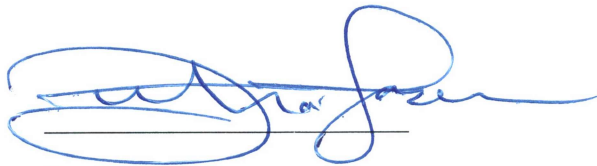
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Dedication

I devoted all my efforts through this Thesis to my Beloved Mother & Father. The Stage of live and achievements that I have today is only through their love, prayers, supports and guidance.

I also dedicated to my spouse whose support is always beside me throughout this journey. Moreover, this dedication is not completed unless & until I mentioned my lovely Daughter who is truly blessing for me.

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LIST OF ABBREVIATIONS

GOR	Gain Output ratio	HCR	Heat capacity Ratio
RR	Recovery Ratio	VPR	Vapor Productivity Ratio
BCE	Bubble column Evaporator	SHI	Specific Heat Input
HP	Heat Pipe	EES	Engineering Equation solver software
ETC	Evacuated Tube Collector	HDH	Humidification Dehumidification
RO	Reverse Osmosis	MSF	Multistage Flushing
Q_{Cond}	heat transfer by condensation	Q_{CE}	heat transfer to the cooling fluid
Q_{AB}	sensible heat transfer	m_{da}	mass of dry air
$m_{\text{w}_{\text{in}}}$	mass of water vapor inside the incoming air	m_{E}	coolant flow rate
m_{mai}	Total air inlet flow	X_{o}	outlet mole fraction
P_{C}	Saturation pressure at column temperature	MW_{w}	Molecular weight of water
MW_{a}	Molecular weight of air	T_{EO}	Outlet temperature of coolant
T_{EI}	Inlet temperature of coolant	T_{C}	Column temperature
R_{DE}	Resistance inside the tube	R_{CD}	Resistance outside the tube
Di	Internal Diameter of column	Re_{bubble}	$\frac{\rho_l V_{\text{rising}} D_{\text{bubble}}}{\mu_l}$
St	Stanton number	Pr	Prandtl Number
Fr	Froud Number	gw	Grams of water
$\Delta \omega$	Difference of Humidity Ratio	CO_2	Carbon Dioxide
Ar	Argon Gas	He	Helium Gas
D_{bubble}	Bubble Diameter	V_{rising}	Velocity of bubble in bulk liquid
μ_l	Viscosity of bulk liquid	ρ_l	Density of bulk liquid

ABSTRACT

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Humidification and Dehumidification (HDH) is a promising and economical technique for desalination . However, the conventional humidifiers and dehumidifiers used in HDH systems have low heat & mass transfer coefficients. Specially, high heat and mass transfer resistances of noncondensable gases in dehumidifier result in poor heat transfer and lower equipment's effectiveness, which make large volume of equipment and higher cost. In this regard, bubble column humidifiers and dehumidifiers in HDH desalination are ground breaking method that have the advantage of high heat and mass transfer coefficient with effectiveness (so lower equipment size) because of condensation/evaporation through numerous gas bubbles in liquid pool instead of any solid surface.

Hence this study commences with a critical review on usage of bubble column in HDH desalination, emphasizing analytical and experimental work of bubble columns. The study shows that in literature work, heat and mass transfer mechanism inside bubble column dehumidifier was analyzed inadequately through analytical modeling that involved assumptions and conflicts. Little attention was paid to mass transfer mechanism without consideration of gas side (bubble) transport mechanism. In addition, shallow column ($L/D < 10$) effects were not incorporated in existing models. Therefore, in the present study, a detailed mathematical model has been developed to accurately predict the heat and mass transfer mechanisms inside the bubble column dehumidifier for shallow bubble column

units while incorporating and dealing with all heat & mass transfer coefficients and inadequacies that were present in previous literature models. Proposed model is validated with literature experimental results and found in closed agreement with accuracy level of 11%. Parametric study of operational parameters such as air flow rate (1 SCFM to 3 SCFM) , air temperature (45°C-70°C) and operating pressure (2bar-8bar) reveal the behavior of all resistances and performance of bubble column dehumidifier. Mass transfer resistance is found to be high in term of magnitude i.e. 7.5 s/m while bubble side resistance is the lowest among all resistances (0.0002K/kW). Bubble diameter is found to be most sensitive parameter, with increment of only 1 mm, mass transfer resistance and bubble inside resistance increased by 25% and 36% respectively. Shallow column effects are presented in form of critical height. For critical height of 5cm, thermal resistances exponentially decreased from 0.12 K/kw to 0.06 K/kw. In addition to this, novel configuration of bubble column humidifier and dehumidifier is examined experimentally with water heated cycle and air heated cycle for different air temperatures and water levels with five air flow rates. Water heated cycle is found to have better performance in term of productivity (0.6 liter/h) and GOR (0.4) than air heated cycle. Experimental setup is modified in term of double stage humidification with reheating for air heated cycle which enhanced its performance producing distillate of 0.3 liter/h at GOR of 0.4 with increment of 7.5% heat input.

ملخص الرسالة

الاسم الكامل: ماجد خان

عنوان الرسالة: انتقال الحرارة والكتلة لمزيل رطوبة عمود الفقاعة الضحلة

التخصص: الهندسة الميكانيكية

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تعتبر تقنية الترطيب وإزالة الرطوبة تقنية واعدة واقتصادية لتحلية المياه. ومع ذلك، فإن أجهزة الترطيب ومزيلات الرطوبة التقليدية المستخدمة في الانظمة القائمة على هذه التقنية لها معاملات انتقال حرارة وكتلة منخفضة. وعلى وجه الخصوص، تؤدي المقاومة العالية لانتقال الحرارة والمادة في الغازات غير القابلة للتكثيف في جهاز إزالة الرطوبة إلى انخفاض نقل الحرارة وفعالية المنظومة، مما يجعل حجم المعدات كبيراً ويؤدي إلى زيادة التكلفة. في هذا الصدد، يعد عمود الفقاعة المستخدم كمربط ومزيل للرطوبة في الانظمة المعتمدة على الترطيب وإزالة الرطوبة لتحلية المياه الطرق الرائدة التي لديها معاملات انتقال حرارة ومادة عالية تمتاز بالفعالية (بالتالي حجم أقل من المعدات) بسبب التكثيف / التبخر من خلال فقاعات غاز عديدة في حوض سائل بدلاً عن استخدام سطح صلب.

تقوم هذه الدراسة بمراجعة نقدية حول استخدام عمود الفقاعة في أنظمة التحلية المعتمدة على الترطيب وإزالة الرطوبة، مع التركيز على العمل التحليلي والتجريبي لعمود الفقاعة. وقد أظهرت الدراسة أنه في الدراسات العلمية السابقة، تم تحليل آلية انتقال الحرارة والكتلة داخل عمود الفقاعة لإزالة الرطوبة بشكل غير كاف من خلال النمذجة التحليلية التي تضمنت العديد من الافتراضات والتناقضات حيث تم إعطاء القليل من الاهتمام لآلية انتقال الكتلة دون اعتبار لآلية نقل الغاز (الفقاعة) الجانبية. بالإضافة إلى ذلك، لم يتم دمج تأثيرات العمود الضحلة ($L / D < 10$) في النمذجة. لذلك تم في هذه الدراسة تطوير نموذج رياضي مفصل للتنبؤ بدقة بآلية نقل الحرارة والكتلة داخل مزيل الرطوبة في عمود الفقاعة الضحلة مع تضمين جميع معاملات انتقال الحرارة والكتلة بالإضافة إلى التعامل مع أوجه القصور التي كانت موجودة

في الدراسات السابقة. تم التحقق من صحة النموذج بمقارنته مع النتائج التجريبية من الدراسات العلمية السابقة حيث وجد ان هنالك مقدار كبير من التوافق. تكشف الدراسة البارامترية لظروف التشغيل الرئيسية أن الزيادة في معدل تدفق الهواء ودرجة الحرارة وضغط التشغيل لها تأثير إيجابي كبير على أداء مزيل الرطوبة في عمود الفقاعة. كما وجد أن قطر الفقاعة هو أكثر المعاملات حساسية لمقاومة انتقال الكتلة من حيث الحجم في حين أن المقاومة الجانبية للفقاعة هي الأقل بين جميع المقاومات. بالإضافة إلى ذلك، تم اجراء فحص تجريبي لتكوين مبتكر لمزيل الرطوبة في عمود الفقاعات باستخدام دورة تسخين المياه ودورة تسخين الهواء لدرجات حرارة هواء ومستوى ماء مختلف مع خمسة قيم لمعدل تدفق الهواء. ووجد أن لدورة تسخين المياه أداء أفضل مقارنة بدورة تسخين الهواء من حيث الإنتاجية والنسبة المكتسبة لمعامل الأداء (GOR). وقد تم تعديل التجهيز التجريبي في فترة الترطيب على مرحلتين مع إعادة التسخين لدورة تسخين الهواء لتحسين الأداء.

CHAPTER 1

INTRODUCTION

The rapid growth of world's population along with technological advancement in industrial, domestic and agriculture sectors have increased the consumption of fresh water. The availability of usable water is reducing day by day while requirement of fresh water is increasing progressively. In fact, all around the world, the accessibility to potable water for the human is narrowing down. In this era, both the developed and undeveloped countries are suffering from substantial water scarcity due to uncontrolled and polluted industrial activities. The available resources of consumable water (rivers, lakes and underground water) are limited by nature and rest of earth's water is not in the form for direct utilization. Quantitatively, the global water reserves are around 1.4 billion km^3 (330 million cubic miles). In this figure, 97.5% is the saline ocean water while only 2.5% (about 35,000,000 km^3) is the fresh water and moreover this freshwater exists in different forms such as glaciers, snow cover, moisture in atmospheres, swamps water (about 24,000,000 km^3). It is estimated that 0.77% (about 11,000,000 km^3) of all earth's water is ground water and surface water (lakes, rivers) Thus this depicts that merely 0.014% of fresh water is directly available to human beings [1]. The pictorial representation of the distribution of water resources is shown in Figure 1-1

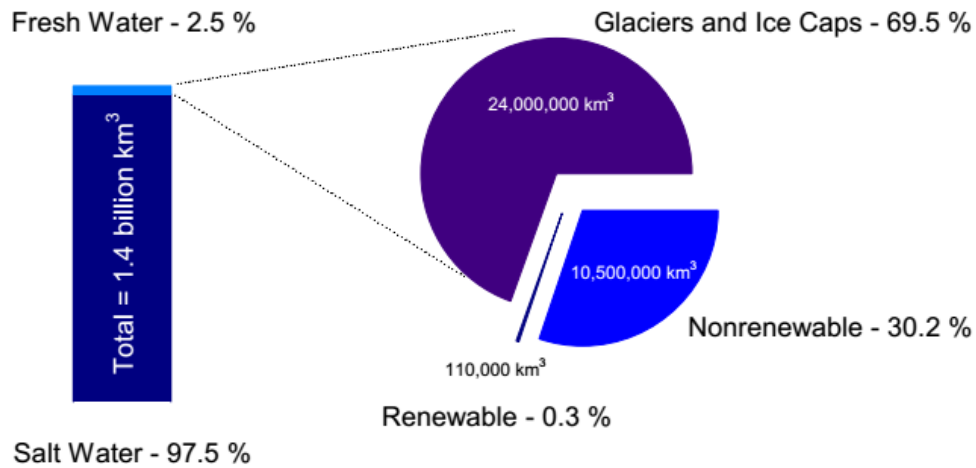


Figure 1-1 Quantitatively figures of distribution of world's water [1]

Therefore, lot of research work has been performed to meet the demand of fresh water against its consumption. Up till now, several desalination techniques have been developed and technological advancement of 21st century has played significant role in conversion of saline to fresh water. All desalination techniques can be classified into two main groups : (i) thermal systems (ii) membrane process . Thermal desalination system mainly includes multi stage flash (MSF), multiple effect desalination (MED), Vapor compression desalination (MVC), Humidification Dehumidification (HDH) and Solar desalination (SD). While Reverse osmosis (RO), Forward Osmosis (FO) and Electro dialysis (ED) come under the domain of membrane process. Desalination process like MSF, MED & RO are rich in energy consumptions [2]. Most of these methods have been commercialized on large scales that require substantial amount of electrical energy. Figure 1-2 compares the electrical energy requirement for mentioned technologies .

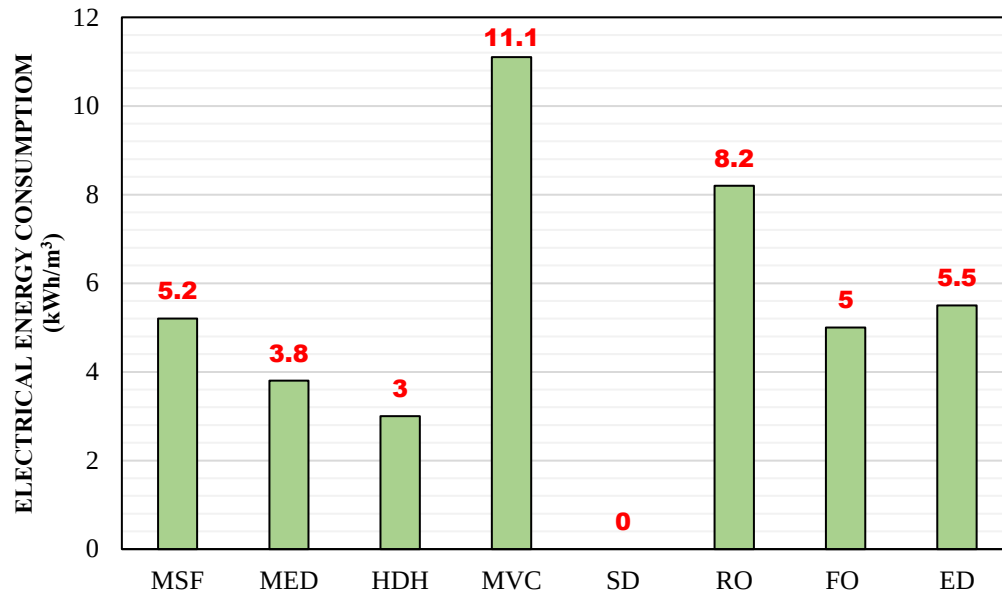


Figure 1-2 Electrical energy consumption of different Desalination technologies [3]–[5]

These high energy consumption large-scale plants are suitable for developed countries but unaffordable to developing countries. The problem becomes more acute in remote or less developed areas where the cost, input energy and technology are in parallel consideration with the production of fresh water. Therefore, small scale desalination system with less technological requirements is the need of time for the fresh water production. In this regard, humidification and dehumidification (HDH) found to be best suited particularly for remote areas. HDH is a thermal desalination technique that imitates the natural rain cycle. It is based on the concept of humidification of carrier gas by brackish water just like what happened in cooling towers and then dehumidification of same carrier gas at low temperature. During dehumidification water resulting from the condensation is distilled one and termed as the output or productivity. In comparison with other desalination techniques, HDH is the simplest one, it has modest design, low energy requirement, easiest operation, less maintenance and has minimum operating cost [3][4] [5].

1.1 Humidification Dehumidification (HDH) Desalination

Basic design of HDH consists of Humidifier, Dehumidifier and a heating system. HDH mostly use carrier fluid as air to extract the heat from the energy source/heating system which is then pass from the humidifier where it comes in contact with the saline water. Here heated air humidified from evaporation of saline water carrying water droplets and leaving behind salts content. This air is allowed to pass through dehumidifier from where condensation of water droplets occurred and resulted into distilled water. The simple schematic of HDH desalination along with its main components is shown in Figure 1-3

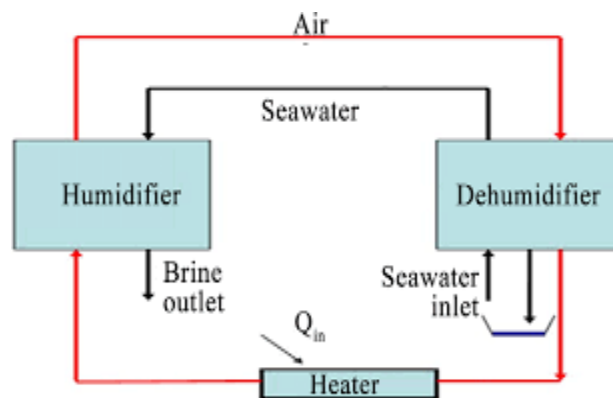


Figure 1-3 General Schematic of HDH Cycle

Attractive features of HDH are : it can be combined with renewable energy sources or any low-grade heat to provide the thermal energy needed to power the cycle and it can be operated at low pressures close to ambient. Low-pressure steam of power plant, solar energy, waste heat energy or even by geothermal sources can be used to power the HDH system. Similarly, HDH systems does not involved any costly material or specialized membranes that require frequent maintenance like RO. Low operational conditions (temperature and pressure) enabled this system to be constructed easily, economically and system can be applied to smaller scale (10-100m³/day) which is not possible for case of

MSF and MVC (due to variation of operational pressure and high temperature $> 100^{\circ}\text{C}$ to flash).

The performance of the HDH system depend upon its main components i.e. humidifier and dehumidifier. Design and configuration of these two components greatly effects the efficiency of whole HDH system. Spray tower, wetted wall tower, packed bed tower and evaporator are conventional humidifiers while condenser, air cooler heat exchangers, shell tube and plate fin heat exchangers are common dehumidifiers for HDH system [6]. However, there are some drawbacks associated with all these conventional humidifier and dehumidifiers. Spray towers have low efficiency due to lower water hold up and considerable pressure drop occurs at water side because of spray nozzles [7]. Packed bed humidifier requires large packing volumes for achieving high heat and mass transfer which makes huge sizes of equipment and costly [13] [14]. Narayan et al. [3] reviewed several packing materials (honeycomb paper, plastic, wood, ceramic etc.). It is shown that somehow performance, durability, efficiency have sacrificed in one or another configuration due to either fouling , poor contact area and pressure drop. In the same manner finned heat exchangers, as dehumidifier are greater in size for effectiveness. Substitutional pressure drop has been noted in tubes of shell tube type exchanger. In addition, with the issue of fluid leakages, all conventional dehumidifiers have low heat and mass transfer coefficient because of degradation of film condensation on solid surface due to presence of non-condensable gases.

Hence recent studies in HDH desalination mainly focus on achieving high heat and mass transfer coefficients in humidifier and dehumidifier. Researches have investigated the several direct contact types of humidifiers and dehumidifiers [13]–[15] to avoid additional

resistances due non-condensable gases and to reduce equipment size and cost with high effectiveness. In this regard, Bubble column as humidifier and dehumidifier is found as one of the ground-breaking techniques in field of HDH desalination. The idea of bubble column in HDH desalination comes from the concept of bubble column reactors. Bubble column reactors are widely used in chemical, petrochemical, metallurgical and biochemical industries for gas-liquid contacting and chemical reactions[16]. In bubble column, gas phase is intentionally sparged through column's bottom using perforated plates in to the column containing liquid phase. Bubble columns are considered as multiphase device because of simultaneous heat and mass transfer between gas and liquid phase that enables to have high heat and mass transfer coefficient with compact size.

CHAPTER 2

DESALINATION WITH BUBBLE COLUMN

2.1 Origin of Bubble Column in Desalination

M. J. Francis and R. M. Pashley. [17] are the pioneers of the concept of non-boiling bubble column for thermal desalination. Their strategy was to find the alternate method of commercialized MSF systems that have drawbacks of high energy demand in boiling, cost and lower efficiencies. Experimentally, it was shown that the amount of water vapor present in air bubble immersed in liquid pool at equilibrium state is same to that of bubble created by boiling. Hence the idea was to develop a process to produce an efficient vapor exchange process with a high water/air interfacial area, without boiling. Unusual property of salty water of prevention of bubbles coalescence become the bases of novel and improved non-boiling thermal desalination technique [18]. In this process water vapor can be captured, transferred and collected using a simple, continuous, fine bubble column operated at temperatures well below the boiling point. This produces a uniform, efficient exchange of water vapor into the bubbles, which can then be condensed and collected as pure water.

2.2 Bubble Column in HDH Desalination

2.2.1 Solution of low heat & mass transfer coefficient in Dehumidifiers

Air is mostly used carrier gas in HDH desalination so in high concentration i.e. from 60-95% by mass, non-condensable gases are present in the condensing vapor air mixture [19].

Presence of noncondensable carrier gas increases the thermal resistance to condensation of water droplets of humid air on a cold surface (tubes or any solid surface). This additional resistance is much higher than in a pure vapor environment. It is mainly because of diffusion resistance to transport of vapors through the mixture of noncondensable gas and vapor. Resulting heat transfer coefficient of condensation of water droplets on coil, tube or on any heat exchanger element is low. It is found that despite of few mole percentage of non-condensable gas present in condensing fluid, the deterioration in the heat-transfer rates is significant [20]–[22] and low in term of magnitude even in order of $1 \text{ W/m}^2\cdot\text{K}$ [23], [24]. Quantitatively the amount of deterioration in heat transfer is a very strong (almost quadratic) function of the mole fraction of non-condensable gas present in the condensing vapor [25].

Bubble column resolve this issue by shifting the condensation from solid surface to surface of large number of tiny gas bubbles. Gas bubbles distributed uniformly into the column liquid through perforated plates. The liquid inside the column is at lower temperature. The enormous number of small bubbles spherical in shape have large interfacial surface area which provide greater contact area between the liquid and gas bubbles. Hence condensation occurs at bubble surfaces. The turbulence created by bubbles significantly enhanced heat transfer. Thus these features of multiphase mechanism of high heat and mass transfer coefficient provide small equipment size (condensing element) and reduced cost.

2.2.2 Low energy Consumption of the system

Bubble column does not require boiling of saline water, the amount of water vapor present in air bubble being immersed in liquid pool is almost identical to that in bubble created by boiling. The elimination of need of boiling ultimately reduces the energy demand. Thus,

Desalination with bubble column can be carried out with less energy requirements M. Schmack et al. [26] compared the energy requirement for bubble column against RO. For bubble column the energy requirement is only for airflow to overcome the static water pressure head and resistance from the sieve plates. It is estimated that for series of eleven bubble column operating at 88 °C requires total energy for air pumping from blower (e.g. HRB 402/1) is around 2 kWh/m³ [17]. This amount is less than half of the energy requirement of Kwinana desalination Reverse Osmosis plants where the total energy used per unit of water is 4.6 kWh/m³ [27]. In another study, M.Shahid et al.[28] compared energy consumption by RO, conventional thermal unit and Bubble column HDH systems in detail through different methods. It is shown that commercial RO system utilized 4-6 kWh/m³, thermal units consumed 14-27 kWh/m³ while bubble column evaporator in thermal desalination due to non-boiling mechanism consumed 7.55 kWh/m³ at 550K of inlet gas temperature.

In comparison with conventional solar still, bubble column has advantage of separate evaporation and condensation. Here condenser can be cooled without affecting the heat input of evaporation. The conventional solar still can give productivity of 2-3liter/m² in a day and even the modified wick type can produce 5 L/ m² in a day [29]. Whereas for bubble column system with condenser minimum area as 0.075 m² with same efficiency and operating temperature of [29] can give 19 Liters/ m² of fresh water in day based on experiment. Despite of any modification for improved productivity of solar still, its productivity will always be limited on scale of m²/m² basis.

2.2.3 High Effectiveness and Compact size of Humidifiers

O.M Muntasheri [30] experimentally demonstrate the substantial change in effectiveness of bubble column humidifier when compared with conventional packed bed humidifier at similar condition. This is shown in Fig. 2-1. interestingly the effectiveness of 03 stages of bubble column humidifier with total volume of 0.036 m^3 is 85% quite greater than 76% of 05 packing bed humidifier with total area of 0.35 m^2 . Hence bubble column humidifier is capable to give high effectiveness with compact size.

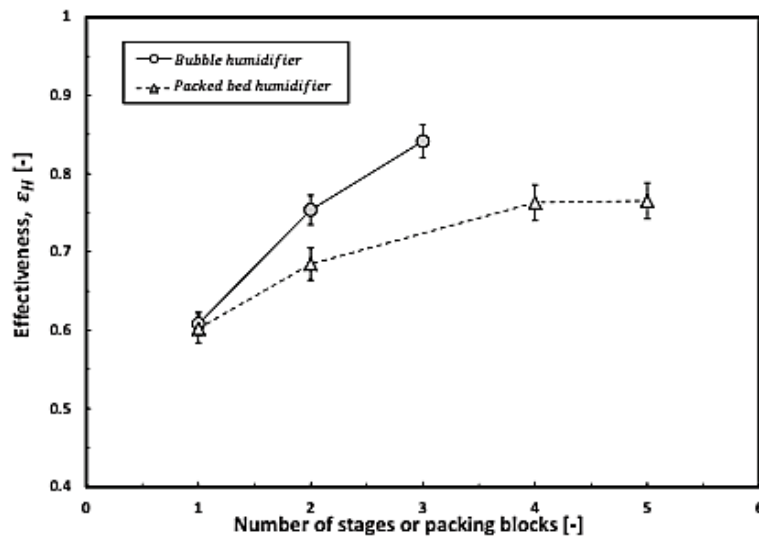


Figure 2-1 Comparison of effectiveness of bubble column humidifier with packed bed humidifier

2.3 Bubble formation and Physics inside the bubble

The Geometrical features of bubble column is simple. It is a vertical cylindrical column where the gas phase is dispersed (usually from the bottom of column) through a perforated plate. The liquid through which gas passes is in stationary state forming a liquid bulk pool. Basic physical mechanism that happens inside the bubble column is ; the gas in the liquid

column after passing from bottom through the perforated plate is shaped in to bubbles due to surface tension property and then it moves upward because of less density in form of bubbles instead of continues gas phase. Bubbles after sparging gets a thin boundary layer of liquid phase (usually water) around its volume. Hence at macroscopic level each bubble is surrounded by a thin boundary layer across it ,called as liquid boundary as shown in Figure 2-2 [20].

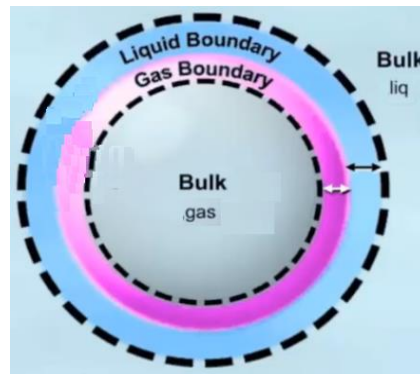


Figure 2-2 Boundary Layers Inside bubble (Physical mechanism)

Generally, for water inside the bubble column, the bubbles formed from sparger are smaller than sparger's holes diameters. Bubble maintain spherical shape to withstand the drag forced induced by bulk liquid. This result in enhancement of gas hold up in the system. Moreover, due to smaller sizes, bubble rise velocities are slower and this result in large residence time.

Formation of bubble can be considered as static and quasi static phenomenon followed by dynamic process namely coalescence and breakup [21]. It occurs in different modes such as bubbling, chain bubbling, wobbling and jetting depend upon different operating and design conditions such as gas-liquid properties, gravitational force, design construction of office and operating condition as gas flow etc. It has been found that bubble formation and

its shape are strong function of gas velocity and liquid depth. Muller and Prince[22] observed different regimes as shown in Figure 2-3.

For low gas flow rates and liquid depth ($<100\text{mm}$) bubbles occurred as singularity in shape as shown in Figure 2-3(A). This happened to certain flow rates where mode changes to single bubble to chain bubble shown in Figure 2-3 (B). Here the bubble size depends upon the orifice configuration, surface tension of liquid and densities difference of two fluids. At high flow rates of gas (15mL/s) the gas emerged as continuous phase and maintain this continuity upto certain distance from mouth of sparger Figure 2-3 (C). As flow rate increases more and liquid height inside the column decreases the bubbles mold into jet regime, bubble formation occurs due to jet breaking which is due to instability of jet shown in Figure 2-3 (D) [21].

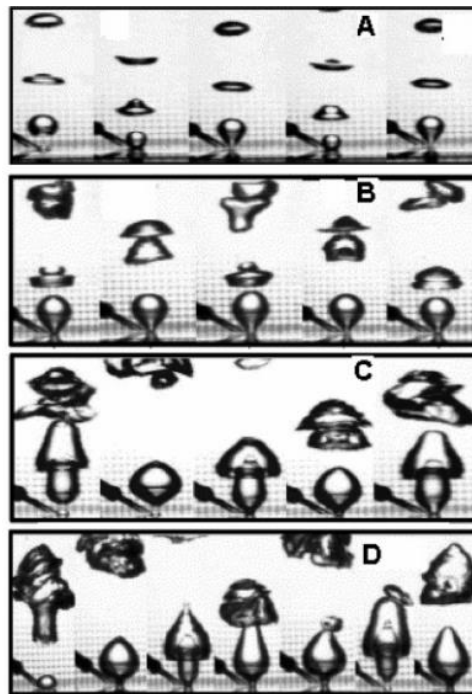


Figure 2-3 Different Flow regimes in bubble column [30]

The main forces for single bubble formation are buoyancy, drag, surface tension and gravity. Formation and detachment of bubble inside the liquid creates local disturbance and only order of volume of bubble is carried along with its drift volume. For case of chain bubbling, driving force in upward direction of bubble motion occurred in the vicinities of path of bubbles that result into weak circulation. Similarly, the strong upward motion of the jet induces stronger recirculation in the liquid with upward motion along the jet and downward flow away from jet [21].

Temperature difference across the formed bubble leads to sensible heat transfer while due to concentration difference mass transfer or condensation occurs. The temperature difference across liquid thin boundary layer is not substantial in comparison with the temperature difference between the inside bubble i.e. bulk gas and columns bulk liquid. This is due to fact that liquid (water) which made boundary layer around the bubble has high thermal conductivity so its gains the temperature of bubble (bulk gas) but due to low thermal diffusivity it does not loss heat to column that it gains from bubble [23].

CHAPTER 3

DETAILED LITERATURE REVIEW

The Figure 3-1 summarized the scope of the literature work that has been carried out in HDH desalination with bubble column.

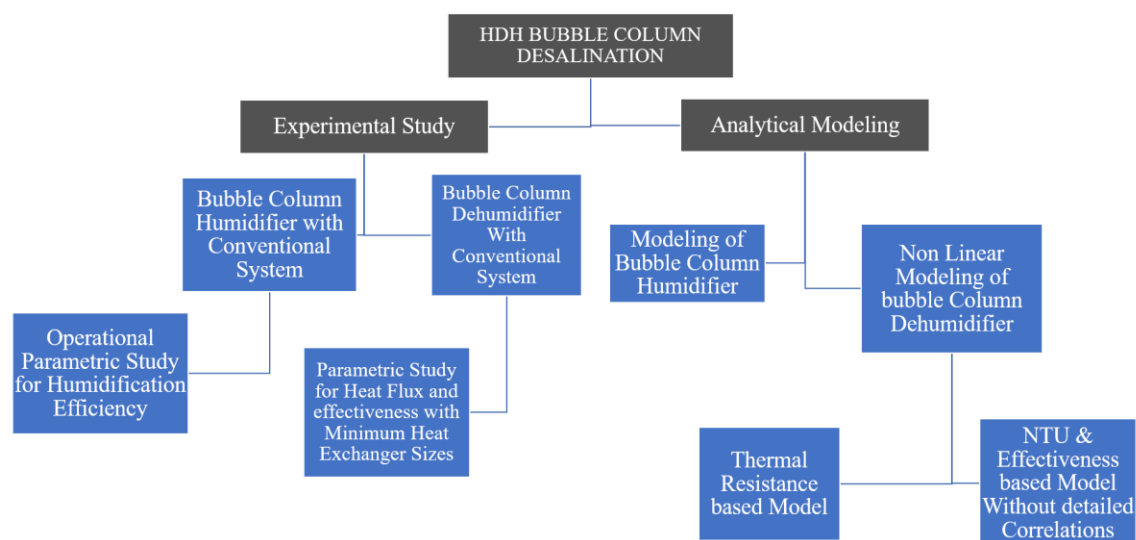


Figure 3-1 Flowchart for literature of study of Bubble Column in HDH desalination

The Table 3-1 is the condensed representation of the literature including the theoretical, experimental work with brief descriptions of each paper

Table 3-1 Synopsis of Literature Review for Bubble Column in HDH desalination

Authors & Years	Type of Study	Humidifier	Dehumidifier	Type of Cycle	Energy Source	Technique/ Software	Focused Parameter	Main features
V .Patel et. al [34] 2019	Theoretical modeling & experimental	Bubble column	Heat exchanger	Air & water heated	Electrical heater	EES	Relative humidity at humidifier outlet	Humidifier bubble column is experimentally and theoretically analyzed through operational parameters as column height, hole diameter, air flow rate and temperature in term of relative humidity (RH). RH found to have direct proportional relation to all mentioned parameters except sparger hole diameter. Maximum RH of setup is 95% at air temperature of 48°C.
V .Patel et. al[35] 2019	Experimental	Bubble column	Thermoelectric cooler (TEC)	Air & water heated	Electrical energy for water and Solar energy for air	N/A	Distillate, GOR & economic analysis	HDH system with bubble humidifier and TEC dehumidifier has analyzed. The productivity is found as 7-13 Liter/day for different operating conditions. Optimum productivity and GOR are 12.91 L/day and 0.8 respectively. The economic analysis shows the payback period of proposed system is 2.5 years with net profit per liter is Rs.17.79/-
P. Raj Ranjitha et al.[36] 2018	Theoretical modeling & optimization	N/A	Bubble column	N/A	Solar energy	MATLAB Simulation	Dehumidifier sizing optimization	Optimization study for bubble column dehumidifier sizing is carried out based on the air outlet temperature. For the air exit temperature of 35°C to 45°C, optimized variables are column height = 0.85m, Sparger hole =0.3m, airflow rate =0.04 kg/s and bubble diameter = 2mm
P. Gao et al. [37] 2018	Theoretical modeling	Bubble column	Heat exchanger	Air heated	Solar collector	Not mentioned	Water production and energy consumption	-Treatment of coal mine waste water along with HDH bubble column desalination. -By analysis of heat and mass transfer, the effects of solar radiation, air bubble velocity and water temperature on water production have studied. -Thermal energy consumption of the system is found moderate (700kJ/kg) in comparison with other water treatment methods.
C. Fan et al. [38] 2018	Experimental	Bubble column evaporator (BCE)	N/A	N/A	N/A	N/A	Energy balance	-Previous Energy balances of Bubble column Evaporator are reviewed experimentally. -Modified Energy Balance is proposed in term of Enthalpy of vaporization of various salt solution. -Experimental Results are compared as basis for the comparison between proposed and previous energy balances.

M. Taseidifar et al. [39] 2018	Experimental	Bubble column as evaporator (BCE)	Cylinder with fresh silica gel	Not mentioned	Gas heater	N/A	Water evaporation efficiency	-BCE is studied for enhanced evaporation of water in 0.5m NaCl solutions, using preheated carrier gases (air, CO₂, Ar & He), each at inlet temperature of 150 °C. -Hot helium improved water evaporation efficiency, by a factor of about 3 times compared with dry air
H.B. Halima et al.[40] 2017	Experimental	Bubbler humidifier in solar still basin	N/A	Air heated	Solar energy with electrical heaters	N/A	Humidification effectiveness	-Experimental study is performed for modified solar still through bubbling effect in its basin along with estimation of heat and mass transfer coefficient b/w air and water. -Humidification efficiency found to be slightly affected by water level while moderately effected by water temperature and air flow rate.
T.Rajaseeni vasan and K. Srithar [41] 2017	Theoretical modeling & experimental	Bubble column	Shell and tube heat exchanger	Air heated	Biomass stove	N/A	Optimum operational parameters of system	Biomass operated HDH system with bubble column humidifier and dehumidifier system is examined. Setup is examined with direct and pre heating of supply air. High distillate is collected for direct supply as 3.5kg/h. While the recovery of waste heat reduced the fuel consumption from 0.36kg to 0.2kg for 1kg of distillate water.
Abd-ur-Rehman and Al-Sulaiman. [42] 2017	Experimental	Multistage Bubble column	N/A	Water heated	Solar energy with electrical heaters	N/A	Humidifier performance in terms of absolute enthalpy	- Optimum operating parameters for a novel multistage stepped bubble column humidifier design is determined. - Studied the variation of the pressure drop across column v/s column's height at different superficial velocities of air. Water column height of 1 cm & superficial velocity of air 25 cm/s give best humidifier performance along with lowest pressure drop.
K. Srithar and T.Rajaseeni vasan [43] 2017	Experimental	Bubble column	Shell and tube heat exchanger	Air heated	Solar Energy	N/A	Specific humidity at humidifier outlet	-The performance of system is analyzed with variation of humidifier's water depth, air mass flow rate, presence of turbulators on absorber plate with direct & preheated air. On comparison with published work specific humidity difference is highest of the proposed system as 0.187 kgw/kg

F. Al-Sulaiman et al. [44] 2017	Experimental	Bubble column	Not mentioned	Water heated	Solar Energy	N/A	Optimum design and performance parameters	-Studied different perforated plate geometries for optimum design that delivers a low pressure drop in the system. -Studied the effects of air superficial velocity, inlet water temperature & inlet air relative humidity on the humidifier's performance w/ & w/o Fresnel Lenses -Average absolute humidity of outlet air increases by 12.3% when superficial velocity increased from 20 to 30 cm/s & it further enhanced by 11% with the presence of Fresnel lenses in system
Al-Qutub et al. [45] 2017	Patent	Multistage Bubble Column Humidifier	N/A	N/A	N/A	N/A	N/A	Invention of multistage bubble column humidifier apparatus comprising a down comer unit for use in a humidification - dehumidification system for purifying water.
T. Rajaseenivasan et al. [46] 2016	Experimental	Bubble column	Shell and tube heat exchanger	Air heated and water heated	Solar Energy	N/A	GOR, Distillate quantity, system cost and humidifier efficiency	-System's performance is studied by varying the mass flow rate of air & cooling water for the three-different configuration such as solar air heater, heater with turbulators and dual purpose solar collector with turbulators. - Humidifier's highest efficiency is 92% with configuration of dual-purpose collector while 3.3 is maximum GOR -Distillate productivity is 16,20 & 23 kg/m² day while the cost of systems is 0.032 \$/kg, 0.026 \$/ kg, & 0.019 \$/kg for conventional solar air heater, solar air heater with turbulators & dual-purpose solar collector respectively.
Abd-ur-Rehman and Al-Sulaiman. [47] 2016	Experimental	Multistage Bubble column	Not mentioned	Water heated	Solar Energy	N/A	Parametric study of bubble column humidifier	-A novel design of solar heated multi-stage bubble column humidifier is proposed. - On average day round absolute humidity, w/o Fresnel lens is increased up to 9% and 23% for the two stages and three stage configurations respectively when in view against the single stage. In addition to this Abs. humidity reaches to 25% for same conditions but with Fresnel lens.

Behnam and Behshad. [48] 2016	Experimental	Bubble column	Exchanger with cooling coils	Air heated	Solar evacuated tube collector	N/A	Complete system performance and distillate productivity	The effect of initial water depth in the humidifier, incoming air bubble flow rates & effect of adding the fluid to the space between heat pipe & evacuated tube collectors on fresh water production was investigated -For Maximum Production: 1) Initial water depth in the humidifier is equal to the length of heat pipe condenser inside the humidifier. 2) High inlet air flow rate 3) Presence of fluid as oil in the space between the evacuated tube collectors and heat pipe.
M. Schmack et al. [49] 2016	Experimental	Bubble column as evaporator (Configuration I)	Bubble column as Condenser (Configuration II)	Water heated	Heating bath	N/A	Economical and effective vapor cooling	-Experimental configuration focuses on achieving effective vapor cooling keeping in view of low energy demand, low environmental impact, cost efficiency, durability & ease of maintenance of system.
M. Schmack et al. [26] 2016	Experimental	Bubble column as evaporator	Flat-plate type Condenser	Water heated	Waste Heat source (Not defined)	N/A	Maximum productivity with simple & Economical technology	-Resembled to solar still, a novel design is proposed for HDH desalination to overcome the draw backs of conventional desalination system with simple technology and inexpensive material. -With various physical operating parameters, it is found that system is producing 19 liters/day equivalent to 5-8 times modified solar still design
Liu and Sharqawy [50] 2016	Experimental & Theoretical modeling	Bubble column	Bubble column	Water heated	Heating coils	Modeling on EES	Effect of variable pressure on HDH bubble column desalination	-High heat transfer & effectiveness of bubble column is achieved in case of lower operating pressure of humidifier & higher pressure for dehumidifier. -Parametric analysis is carried out for effectiveness and heat transfer rate. Effectiveness and heat transfer rate is independent from liquid column height but increases with superficial velocity of incoming air. -Modified e- NTU mode is proposed and found in closed agreement with experimental results.

A. Khalil et al. [15] 2015	Experimental	Bubble column	Shell and tube heat Exchanger	Water heated	Solar water collector	N/A	Distillate, efficiency & GOR	Flow characteristic of bubble inside the bubble column is investigated through different sets of perforated plate. Proposed system achieved daily productivity, daily efficiency and GOR 21 kg, 63%, and 0.53, respectively, at inlet water temperature is 62 °C.
K.M. Chehayeb et al. [51] 2015	Theoretical Modeling	Packed bed	Multistage Bubble column	N/A	N/A	MATLAB	System performance, Entropy Generation.	-Analyze the effect of the mass flow rate ratio on fixed-size system along with consideration of entropy generation effects. -Implement air extraction/injection & simulate a wide range of operating conditions
M. Schmack et al. [52] 2015	Conceptual Design	Multistage Bubble column evaporator	Multistage bubble column condenser	Water heated	Solar Energy & Waster Heat from Source	N/A	Energy Efficiency of system	Novel design for thermal desalination along with green house food production called as “bubble-green house” is proposed based on multistage bubble column evaporator, condenser using low grade renewable energy sources.
Abd-ur-Rehman et al. [53] 2015	Experimental	Bubble column	Not mentioned	Water heated	Solar energy	N/A	Optimum performance parameters of Humidifier	-Identification of humidifier’s optimum operating conditions -Minimum pressure drop occurs at small water height with minimum superficial velocity. Vapor density in the bubble column humidifier increased with water temperature and air velocity.
Sharqawy et al. [54] 2015	Theoretical Modeling & Experimental	N/A	Bubble column	N/A	Heating coils	EES	Dehumidifier performance	-Studied experimentally the effect of operating pressure on dehumidifier performance & developed a new mathematical model to represent heat and mass transfer inside dehumidifier in relation with its effectiveness. -Superficial velocity favors the heat transfer and effectiveness while pressure decreases effectiveness -Model is developed based on ϵ -NTU, suggested that bubble column dehumidifier model as HX with one fluid in phase change
Abd-ur-Rehman and Al-Sulaiman. [55] 2015	Theoretical Modeling	Bubble column	N/A	N/A	N/A	EES	Heat transfer and its coefficients	-Model of bubbler humidifier has developed to predict the heat and mass transfer while efficiency of humidifier in terms of the water vapor contents in the air stream at its outlet is reported. -Results showed increase humidification with the increase in temperature difference between water and air stream.

K. M. Chehayeb et al [56] 2014	Numerical solution algorithm	Packed bed	Multi tray Bubble column	N/A	N/A	MATLAB	GOR & RR	-Model reveals that HCR should be unit for maximum energy and water recovery -Depicts the need of control system for mass flow rate ratio in order to keep system thermodynamically balanced in off design condition
Tow and Lienhard [57] 2014	Theoretical Modeling & Experimental	N/A	Bubble column	Water heated	Electrical Heater	EES	Heat flux and effectiveness	-03 coils of different length are used as parameter to find its effect on coil heat flux and parallel flow effectiveness for single stage dehumidifier column. -Theoretical model based on thermal resistance network is developed by neglecting gas side resistances. Mode is in closed agreement with experimental results of 8%
M. Shahid and R.Pashley. [58] 2014	Experimental	Bubble column Evaporator	Not mentioned	Air heated	Gas heater	N/A	Vaporization	-Showed that efficiency of evaporation of water in bubble column increased by Inhibition of bubble coalescence caused by the presence of salts -Investigated the effect of bubble temperature on coalesces
Tow and Lienhard. [59] [62] 2014	Experimental	N/A	Bubble column	N/A	N/A	N/A	Heat Transfer coefficient	- Heat transfer coefficient is developed through experimental method for shallow bubble column in HDH desalination. -Developed correlation present the effect of sparger on heat transfer coefficient identifying the critical height.
M.T. Ghazal et al.[61] 2014	Experimental	Rectangular X-sectional bubble column	N/A	Air heated	Solar heater	N/A	Feasibility of Direct contact of carrier gas with water	Examine that the merging of heating of air, water and humidification process in compact system is efficient for HDH system. It was found that for an average intensity of solar radiation of 700 W/m ² and a mass flow rate of 12.6 kg/h of air; the amount of water evaporated was 0.75 kg/h on a square meter basis. Introduction of a reflector mirror at the bottom side of the humidifier increased the average absolute humidity by 32%.
O.M Muntasheri [30] 2014	Theoretical modeling & experimental	Multistage bubble column	N/A	N/A	Heater	EES	GOR, VPR & HCR	-A novel design is modeled which is the improved version of varied pressure HDH system incorporating turbocharger -Experimental study of multistage bubble column humidifier is carried out for selection of optimum operating parameters.
M. Schmack et al. [63] 2013	Experimental	Bubble column Evaporator	Heat Exchanger	Water heated	Electrical Heater	N/A	Water quality and distillate productivity	A novel design of porotype condenser with bubble column is proposed to show that energy demand is low in addition to High quality of water.

Tow et al. [64] 2013	Theoretical Modeling & Experimental	N/A	Bubble column	N/A	N/A	EES	Coil heat flux and Effectiveness	Experimental study is carried out for finding the effect of coil area, coolant and air temperature of maximum heat flux and parallel flow effectiveness
Tow et al. [65] 2013	Theoretical modeling	N/A	Bubble column	N/A	N/A	EES	Algebraic equation for heat & Mass transfer driving force	-Model is developed for bubble column dehumidifier as one single-stream heat exchanger and one single stream heat and mass exchanger in contact with isothermal column liquid. Instead of LMTD for stream with both heat and mass exchange, a log-mean mass fraction difference was shown to be the driving force.
Narayan et al. [19] 2013	Theoretical Modeling & Experimental	N/A	Bubble column	Water heated	Electrical heater	EES	Modeling of mass and Heat transfer mechanism	Initiated the concept of using Bubble column as dehumidifier for high heat transfer rate for the condensation of vapor in noncondensable gas. Closed agreement found between the developed model for heat flux and experimental study.
S. Lam [66] 2012	Experimental	N/A	Multistage bubble column	Water heated	- Electrical heater	EES	Heat transfer rate & dehumidifier effectiveness	-Experimental Study for Single Stage Dehumidifier for reducing the thermal resistance and then modified the setup for 03 stages to achieve the effectiveness of 89%
L. Zhang et al. [67] 2011	Experimental	Bubble column	Horizontal heat exchanger	Air heated cycle	Solar Air heater	N/A	GOR & Cost of Water	-Humidification increases with the increase of air flow rate (high power required) and Vice versa -Water and air temperature in humidifier increase the GOR and decreases the cost production
L. Zhang et al. [68] 2011	Experimental	Bubble column	Heat exchanger	Air heated cycle	Electrical air Heater and solar energy	N/A	Influence of parameters	-Effect of Water temperature and its height, sieve plate diameter and no. of holes in it are studied on humidification process -As the air flow rate increased, the quantity of air humidification, the pressure drop of air passing though the holes and the water level above sieve plate were all increased; as the temperature of air and water in the humidifier increased.
M.Francis and R. Pashley [17]	Experimental	Bubble column	Condenser	Air heated cycle	Heat recovery through	N/A	-	Experimentally two systems are tested for utilization of bubble column in thermal desalination a separate chamber for

2010					warm sea water			condensation & evaporation and second one combining both in one bubble column
S.A. El-Agouz . [14] 2010	Experimental	Evaporative Chamber	Coil type heat exchanger	Water Heated	Electric water Heater	N/A	Principle operating parameter	-Analyzed the desalination performance of the system by considering brackish water temperature, its level and flow rate. -The productivity of the system increases with the increase of the brackish water temperature and the decrease of the airflow rate.
El-Agouz and Abugderah [13] 2008	Experimental	Bubble column as Evaporative Chamber	N/A	Air and water heated	Electric water and air heater	N/A	Parametric study of on Humidification efficiency	Experimentally analyze the humid air behavior through single-stage of heating-humidifying processes.

3.1 Literature on Experimental Worked on Bubble Column HDH

There is a lot of work has been done experimentally on HDH desalination using bubble columns. It consists of mostly the parametric study of operational variables for selection of optimum design with operating conditions.

3.1.1 When Humidifier is bubble column

First the humidifier as bubble column is reviewed in detail along with each parameter that effect its performance as shown in Figure 3-2. The details of experimental setup of work is shown in Table 3-2

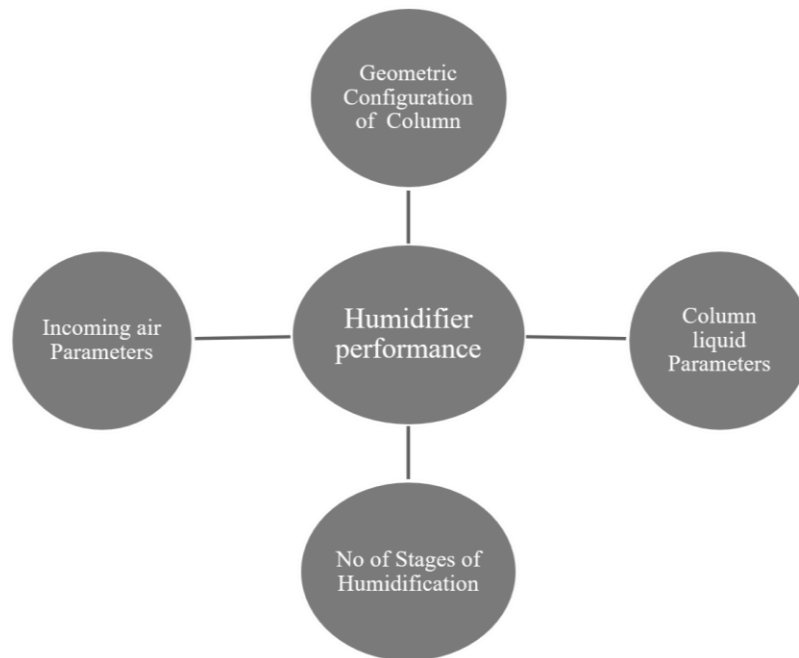


Figure 3-2 Performance parameters of Humidifier bubble column

3.1.1.1 Effect of Liquid Level Inside Column

There is wide spectrum of study that has been carried out experimentally for the variation of liquid present inside the humidifier bubble column. Mostly for humidifier, relative humidity at outlet is analyzed as a performance parameter against the liquid height. While some have investigated the effectiveness. Liquid height has not significant effect on absolute humidity or relative humidity of air leaving the humidifier. However, few authors noticed some changes in performance if the height is lower than critical height.

L. Zhang et al. [61] measured the performance of humidifier in term of relative humidity of air at outlet of bubble column humidifier . It was shown that that there is minimum height of liquid level inside the bubble humidifier for having 100% relative humidity at outlet. It was examined that from 6-18cm, relative humidity is 100% but below the critical height i.e. 6cm for that case the relative humidity as well humidifier performance decreases. Contrary to this L. Zhang et al [62] in another experimental study, noted that level of water should be low for less air resistance and lower pressure drop. From height of 2cm to 6cm, relative humidity is 100% and even below the height of 2cm air exit the same saturation condition at outlet. O.M Muntasheri [26] investigated the effect of column height in term of effectiveness of humidifier. Three heights from 1cm -5cm is considered at three superficial velocities. It was demonstrated that effectiveness is slightly increased for case of higher height. This is because of foam formation of countless bubbles that increases heat and mass transfer as well as contact time and area.

H. M. Abd-ur-Rehman [46] studied the effect of height for the air pressure drop inside the humidifier. Three column heights as 1,3 and 5cm at four superficial velocities ranging from

15cm/s to 30cm/s are considered in parametric study. Minimum pressure drop was achieved when height is lowest among all as well as at low superficial velocity i.e. at combination of 1cm and 15cm/s. Although pressure drop is lowest at this superficial velocity, but this was not applicable because with such low flow rate of air quite less to overcome the static head of water liquid which resulted in leakages from perforation. However the extension of this work [40] showed that humidity (absolute) leaving the humidifier is unaffected by the water column. A. Khalil et al. [44] measured relative humidity at column's outlet for describing humidification efficiency at four different heights of columns from 5cm-20cm with fixed airflow rate and sieve configuration as 1mm holes (1200 numbers). The relative humidity is found to be independent of column's liquid height through the available solar radiations. Its remain constant at 100% at each value of height. In addition to this, study is extended for measuring the temperature difference of inlet and outlet across humidifier for both streams air and water. The difference of temperature for water is nearly constant means that outlet condition of water is not affected by water column height. However, at same levels of heights the air temperature difference is substantial i.e. in range of 20°C-25°C in comparison with water difference of temperature which is around 2.5°C This is because of difference in specific heat of water and air. H. Liu et al. [43] used the height of 3-7 cm inside humidifier and not immersed completely heating coil due to limitation of overflow of water with air. Effectiveness and total heat transfer rate are determined for two cases of superficial velocity at 3cm/s and operating pressure 0.48 bar while the other one is 7cm/s and pressure is 0.83 bar. The calculation of NTU for effectiveness and heat transfer from experimental data plotted with static head of liquid clearly reveal that level of liquid is independent from

effectiveness and so from the total heat transfer. Interestingly later one is almost double for lower set of pressure albeit constant for all liquid level while effectiveness is lower for lower pressure.

P. Behnam et. al [41] used bubble column Humidifier in a novel configuration with evacuated tube collector and Heat pipe They claimed to study the effect of liquid level in humidifier in presence of heat pipe condenser unlike others that have studied with dedicated electrical heaters inside the humidifiers. Productivity was measured in kg/hr for three different level of water. Three different heights (5cm, 7.5cm & 10cm) were tested with approximately equal solar intensity. It was found that for complete submerged of heat pipe condenser (for 7.5cm), the heat transfer is maximum and hence productivity, while as height decreases, heat transfer decreases (for 5cm). While increasing the height beyond the complete submerged level of heat pipe, productivity decreases due to increase in heat transfer losses and thermal capacity of system. Therefore, maximum productivity of 0.32kg/hr at peak hours of day is obtained for the complete submerged case of heat pipe, leading to quite similar conclusion draw from other authors for case of critical height of heating coil.

3.1.1.2 Effect of incoming air

The effect of incoming air on performance is investigated in term of flow rate or superficial velocity. Higher superficial velocity or flow rate increases the bubble formation and gas hold up in bubble column Humidifier. Relative humidity and absolute humidity of air at exit of the humidifier is directly increasing with the increase of air superficial velocities. Same conclusion has been drawn by most of authors that increasing air flow rate creates

turbulence inside the humidifier which increase the no of bubbles and in turn large contact interfacial area and thus provide greater heat and mass transfer coefficient. However, with increase of air flow rate the pressure drop across the humidifier is also noted. S.A. El-Agouz [63] showed that air incoming flow rate has minor effect on fresh water production. It slightly increases the distillate quantity. This increase in the system output is due to the decrease difference between the air dry-bulb/wet-bulb temperatures. The bubble radius increases to account for the larger flow rate. The mean radius of the bubble produced can be modeled as an exponential function of flow rate. Increase of the number of air bubbles increases the rate of interface area, which increase the productivity. At 80°C of saline water temperature productivity of the system is 3.41, 6.8, 7.45 and 8.22 kg/h at 4.2, 8.4, 12.6 and 14kg/h, respectively. L. Zhang et al[62] in another study showed that increasing the air flow rate increase the relative humidity and remains at 100%. If the height of water is unchanged, the total pressure drops increases with increase of air flow rate. For fixed flow rate of air, the lower level of column's liquid is favorable for low pressure drop.

O.M Muntasheri [26] studied performance in form of effectiveness of humidifier with relation to superficial velocity. Velocity is varied from 15cm/s to 25cm/s and found that effectiveness increases slightly. With the increase of superficial velocity, water height and gas holdup are effected and increases with superficial velocity and because of this time and surface area of contact increases turning to high mass and heat transfer[65]. H. M. Abd-ur-Rehman et al. [46] experimentally examine the humidifier performance combining the solar collector as a heating source . They measured humidifier performance in term of vapor density in g/cm^3 at outlet of humidifier by varying the superficial velocities of incoming air from 20 cm/s to 30m/s. The increase in air superficial velocity contributed

significantly in increasing the vapor density in the moist air at the exit of humidifier from 36 g/cm³ to 42 g/cm³ at peak hours of solar radiation.

Later on A. Khalil et al. [44] experimentally demonstrate the effect of airflow rate in humidifier performance. They investigated the effect of air mass flow rate on performance parameters as pressure drop, temperature drop (of air and water), humidified efficiency and on system productivity. Three different air flow rates were considered as 11, 13.2, 15.4, and 17.7 kg/h through 1mm holes diameter and 1200 no holes of sieve plate. Firstly, pressure drop across the humidifier is measured in Kpa by changing the inlet pressure from 9 to 23Kpa and keeping outlet as atmospheric pressure at different column heights. It was found that higher the flow rate of air increases the pressure drop across the humidifier. High air flow rates lead to high rise velocity of bubble and its size. This make greater interface area and hence humidification increases. Similarly, the temperature differences across the humidifier of air and water at mentioned flow rates were measured. It was shown that temperature difference is nearly same at all flow rates irrespective of height, meaning that efficient heat exchange is taking place. Air is able to get humidified efficiently as its mass increases. While the extracted water amount is found to be get slightly influenced by airflow rate. H. Liu et al [43] studied humidifier performance at sub atmospheric pressure by varying the superficial velocities. Pressure are varied from 1.03bar to 0.48 bar with column liquid level at 5cm and fixed incoming air at 15°C. The variation of superficial velocities from 2-20cm/s depicts that the temperature of air leaving the humidifier decreases. The increment of superficial velocity leads to enhanced turbulence which increased the heat transfer and hence reduced the exit air temperature. However, for a given superficial velocity, exit air temperature is slightly increased with decrease in pressure from

1.03 bar to 0.48 bar. This was mathematically claimed that decrease of mass transfer coefficients reflected in this way. H. Liu et al [43] calculated the total heat transfer and effectiveness of humidifier and plotted against the superficial velocity. Humidifier operated well for high superficial velocities. Heat transfer increased from 50W to 160 W from the interval of 2-16cm/s. This is due to fact that increment of superficial velocity means high flow rater which means high sensible heat transfer. However, effectiveness found to decrease significantly from 0.8 to 0.6 for same set of superficial velocities. They also concluded the humidifier performance in term of operating pressure of humidifier and observed that heat transfer increases for lower operating pressure. It means that more water vapor molecules humidified to air stream while the effect of pressure is slightly on effectiveness hence mass transfer coefficient is slightly increases by operating pressure.

3.1.1.3 Effect of air temperature

The air temperature inlet to humidifier can be controlled from heating source. Increasing temperature of air provide high humidification efficiency or thus productivity. Below is the experimental observation relating the humidification efficiency with air inlet temperature.

Incoming air to inlet of bubble column Humidifier effect the relative humidity at the outlet. S.A. El-Agouz et al.[64] studied that specific humidity increases from 6.8% to 11% for temperature increment of ambient air of 40°C. L. Zhang et al.[61] showed humidification increases with air temperature. For temperature increment of 10°C the humidification substantially increases from 901 g/h to 1762.7 g/h i.e. 95.6% P.Behnam et al. [41] studied novel configuration of desalination system as with bubble humidifier connected with evacuated tube collector (ETC). They studied that the increase in air inlet temperature from

the ETC lead to increase in water temperature inside the humidifier which gives high evaporation thus high heat transfer and productivity. For the case of 7.5cm water depth inside the humidifier, the air temperature at inlet of humidifier is approx. 38°C (for peak hours) which gives productivity of 0.33kg/hour. While for inlet air temperatures of 35°C and 32°C the productivity is reduced at 0.28 and 0.26 kg/h for 5cm and 10cm respectively.

3.1.1.4 Effect of Saline temperature

Saline water's temperature is important factor for humidifier as indicated by various work performed by researchers. The increment of saline temperature directly increases the humidification efficiency or the vapor density at the outlet of humidifier. S.A. El-Agouz et al.[64] studied the humidifier as separate unit experimentally, found that humidification efficiency strongly influenced by saline water temperature inside Humidifier. The ability to carry water vapor increases with the temperature. Range of temperature considered is from 30°C to 70°C at air velocity of 6m/s at which humidification efficiency increases from 78% to 91%. Later S.A. El-Agouz [63] studied in a separate experiment that productivity slightly affected by water temperature. For fixed flow rate of air, productivity of water in maximum 3 kg/hr for range of saline water temperature of 50-80°C L. Zhang et al[62] showed that saline water temperature is an important factor for humidification efficiency. Increasing the temperature with increment of 10°C, humidification efficiency increases upto 80%. H. M. Abd-ur-Rehman et al. [46] measured the water temperature effect in term of vapor density (air humidity) at outlet of humidifier. The presence of Fresnel lens in their experiment increase the water temperature from 33°C to 48°C because of concentrated solar radiation. A. Khalil et al. [44] measure the amount of extracted water against the water temperature inside bubble column humidifier. At fixed flow rate of air of

15.4 Kg/hr and at all heights. The water extracted amount in g/kg of air is found to increase rapidly as the temperature increases. It increases by 51.4, 76, 73, and 60.7% as temperature rises from 60 to 70 °C, at various flow rates of 11, 13, 15, and 17 kg/h (H=20cm).

3.1.1.5 Effect of sieve Plate configuration

Selection of optimum design of perforated plate is important in HDH desalination. Two important aspects of sieve plate configuration are to reduce the air pressure drop and zero leakage of liquid through perforations. Several authors investigated the different sieve plate geometric configuration, design and its effect on bubble column performance.

L. Zhang et al [62] studied the effect of number of holes on humidifier performance. In their experiment total number of holes are 91 and take two different measurements by defining open rate of holes as 1.12% and 2.52% and found that total air pressure of air dropped when the open rate of sieve plate holes increases. This also reduced the blower power consumption due to less air resistance.

O.M Muntasheri [26] experimentally studied performance of single humidifier with 02 different designs of spargers having 27.7 cm diameter and 2mm holes diameter with open ratio 0.68 % and 2.2% and 130 and 425 number of holes respectively. Humidifier effectiveness is 2.5% higher for the sparger of fewer holes and low open ratio than the other one however the pressure loss is much higher which is around 400% in comparison with other. Less number of holes lead to high air jet velocity passing through holes turning to high dynamic pressure loss

H. M. Abd-ur-Rehman et al. [46] studied about geometrical configuration of sieve plates by using three different designs of 105, 105 and 149 number of holes with hole diameter

3mm,2mm,2mm and open ratio 0.77,0.33,0.49 respectively. The design with less number of holes 105 nos. has minimum pressure drop due to greater diameter 3mm but due to large open ratio water leakage is noticeable. While Reducing the diameter of sparger from 3mm to 2mm reduces the water leakage but substantial pressure drops due to low % of open ratio. Hence the optimal design was found as one with large no of holes and greater open ratio but with reduced hole diameter. A. Khalil et al. [44] studied effect of three perforated plates with holes diameter as 1mm,3mm and 5mm each having 1200 holes on the GOR, humidified water quantity and humidifier's efficiency. The amount of humidified water reduces with increment of sieve holes diameter and noteworthy 5 mm orifice diameter is the case for lowest productivity. It was noted that at 5 mm sieve's flow began to shape in jet. This give the best performance is observed from sieve with 1mm hole diameter.

3.1.1.6 Effect of Inlet Air Relative Humidity

Most of the experimental study has been carried out in laboratory environment. Due to controlled volume conditions there is not much difference in inlet air condition to humidifier such as effect of ambient air temperature and inlet air relative humidity. However, some authors have studied the effect on inlet air humidity on humidifier performance as it is an important parameter for integration of humidifier with dehumidifier

3.1.1.7 Effect of Carrier Gas on Humidification efficiency

In almost all the studies of bubble column humidifier/dehumidifier especially those that deals with desalination techniques used air as carrier gas. Air is widely economically available and has low thermal capacity. However, M. Taseidifar [44] compared the performance of bubble column evaporate by using carrier gases as CO₂, He, Ar against air.

He used 0.5m NaCl solution as salt aqueous solution in which all these three gases pass at fixed inlet temperature of 150°C. The total loss of solution is measured using weight balance. The ideal water vaporization is calculated by addition of all equilibrated water vapor densities at measured column temperature multiplied by the supply air flow rate of each gas. This gives desalination efficiency, on comparison Ar was found to be lowest with 0.75 vaporization efficiency while CO₂ and Air are second in number with 1.03 and surprisingly, He has the highest vaporization efficiency among all as 3.3 times higher than the expected theoretical water vaporization rate. He carrier gas was observed to produce significantly smaller bubbles due to its slightly lower surface tension [66].

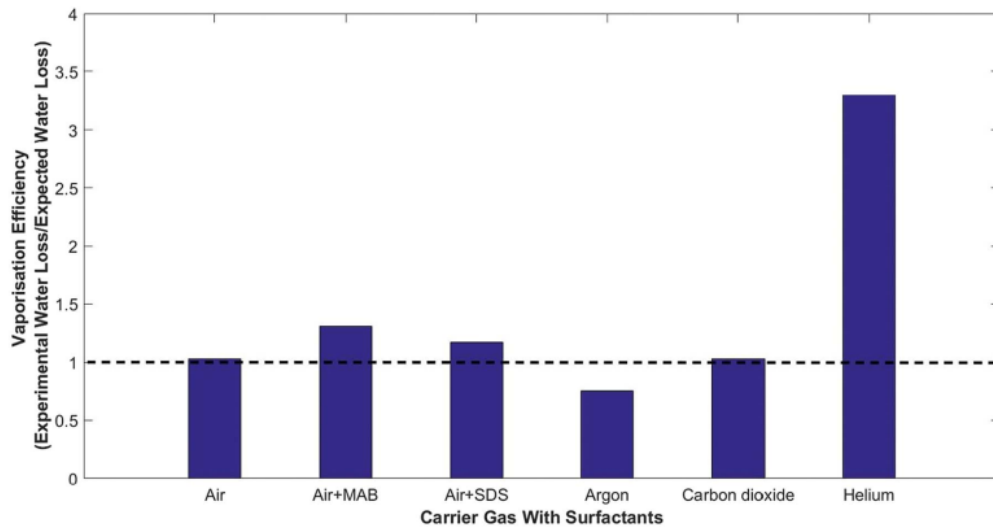


Figure 3-3 Comparison of different carrier gases for vaporization efficiency

Table 3-2 Experimental details of the work of bubble column humidifier

Author	Bubble humidifier	Input Energy	Carrier Gas flow rate	Temperature	Column Liquid Level	Outcome
S.A. El-Agouz et al. [64]	200 x 500mm H=700mm	2KW (Electric Heater)	2.5m/s -7.5m/s	30°C to 70°C	4.5cm-21cm	Specific humidity = 222 gw/kga (Max)
S.A. El-Agouz [63]	400 x 300mm H=250mm	3.6KW (Heater)	4.2-14 kg/hr	50°C to 80°C	20cm-60cm	Max productivity = 8.22 kgw/h
L. Zhang et al.[61]	D= 200mm H=500mm	8.2W (blower)	0.5 – 2.5 m ³ /h	40°C to 80°C	6cm-18cm	GOR =2.3
L. Zhang et al[62]	Di= 2mm	-	2.56 m/s -4.6 m/s	30°C to 60°C	2cm-16cm	-
O.M Muntasheri [26]	D = 27.7 cm	800W (air blower)+7.5 KW (water heater)	4.5 m ³ /min	-	1-5cm	GOR = 30 & Effectiveness = 89%
M.T. Ghazal et al.[54]	Width = 50cm Length = 100cm		8.2,10.6,12.6 kg/h	25°C -55°C (Air Temperature Outlet)		<u>Setup 1: W/O Sieves</u> Humidification efficiency = 95-100% Humidifier effectiveness = 70% Productivity = 0.6 Kg wat/ kg air hour <u>Setup 2: With Sieves</u> Humidification efficiency =100% Humidifier effectiveness = 95% Productivity = 0.62 Kg wat/ kg air hour

						<u>Setup 3: Addition of Reflector mirror in setup2</u> Humidification efficiency =100% Humidifier effectiveness = 98% Productivity = 0.90 Kg wat/ kg air hour
Muhammad Shahid et al. [51]	D = 120 mm		23 L/min	150°C -275°C (Air) 56°C (Water)	50 mm	
M. Schmack et al [56].	H= 50cm D= 12cm		10 L/min	55°C (inside Colum)		Water productivity =88ml/h
H. M. Abd-ur-Rehman[46]	A =30cm ² H = 40cm	400W (air blower)+ 2000w/m ² (solar Radiation)	10-15cm/s		1-20cm	
A. Khalil et al. [44]	580 mm × 580 mm cross section H=900 mm t=10 mm	200w/m ² -900 w/m ² (solar Radiation)	11, 13.2, 15.4, and 17.7 kg/h	45°C -70°C	5, 10, 15, and 20 cm	Enhancement in humidifier GOR =0.53, Daily Efficiency =63% & Daily productivity = 21kg
Huicheng Liu et al [43]	D = 10cm H =25cm t = 10mm	Heating inside humidifier = 1kw	0.5–4.5 ft ³ /min Super facial velocity 2-20 cm/s		3,5,7cm	

M. Schmack et al [24]	D = 12cm H =50cm		13.5 l/min			
P.Behnam et. al [41]	Rectangular shape X-section 90cm x 15cm H =30cm	10 ETC and Thermosyphon pipe with absorber area 0.1m ² and Heat pipe material Copper	Inlet air to Humidifier = 4L/min,6L/min, 8L/min.	30°C -45°C (Air temperature)	5cm,7.5cm & 10cm	Highest Productivity = 0.33kg/hr Optimum water level =7.5cm (Complete Submerged case) Optimum Flow rate = 8L/min
H. M. Abd-ur-Rehman et. al [40][35]	300 mm X 300 mm t = 10mm	400W (air blower)+ 2000w/m ² (solar Radiation)	25cm/sec		1cm,3cm & 5cm	Three stages humidifier performance is better and 23% higher in absolute humidity compared to single stage
T. Rajaseenivasan et al. [39]	D = 3.9cm L =45cm	Solar collector + Turbulator	0.014-0.018 kg/s	58°C -65°C	8cm	Three configurations of system equipped with bubble humidified is tested which shows that maximum productivity is 23.92 kg/m ² for dual purpose collector. Moreover, Humidifier efficiency is also higher for this design around 85-92% that increased with air flow rate.
K. Srithar et. al [36]	A =0.5 x 0.5 m ² (shape similar to solar still) t =2mm		0.014-0.018 kg/s	55°C -68°C	8-12cm	
Penghui Gao et al. [32]	D= 190cm L=210cm	Solar radiation of 800W/m ²	3.0 m ³ /s- 8.3 m ³ /s	20°C-35°C (water Temp)		System is effective with Thermal energy consumption of 700 kj/kg

3.1.2 Dehumidifier as Bubble Column

The main performance parameters of interest in dehumidifier is the heat flux between the coolant coil and pool liquid and the heat and mass transfer coefficients. The Figure 3-4 shows the performance parameter of bubble column dehumidifier Usage of bubble column as dehumidifier enhances heat and mass transfer with less equipment size

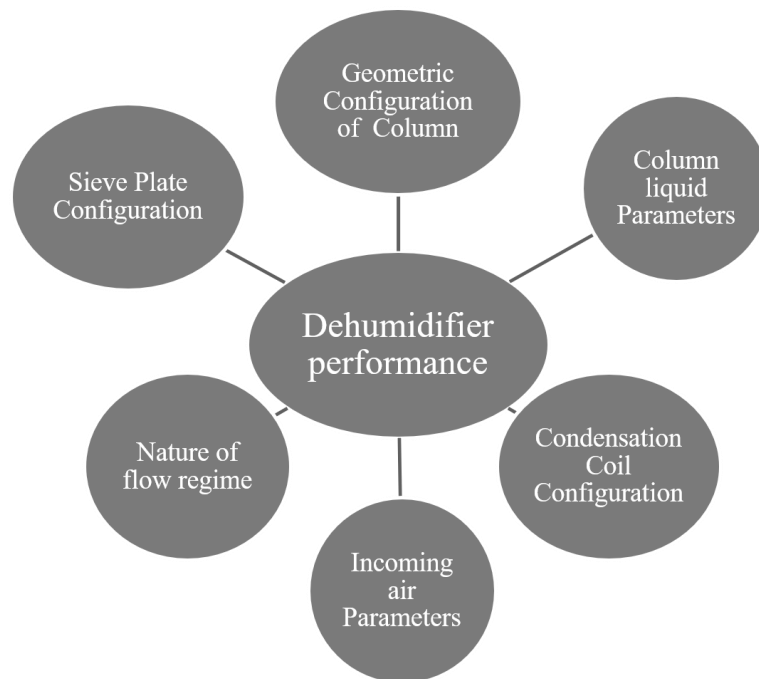


Figure 3-4 Performance parameters of bubble column Dehumidifier

All literature review for bubble column Desalination showed that G. Prakash Narayan [59] was the first Who utilized the bubble column as dehumidifier experimentally for countering the additional resistance to vapor condensation heat and mass transfer due to presence of non-condensable gases. Square dehumidifier of 304.8 mm^2 with height of 457.2mm having cooling coil of 6.3mm dia. cooper diameter is filled with water. Air after

passing through humidifier and heated from 1.5KW heated allowed to bubble through bottom of dehumidifier with four different design of sparger plates.

3.1.2.1 Effect of Superficial Velocity

Previous studies has been carried out to find the effect of superficial velocity on either mass transfer or heat transfer in bubble column but they have not consider the both effect simultaneously[67] In general case of bubble column[68] heat and mass transfer coefficient are high for higher superficial velocities of air. The rate at which heat and mass transfer changes with superficial velocities depend upon the flow regime i.e. the flow is homogenous bubbly flow or churn turbulent region[69].

Narayan et al [59] experimentally studied the effect of simultaneous heat and mass transfer with condensation. For the case of Narayan et al the superficial velocity (3-8cm/s) lies in range of bubbly flow and found that for fixed column height and fixed inlet condition, the heat flux increases almost linearly with the increase of superficial velocity. E.W. Tow [50] chose airflow rate for finding its effect on heat transfer rather than superficial velocity. Three flow rates from 1.5 L/s to 2.5 L/s were used at same air temperature of 48°C and liquid height of 20cm, concluded that heat flux increases with air flow rate while effectiveness decreases. The increase in heat flux is due to the increase in maximum single-stage heat transfer rate. The decrease in effectiveness occurs because the thermal driving force and resistance across the coil are largely unchanged while the maximum heat transfer rate increases. Mostafa H. Sharqawy et al.[47] investigated the effect of superficial velocity in conjunction with operating pressure of bubble column dehumidifier on its performance. Keeping the inlet temperature of air fixed at 45°C, water temperature at 15°C and water

height at 5cm, the superficial velocity is varied from 3-7cm/s at each set of operating pressures of 15 psi, 20psi, 25psi & 30psi respectively. It was observed that temperature of outlet air from dehumidifier decreases with superficial velocity while increase with the pressure. Increasing the superficial velocity increases the turbulent dissipation inside the column which increases the mass transfer coefficient. This increment of mass transfer coefficient leads to increase of total heat transfer which represent in form of decreased values of air exit temperature. While at a certain superficial velocity, increment of pressure means increase in gas mass flow rate due to greater density of air. More mass of air inside the dehumidifier result in more heat transfer therefore low difference b/w air inlet and departure enthalpy which is the reason for slight increase in air exit temperature with pressure. While H Liu et al [43] studied effect of superficial velocity on dehumidifier performance operated at elevated pressure. Pressures are varied from 1.03bar to 2.07 bar with column liquid level at 5cm and fixed incoming air at 45°C. The variation of superficial velocities from 2-20cm/s depicts that the temperature of air leaving the dehumidifier decreases. The increment of superficial velocity leads to enhanced turbulence which increased the coefficient of heat transfer and hence reduced the exit air temperature while it was found that the with the increase of airflow rate the sensible heat transfer increases due to which the coolant leaving the dehumidifier show increases temperature. However, for a given superficial velocity, exit air temperature is slightly increased with increase in pressure from 1.03 bar to 2.07 bar. This was mathematically claimed that increase of mass transfer coefficients with pressure reflected in this way.

3.1.2.2 Effect of Inlet Mole Fraction

Narayan et al [59] studied the effect of inlet mole fraction on heat transfer coefficient inside bubble column dehumidifier. At fixed values of superficial velocities, liquid height the inlet mole fraction is varied from 10 to 25% which is 3.6 to 9 times the lower than regular condenser. It was noticed that the effect on heat flux is linear and much weaker effect than literatures reported for film condensation of standard dehumidifiers.[70]

3.1.2.3 Effect of Liquid height

The liquid height should be minimum because it creates hydrostatic head pressure that must be overcome by the supply of air vapor mixture. At the same time height should be such that which completely immersed the cooling coil so that energy cannot be waste. Authors called this as critical height.

Narayan et al [59] found changing the liquid height has no effect on heat flux. Reducing the height in range of from 250mm to 150 mm (critical height) give constant value of heat flux to the coil. Hence liquid height should be kept as low as possible just to cover the coil. This claim was found to be valid only provided that minimum height should be order of bubble diameter. E. W. Tow [57] in his experiment found the same result that Narayan et al [59] postulate about the effect of liquid height on performance of Dehumidifier . E. W. Tow [57] studied the variation of column height for the effectiveness and coil flux and showed that for above the critical height effectiveness remain the same around 0.86 as constant value and heat flux of 23KW/m^2 [50] . The critical height in case of E. W. Tow is 4cm which is much lower than Narayan.

E.W. Tow [50] further added that lowering the liquid level below the critical one decreases the effectiveness because much of the heat transfer in early bubble residence. However Mostafa H. Sharqawy [47] investigated the effect of liquid column height from minimum 3cm to 7cm at two different superficial velocities 3cm/s and 7cm/s with column temperature of 15°C and air inlet temperature of 45°C. It is worth mentioning that coil height itself is 9cm and hence the aerated height during operation submerged the coil. Nearly heat transfer is constant at value of 200W and 50 W at mentioned superficial velocities respectively. However slight increase in effectiveness about 2% is found and authors claimed it within the uncertainty limit of experiment. H. Liu et al [43] used the height of 3-7 cm inside dehumidifier and not immersed completely heating coil due to limitation of overflow of water with air. Effectiveness and total heat transfer rate are determined for two cases of superficial velocity at 3cm/s and 14cm/s operating pressure 1.03 bar while. The calculation of NTU for effectiveness and heat transfer from experimental data plotted with static head of liquid clearly reveal that level of liquid has no effect on effectiveness and total heat transfer. Interestingly heat transfer is almost double for high superficial velocity e.g. 200KW albeit constant for all liquid level while effectiveness is almost 0.93 for 14cm/s in comparison with 0.91 for 3cm/s

3.1.2.4 Effect of Impact of Bubble on Coil

Study is also carried on heat transfer for impact of bubble directly hitting the coolant coil. This will increase the heat flux of coil due to mainly sensible heat transfer between the moist air and coil. Narayan et al [59] carried out the study of bubble coil impact by using serpentine coil and circular coil for intentionally maximize the bubble coil hitting. It is found that heat transfer rate is almost twice for the case of bubbles hitting the coil.

Contrary to Narayan, E. W. Tow et al. [50] studied the impact of bubbles on coil by changing the sparger rather than changing coil as Narayan did. They highlight that Since coil surface area effect the heat flux so it must be remain constant when analyzing one parameter. Similarly, the effect of curvature of coil and its geometry also influence the heat transfer coefficient because of secondary flow induces inside coil. This condition also overlooked by Narayan. Hence Tow kept the surface area of coil constant as 900 cm^2 while studying the influence of bubble coil impact and changes the sparger configuration only. What Emily W. Tow et al evaluated is against the results given by Narayan. He concluded that bubble impact has no significant contribution in improvement of bubble column dehumidifier. The effectiveness and heat flux found to remain same with and without impact at three different flow rates of air.

3.1.2.5 Effect of No of Stages

Rare study is done on multistage bubble column dehumidifier. Steven [60] compared the single stage dehumidifier effectiveness with multistage (03 nos) and experimentally showed that effectiveness is 90% for multistage dehumidification. The reason behind this attribute that the flow regime inside the dehumidifier is not homogenous bubbly as mostly in case of single stage dehumidifier.

3.1.2.6 Effect of Coil Area

Some authors as E. W. Tow [57] also studied the effect of coolant's coil area of on the heat flux and effectiveness (parallel flow) of Dehumidifier in search of optimal coil area. Three coil of area 26 cm^2 , 200 cm^2 and 900 cm^2 were tested with same inlet condition of moist air and found that effectiveness increases with coil area in asymptotic manner and

reached to unity at largest coil area. In contrary to this heat flux shows decreasing trend with coil area exponentially. Hence for optimum dehumidifier performance gain of heat flux comes with loss of effectiveness and selection of optimum coil area should be based on the complete performance of HDH system. Later on Emily W. Tow et al. [50] studied the effect of coil length while keeping the diameters of coil constant. Same conclusion is drawn for effectiveness and heat flux i.e. effectiveness increases with coil Length and reaches to unity at very larger length (around 3m) while heat flux decreases exponentially

3.1.2.7 Effect of Moist Air Condition

E. W. Tow [57] , in his experimental study also analyzed the effect of moist air inlet temperature on dehumidifier effectiveness and heat flux. Heat flux increases sharply with the moist air because of higher enthalpy or energy content while effectiveness decreases with the increase in temperature. Decrease in effectiveness is justified by the fact the actual heat transfer is constant because heat transfer coefficients are constant and not strong function of temperature. While the maximum heat transfer rises to a greater degree because of increase of water vapor pressure at temperature as depicts by enthalpy temperature curve of moist air. Hence the effectiveness decreases. E. W. Tow et al. [50] studied the same thing but for three different coil length and showed that irrespective of coil length , the effect is same as conducted in his first experiment [57].

Table 3-3 Experimental details of the work of bubble column dehumidifier

Author	Bubble Dehumidifier	Coil Detail	Carrier Gas flow rate	Column Liquid Temperature	Column Liquid Level	Outcome
G. Prakash Narayan [59]	304.8 mm ²	Copper coil OD =6.35 mm	3-8 cm/s		15-25.4cm	Heat flux max = 8KW/m ²
E.W. Tow[57]	28cm x 28cm H=36cm	02 nos Copper tubes 9.5mm OD ID1 =8.5 cm ID 2 = 13.5cm	1.56 L/s	44°C to 52°C	4cm-15cm	-Effectiveness = 0.86 (max). -Parametric study of heat flux
Emily W. Tow et al. [50]	28cm x 28cm H=36cm	03 copper coil 9.5mm OD 8.0mm ID L=0.087 m, 0.67m,& 3m	1.5 L/s 2.1 L/s 2.5 L/s	40°C -60°C (air)	4cm-20cm	Tradeoff between heat flux and effectiveness
Mostafa H. Sharqawy et al. [47]	D =120 mm H =250mm	Copper tube 9.5mm OD L =9cm	0.5-4.5 ft ³ /min	15°C	3-7cm	Operate Dehumidifier at high pressure for high effectiveness but with large equipment size.
Huicheng Liu et al [43]	D = 10cm H =25cm t = 10mm	Copper tube 9.5mm OD t =0.9mm L =9cm	0.5–2.25 ft ³ /min			

Table 3-6 Heat Transfer Correlation from Literature of Bubble Column Reactors

Author	Correlation	Derived Formula	Depended parameters	Type	Theory
Deckwer [69]	$St = 0.1(Re FrPr^2)^{-\frac{1}{4}}$	$h = k^{0.5} \rho^{0.75} C_p^{0.5} \mu^{-0.25} g^{0.25} V_g^{0.25}$	Gas superficial velocity, Liquid properties	Theoretical	Kolmogorov Isotropic turbulence, mass transfer surface renewal
Kast [71]	$St = 0.1(Re FrPr^2)^{-0.22}$	$h = k^{0.44} \rho^{0.78} C_p^{0.5} \mu^{-0.22} g^{0.22} V_g^{0.33}$	Gas superficial velocity, Liquid properties	Empirical	
Kolbel & Langemann[69]	$St = 0.11(Re FrPr^{2.5})^{-0.22}$	$h = k^{0.55} \rho^{0.78} C_p^{0.45} \mu^{-0.33} g^{0.22} V_g^{0.34}$	Gas superficial velocity, Liquid properties	Empirical	
Shaykhutdinov et al[72]	$St = 0.124(Re FrPr^{2.5})^{-0.22}$	$h = k^{0.55} \rho^{0.78} C_p^{0.45} \mu^{-0.33} g^{0.22} V_g^{0.34}$	Gas superficial velocity, Liquid properties	Mathematical	
Steiff & Weinspach[73]	$St = 0.113(Re FrPr^2)^{-0.26}$	$h = k^{0.52} \rho^{0.74} C_p^{0.48} \mu^{-0.26} g^{0.26} V_g^{0.22}$	Gas superficial velocity, Liquid properties	Experimental	
Hikita et al.[74]	$\frac{StPr^{\frac{2}{3}}}{(V_g \mu)^{-0.851} \mu_l g}$	$h = 0.411 k^{0.667} \rho^{0.692} C_p^{0.33} \mu^{-0.286} g^{0.308} V_g^{0.149} \sigma^{-0.073}$	Liquid properties	Experimental	Measured Relation
I.S Suh et al.[75]	$St = 0.1 \left((Re FrPr^2) \frac{\mu_b}{\mu_l} \frac{P_v}{\rho_l g V_{int}} \right)^{-\frac{1}{2}}$	$h = 0.0722 \left(k_l C_{p1} \rho_l \{ [V_g (\epsilon_g \rho_g + \epsilon_l \rho_g + \epsilon_s \rho_s)] g (\mu_l \epsilon_l)^{-1} \}^{\frac{1}{2}} \right)^{\frac{1}{2}}$	Liquid properties, Energy input rate	Semi theoretical	Isotropic turbulence, Higbie surface renewal
Zehner P [76]	$h = 0.18(1 - \epsilon_g) \left(\frac{k^2 C_p \rho_l^2 V_f^2}{\mu_l d_p \left(\frac{\pi}{6 \epsilon_g} \right)^{\frac{1}{3}}} \right)^{\frac{1}{3}}$ $V_f = \left(d_p \left(\frac{\pi}{6 \epsilon_g} \right) \frac{\rho_l - \rho_g}{\rho_l} g D_c V_g \right)^{\frac{1}{3}}, d_p = \left(\left(\frac{6 \sigma d_o}{(\rho_l - \rho_g) g} \right) \right)^{\frac{1}{3}}$ $\epsilon_g = \frac{V_g}{0.3 + 0.2 V_g}$		Liquid properties, Column Dimensions, Sparger size, Gas holdup	Qualitative Quantitatively,	Based on Eddies theory
Kim et al.[77]		$h = 0.0722 \left(k_l C_{p1} \rho_l \{ [V_g (\epsilon_g \rho_g + \epsilon_l \rho_g + \epsilon_s \rho_s)] g (\mu_l \epsilon_l)^{-1} \}^{\frac{1}{2}} \right)^{\frac{1}{2}}$	Liquid properties, Energy input rate	Semi theoretical	Isotropic turbulence, Higbie surface renewal

CHAPTER 4

HEAT AND MASS TRANSFER MODEL FOR SHALLOW BUBBLE COLUMN DEHUMIDIFIERS

4.1 Theoretical Background

Bubble columns are widely used in industrial devices where large number of gas bubbles come in contact with a pool of liquid and heat and mass transfer occurs simultaneously. The gas is generally sparged at the bottom of the column through perforated plates or tubes whereas the column is filled with liquid which is static. The main purpose is to produce small gas bubbles in order to have maximum interfacial contact area between the incoming gas and the pool of liquid. Result of large interfacial area is the high heat and mass transfer rates inside the bubble column.

Recent advancement in desalination methods includes the use of bubble columns as humidifiers and dehumidifiers. In these systems, cold air is sparged in a bubble column contained hot water where it is heated and humidified. Then it passes to a dehumidifier containing cooling coil with cold pool of liquid for condensation of water vapor carried by moist air. Heat and mass transfer mechanisms inside the bubble column play an important role in determining the overall system performance. Transfer mechanism occur as long as there is temperature and concentration difference between the vapors carried by air bubbles and pool liquid column. The condensation of water vapor mainly depends upon gas hold

up, bubble diameter, superficial velocity of gas, operating pressure, height of coil and fluid properties.

The heat transfer coefficient during film condensation of air vapor mixture is quite low due to presence of large masses of non-condensable gases (i.e. air along with its gaseous constituents). Unlike conventional humidifiers and dehumidifiers, this is not the issue with bubble column. Here the gas is dispersed in liquid to make bubbly flow instead of contact with any solid surface as in case of other dehumidifiers. Very tiny but larger number of bubbles provide greater interfacial contact area. Accordingly, this leads to improved heat and mass transfer process and reduces sizes of devices, specially of condensing coil used in dehumidifiers. This is the main advantage of using a bubble column in desalination over other HDH techniques. Water vapor carried by air bubble condense on the interfacial area on bubble outer surface and cold pool of liquid. The high amount of latent heat is released from air water vapor mixture to the pool of liquid.

In a complete system of HDH desalination with bubble column, the dehumidifier is more important to investigate. There are two key reasons associated with this. The dehumidifier's pool of liquid is the condensate or the productivity of the system. Hence, the goal is to maximize it and secondly the condensing coil of heat exchanger represents a capital cost, which must be minimized. There are wide range of operating parameters and equipment dimensions that influence the performance of bubble column dehumidifiers.

Experimental studies [8], [26], [61], [62], [65], [78]–[82], [37], [40], [43], [44], [47], [50], [53], [59] have been carried out to study the influence of several parameters such as effect of superficial velocity, inlet temperature, liquid pool height, sparger effect, bubble

diameter, coil length, coil size, coil placement and so on. All these parameters have been studied for evaluations of the heat flux and effectiveness of the bubble column. Heat flux and effectiveness are performance determining parameters and are strong functions of a wide range of bubble column parameters. So, it needs extensive work to study experimentally which makes it quite tough for experimental setups to cover the effect of all parameters. On other hand , analytical modeling of the dehumidifier is used to provide useful parametric results since it helps to cover a wide range of parameters.

Analytical modeling of a bubble column dehumidifier is carried out through the thermal resistance network. Each phase (gas and liquid) is represented by its bulk temperature along with associated thermal resistance network that occur in both of heat and mass transfer inside column. Below are some highlights of the notable work that has been carried out in modeling of bubble column Dehumidifiers.

4.2 Comparison of Previous Models

The Table 4-1 presents a brief comparison of previously developed models along with their limitations and accuracies. The table highlights that some resistances have been considered or neglected by previous investigators along with the assumption employed. The table has been developed from the published work of HDH bubble column.

Table 4-1 Comparison of work of Bubble column Dehumidifier Models

Author	Theory	Model Objective	Brief Description	Assumptions	Thermal Resistance	Limitation/Drawbacks	Accuracy of Model
G. P. Narayan et al. [59] (2013)	Higbie's theory of surface renewal [83]	Heat flux	Nonlinear equation model representing Heat and Mass transfer inside bubble column. Bubble side mass transfer are approximated with simple diffusion equations. Few bubble parameters such as gas hold up, interfacial area and bubble diameter are considered. The inside/outside gas mass transfer Is based on humidity ratios	-No bubble directly hit coolant coil. -Condensation of bubble's moisture occur outside the bubble boundary - Lewis factor is unity for the dehumidifier	-Rcontact (Both Heat & Mass Transfer) Neglected. -Conductive Resistance of coil Neglected	-No correlation given for heat transfer inside the coil. -No detail is given regarding the length of coil used -Poor prediction for effect of bubble diameter on heat flux	20% of satisfaction level with experimental values
Tow et al. [50] (2014)	Thermodynamic Energy balances	Heat Flux & Parallel flow Effectiveness	Simplified model based on only thermal resistances is proposed. Only two resistances among all are considered i.e. the coolant inside the coil and column liquid in touch with coolant coil. Parametric study is carried out for 03 different coil lengths. Parallel flow effectiveness is proposed for single stage dehumidifier. The outstream temperature of coolant and air both assumed to be equal column temperature. Model does not involve any parameter related to column height or coil height (i.e. effect of sparger)	- Heat Transfer above the column liquid is neglected. - Air exit from column at temperature always be equal to Column's temperature -03 Temperatures as Condensation Coolant out and air exit temperature assumed to be equal to column temperature.	-Gas side Resistances are completely ignored -No mass transfer mechanism for concentration difference is considered. -Resistance between the column liquid and condensation water is not considered -No thermal Resistance from coil material.	-Not able to predict the effectiveness of dehumidifier column for short length of coil i.e. below 0.67 cm -Unable to Predict heat transfer above the column' fluid surface -Not applicable for multi tray Dehumidifier -Model is only valid if the condensation, air exit and coolant outlet all temperature are taken to column pool temperature.	2.8% in agreement with Experimental values but the measured heat transfer rate leaving the air stream was 64% higher than that entering the coolant stream

Tow et al [79] (2013)	Energy balance	Maximum coil heat Flux	Simplified model is presented for evaluation of maximum heat flux with the effect of coil area, coolant and air temperature.	Critical height is only few millimeters from the sparger	-Gas side Resistances are completely ignored	Unable to Predict heat transfer above the column' fluid surface -Valid only for single stage dehumidifier effectiveness -No heat and mass transfer coefficients	Presented graphs are in 5% in agreement with Experimental values
Tow et al [58] (2013)	LMTD can't apply because of violation of constant Cp assumption for stream of mass exchange	Development of Log mean mass fraction difference as driving force for stream with mass exchange in replacement of LMTD	Model is developed for bubble column dehumidifier as one single- single stream heat and mass exchange in contact with isothermal column liquid. The seawater side, for which a log-mean temperature difference is appropriate, and the gas side, which has a varying heat capacity and mass exchange, a log-mean mass fraction difference is shown to drive latent heat transfer, and an expression for the mean temperature difference of the moist gas stream is presented. These expressions facilitate modeling of bubble column heat and mass exchangers	-The column's liquid is in isothermal condition. - Bubbles do not have significant thermal interactions with the coil -Heat Transfer above the column liquid is neglected. -Lewis factor is one - Boundary layer outside the bubble is very thin.	-Thermal resistances and their associated heat transfer coefficients have not been presented.	-Dimensionless temperature profile holds for Lewis factor 1 only -Complex set of nonlinear equation that employed iterative procedures	Profile for mean mass driving force is similar to previous results (miller one)
Sharqawy et al.[47] (2015)	NTU method	Parametric Study for effect of Pressure in Dehumidifier	Analytical model is presented showing connection in between the bubble column effectiveness and the NTU which combines the influence of the heat and mass transfer processes.	-Water in column is at constant temperature. -Bubble surface temperature is assumed to be taken	-Resistance from the column liquid to coolant coil is taken as zero.	Mass transfer coefficient is taken as constant w.r.t height	Maximum deviation of 2% in results of total heat transfer rate and effectiveness

				as column temperature.			from experimental conclusion
Liu et al [82] 2016	Effectiveness-NTU model	Effectiveness, exit air temperature and heat transfer rate.	Model is a modified one that does not depend upon constant column temperature. Developed model is like the counterflow heat exchangers	-	-	-	1.1-1.2 is the range of agreement of model and experimental values
T. Rajaseenivasan et al. [84] 2017	Energy balance	Heat and Effectiveness	Model based on effectiveness and extension of Sharqawy model as used for performance analysis of bubble column Dehumidifier			-Not detailed model -	Not mentioned
P Raj Ranjitha et al.[31] 2018	Thermal Resistance Network	Optimization of dehumidifier sizing	Model is an extension work of Tow analytical model. However, gas side resistances are also neglected here. The modification is done by incorporating the bubble parameters in Tow model such as gas hold up, bubble diameter, interfacial area relation and so on	-Column liquid is adiabatic -Only homogenous flow inside column -Direct condensation is taking place	Complete gas Side resistances are neglected	Chilton-Colbourn relation is used as conservative estimate for mass transfer coefficient.	8% in agreement with Li et al results

Hence in view of above modeling work carried out by researcher, the Figure 4-1 represent the distinct features of the proposed model and description of each feature is presented in section 4.3.

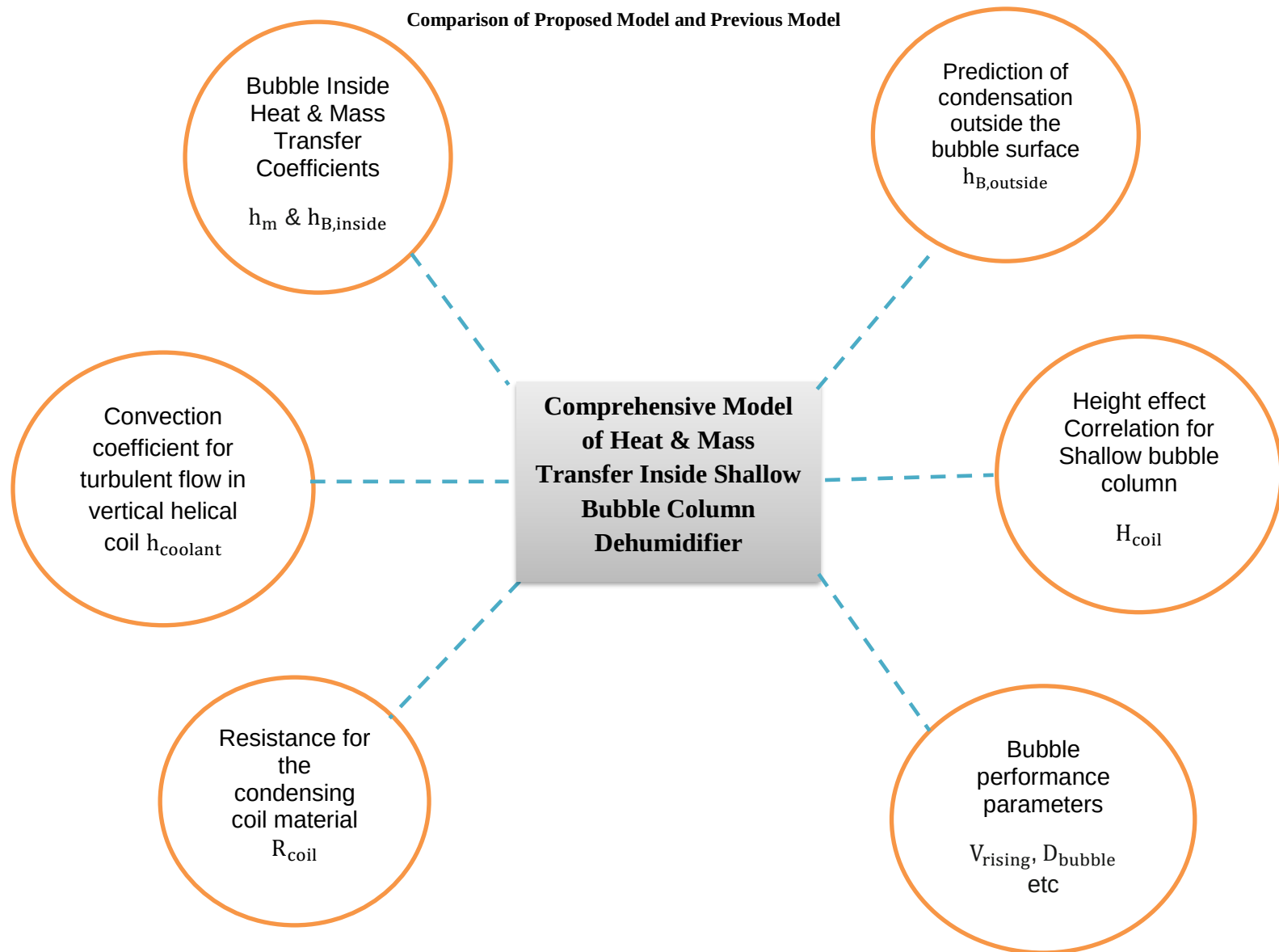


Figure 4-1 Representation of Features of Proposed Model

4.3 Distinct Characteristics of Presented Model

4.3.1 Heat and Mass transfer Mechanisms Inside the Bubble

Naryan et al. [59] performed an analytical modeling of a dehumidifier considering heat and mass transfer inside the bubble of moist gas. But the model relies on several unconfirmed assumptions. They estimated heat transfer coefficient based on the Lewis factor mass transfer analogy. In addition, mass transfer inside bubble is modelled by simple steady diffusion through a slab of which thickness is taken as the bubble radius (upper limit). Then it would be obvious to predict that heat flux can be improved by reduction of bubble diameter. They also predicted a non-zero mass transfer resistance to unphysical diffusion of condensed liquid to column. Similarly, they did not define the correlation for the use of convection heat transfer coefficient inside as well as outside the coolant coil.

Tow et al. and other studies [47], [50] have not considered the mass transfer coefficient, which must be there because of condensation of vapor. Tow [58] neglected the gas side resistances based on the discussion and conclusion drawn from the literature of bubble column reactor. However study of long bubble column in relation with the HDH desalination reveals [85][69][86] [87][88] that operating conditions for the bubble column reactors differ from columns used in desalination. Even the geometrical aspect ratio is smaller for the column and there is no presence of solid particles inside bubble column dehumidifier, which is usually the case in bubble column reactor. Finally, the study of presented model suggested that all the resistances are not negligible for every condition

specially for the case of shallow bubble column. The resistance inside the bubble is greater in magnitude in comparison with all other resistances.

The present model utilizes the correlation of mass transfer inside the bubble based on numerical simulation [89]. This mass transfer is then related with heat transfer coefficient through Lewis factor. The correlation used is in good agreement with results of [90]

4.3.2 Consideration of Heat Transfer Mechanism Outside the Bubble

As vapor reaches the bubble–liquid interface after the diffusion, it simply combines after condensation without further diffusing into column liquid as it is the same liquid as that of column. Naryan et al. [59] has not considered this thermal resistance while Tow [58] used the assumption of perfect mixing and assumed that the temperature after the condensation of vapor mixture is same as that of column liquid and hence neglected the resistances present between the condensate liquid and the pool liquid column.

Practically, there is a different temperature for outer surface of bubble liquid from column's temperature. Since condensation takes place at the outer surface of the bubble, hence, its temperature can never be equal to that of the column liquid at any instant. For simplicity, Tow [58] has taken all the outstream temperatures such as coolant outlet, air outlet and bubble surface temperature equal to that of column liquid. Present model overcomes this drawback and considers a separate temperature T_B at bubble surface where condensation occurred. Hence it is necessary to incorporate a resistance between T_B and T_{column} with its associated convection coefficient.

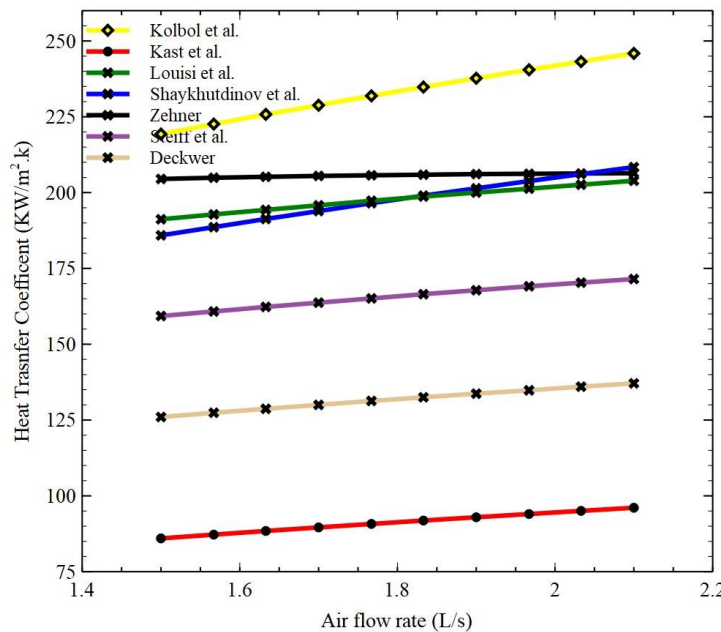
The present model uses the experimentally developed correlation that account for direct condensation of moisture in air at the circumference of the bubble when it is sparged into continuous phase of the liquid at column temperature [91] However we found the original equation as developed in [91] as dimensionally incorrect. So we make analytical substitution and correct the equation which is also given in modified form in [92]

4.3.3 Correlation representing the Heat transfer mechanism for Shallow Bubble Column.

Bubble columns used for dehumidification should be shallow i.e. L/D ratio varies from 2-7.2 as per definition of shallow column [93], [94], to minimize the hydrostatic pressure drop of incoming air and thus reduce the blowing power necessary for blowing air. Shallow bubble columns mostly used in HDH desalination have complicated and different fluid dynamics than tall bubble columns because of the short distance among sparger, free surface and internal heat transfer elements. The bubble column reactors that have been widely studied in literature are taller than column used for desalination. Thus, previous studies focused on design, correlation and modeling of developed region of fluid. Whereas for shallow column, it may be possible that flow is in developing region. The effect of entry region which is very close to bubbling surface is crucial in shallow column and all the effects of flow region are needed to be considered for analyzing the bubble column for HDH desalination.

The effect of sparger on fluid entry region and bubble coalescence on free surface is significant. Its effects on heat transfer in presence of the coil inside the column. Hence heat transfer to tubing in shallow column require further study. Table 3-6 presents the

correlations for heat transfer element immersed in long bubble column, each evaluated at the same conditions. Results indicate that a significant disagreement exists among the applicability of these relations as shown in Figure 4-2. Mostly the existing correlations are semi-theoretical ones that are based on assumed mode of heat transfer mechanism. Some of these correlations rely on the fact that heat transfer occurs by microscale eddies that are generated by the dissipation of bubbles' flow work. While Others consider fluid elements with a different length scale for heat transfer characteristic length , such as the bubble diameter or distance between bubbles. The conflicts that occurred may also be due to differences in measurement methods and, particularly in the case of correlations for internals, geometry etc.



**Figure 4-2 Heat Transfer coefficient for column fluid taken from bubble column Reactor Literature
evaluated at same condition**

Tow et al. [69] conducted an experimental study for a heat transfer coefficient correlation between the coil and liquid in a shallow bubble column. Liquid height, element (HX)

height and its horizontal placement with respect to the perforated plate are varied to analyze the effect of liquid height, column region, flow regime and bubble impact on the heat transfer coefficient. Among the mentioned variables, Heat transfer element height is found to have the most significant effect, which varies significantly upto certain height called critical height. Below the critical height, the effect of sparger must be considered. Above a certain depth, the liquid is found to be in bulk motion termed as swirling or sloshing while below this depth, the effect of the sparger plate prevents bulk motion and thus only strong splashing is detected. Keeping all of these, the present model used the modified correlation to incorporate all the effects relating to shallow columns.

4.3.4 Additional Significance of Model.

In addition to all the above, the present model also considers the conduction resistance in heat transfer path by coolant coil material. This may provide a closer picture to real time scenarios. Similarly, most of the experimental study as discussed below used turbulent flow. This model captures the turbulent flow in vertical helical coil. Noteworthy, this model also captures some aspects of bubble parameters like bubble diameter, bubble rising velocity and nature of bubble flow through bubble Reynold number (Re_{bubble}). Figure 4-3 represent the temperature profile inside the bubble column Dehumidifier and Figure 4-4 represent the complete thermal circuit that is used for present modeling.

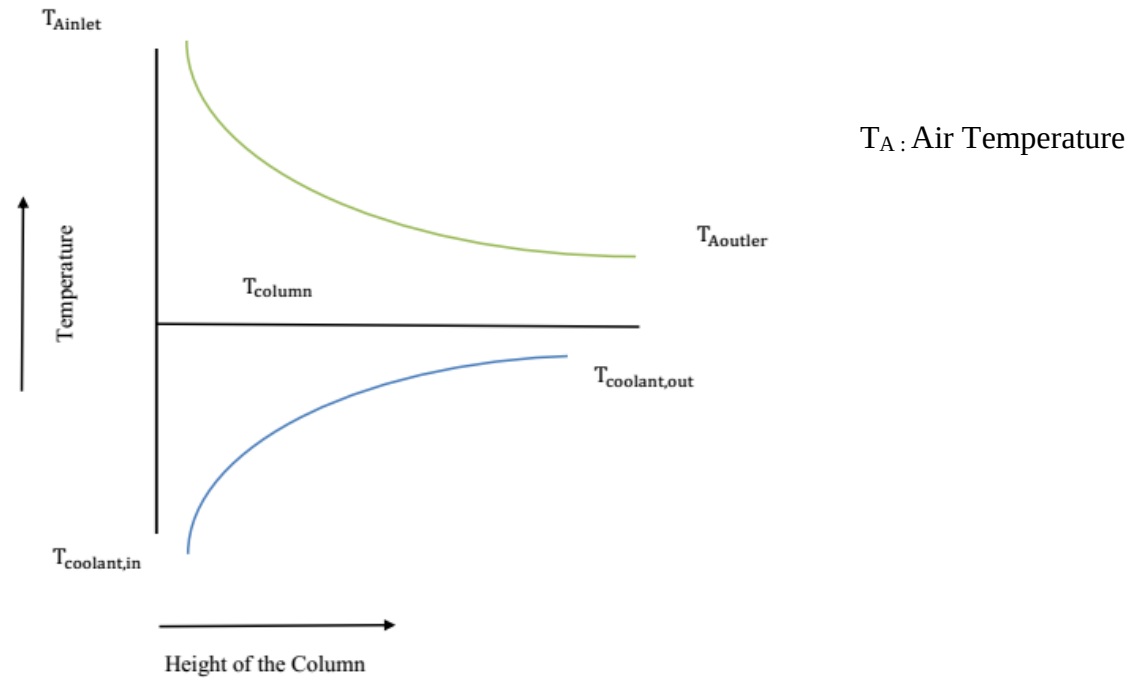


Figure 4-3 Temperature Profile for Stream inside Bubble Column With respect to Liquid Height

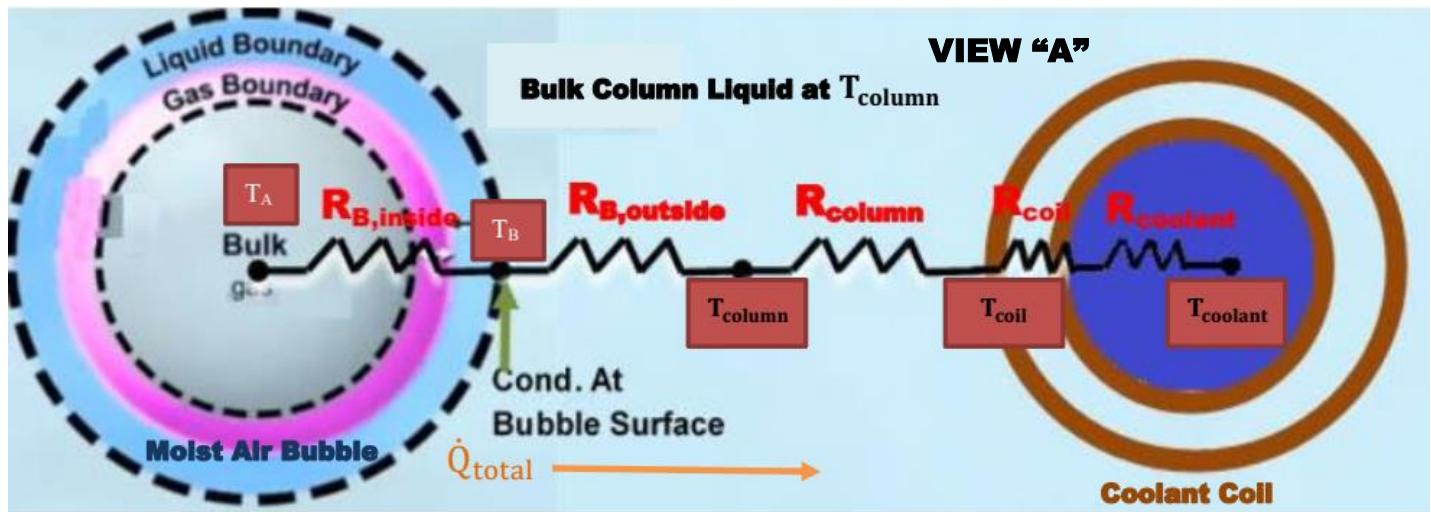
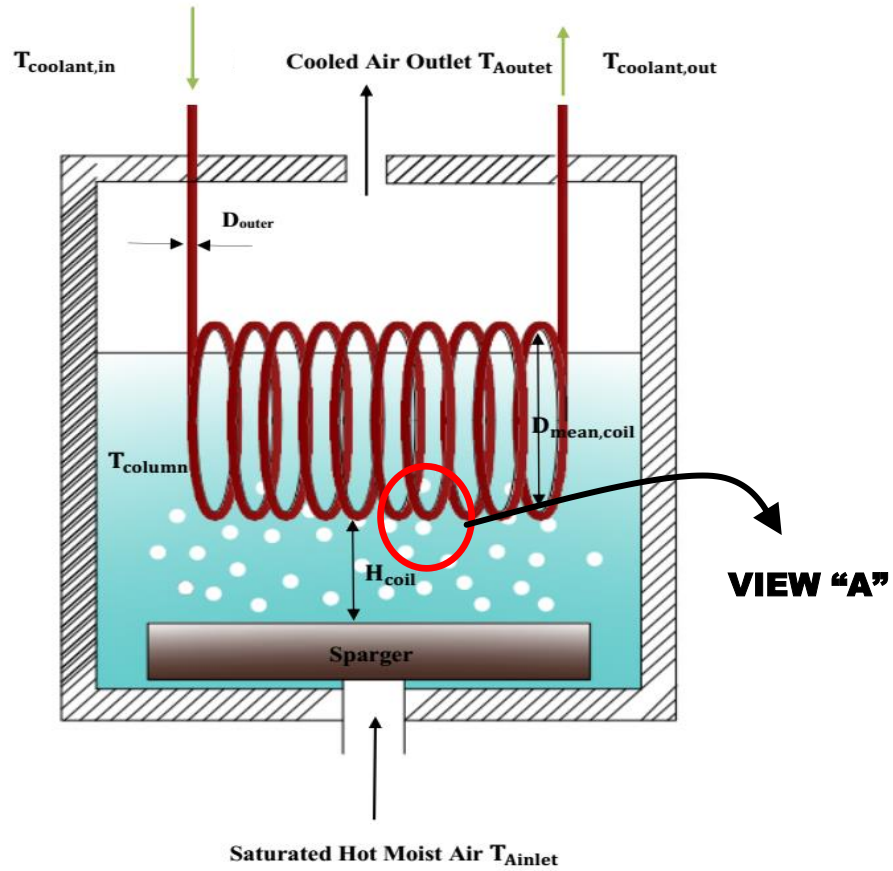


Figure 4-4 Complete Heat & Mass Transfer Resistance Network Inside the Shallow Bubble Column Dehumidifier Used by Proposed Model

4.4 Mathematical Modeling

This section presents detailed analytical model for heat and mass transfer mechanism inside a Shallow bubble column Dehumidifier. This model can easily be described based on thermal Resistance Network for the transfer of heat and mass from the incoming moist air to inside the coolant coil immersed in a liquid pool. Figure 4-4 represents the complete thermal resistance network in the bubble column for the total heat transfer from moist bubble to the liquid column including the mass transfer that occur due to condensation at bubble surface

Air entering to the dehumidifier is assumed to be completely saturated with water vapor at incoming air temperature. Moist Air is shaped into spherical bubbles once after passing through sparger due to surface tension property of the liquid present inside the column. The bulk temperature inside the air bubble is represented by T_A . The vapor inside the bubble condenses at the bubble surface. The bubble surface's temperature is slightly lower than T_A it is represented by T_B . Two resistances are considered inside the bubble with this condensation from T_A to T_B . One is the heat transfer resistance represented by $R_{B,inside}$, inside the bubble mainly due to sensible heat transfer because of temperature differences. While other is the mass transfer resistance $R_{m,Inside}$ due to concentration difference i.e. due to condensation. These two resistances are termed as gas side resistances or resistances present inside the bubbles for heat and mass transfer. After condensation of vapor at bubble surface, the liquid side resistance is considered, which is between the liquid pool of temperature T_C with immediate outer bubble surface. It is termed as $R_{B,outside}$.

Similarly, the convection heat transfer resistance present between the outer surface of cooling coil and liquid pool is liquid side heat resistance denoted by R_{column} . The conductive resistance R_{coil} , due to coil material is also considered in this model in order to closely predict the performance and comparison of system with actual conditions. Finally, the coolant that is circulating inside the cooling coil has forced convection resistance with coil inner surface, termed as $R_{coolant}$. Table 4-2 summarizes the resistances and their associated node temperatures.

Table 4-2 Resistances with their node temperature for Heat Transfer in Bubble Column Dehumidifier

Resistance	Nodes Temperature	Type
$R_{B,inside}$	<i>Heat Transfer from T_A to T_B</i>	<i>Bubble inside Resistance</i>
$R_{B,outside}$	<i>Heat Transfer from T_B to T_C</i>	<i>Resistance for Direct contact condensation</i>
R_{column}	<i>Heat Transfer from T_C to T_{coil}</i>	<i>Convection Resistance</i>
R_{coil}	<i>Heat transfer with coil material</i>	<i>Conduction Resistance</i>
$R_{coolant}$	<i>Heat Transfer from T_{coil} to $T_{coolant}$</i>	<i>Forced Convection Resistance</i>

Moreover, few authors [59] have considered another resistance for case of bubble emerging from sparger directly hitting on coil surfaces for direct transfer of heat from moist air to coil is termed as impact resistance $R_{contact}$. They experimentally studied the effect of bubbles hitting the coil using serpentine coil and concluded that the heat flux almost increases to double with bubbles impact compared with the case of without bubble impact. However, It is important to realize that Coil Surface area has great effect on heat flux particularly for long coils [50] so it must remain constant while analyzing the bubble impact parameter. Similarly coil shape also affects the heat flux significantly because of internal heat transfer coefficient. The turn radius and shape induced secondary flow inside

coils and this effect is significant for small curved pipes [95]. These conditions were overlooked by Naryan et al. [59] as they studied the bubble with two different coil shapes and sizes i.e. circular coil and serpentine coil. The study of Tow [58] was conducted with different Sparger sizes for coil impact without alteration of coil shapes and size. They found that the effect of impact of bubble is not significant. Hence present analysis has not considered it in modeling the bubble column dehumidifier.

4.5 Assumption in Modeling

Following are the assumption that have been employed in modeling

- 1- No heat transfer is taking place above the liquid height inside the bubble column.
- 2- System is well insulated, there is no heat loss to the surrounding.
- 3- No bubbles emerging from the sparger directly hitting the coolant coil.
- 4- Air is in saturation state with water vapors at the inlet and outlet at respective temperatures.
- 5- Coil material has conduction thermal resistance due to its thickness.

4.6 Modeling Equations

Energy effectiveness of a single stage Dehumidifier that is equally applicable to all heat and mass exchange devices is given by [96]

$$\epsilon = \frac{\dot{Q}_{\text{total}}}{\dot{Q}_{\text{max}}} \quad (4.1)$$

Maximum Heat Transfer $\dot{Q}_{\text{max}}$ is obtained if the coolant circulating inside the coil as ideally achieved the Liquid pool temperature or Column Temperature T_C

$$\dot{Q}_{\text{max}} = \dot{m}_{\text{coolant}} C_p (T_{\text{column}} - T_{\text{coolant,in}}) \quad (4.2)$$

$\dot{Q}_{\text{total}}$ is the heat transfer from the moist air stream to the coolant. If there is no loss to surrounding, then total heat transfer is the sum of latent heat and sensible heat transfer from the moist stream of air through the liquid column.

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{latent}} + \dot{Q}_{\text{sensible}} \quad (4.3)$$

The condensation occurred at the bubble surface at temperature T_B . The heat released due to condensation is given by following equation

$$\dot{Q}_{\text{latent}} = \dot{m}_{\text{condensate}} h_{fg} \quad (4.4)$$

Where h_{fg} is the latent heat of vaporization evaluated for steam_{iawps} at temperature T_B . The amount of condensate $\dot{m}_{\text{condensate}}$ is given by

$$\dot{m}_{\text{condensate}} = \dot{m}_{w,\text{inlet}} - \dot{m}_{w,\text{outlet}} \quad (4.5)$$

Where $\dot{m}_{w,\text{inlet}}$ and $\dot{m}_{w,\text{outlet}}$ are the mass flow rates of water vapors carried by moist air at inlet of dehumidifier and at outlet of dehumidifier respectively.

Usually the flow rate of moist air ($\dot{v}_{\text{moist air}}$) at the inlet of dehumidifier is known through mass flow rate of moist air, $\dot{m}_{\text{moist air}}$ can be calculated that leads to inlet mass fraction $Y_{W,\text{inlet}}$ and mole fraction of water vapor, $X_{W,\text{inlet}}$ at inlet conditions.

$$\dot{m}_{\text{moist air}} = \dot{v}_{\text{moist air}} \rho_{\text{moist air}} \quad (4.6)$$

Mole fraction of water vapors at inlet $X_{W,\text{inlet}}$, can be calculated by following simple relation

$$P_{W,\text{inlet}} = P_{\text{total}} X_{W,\text{inlet}} \quad (4.7)$$

Where P_{total} is the ambient pressure because it is assumed that incoming air is at ambient pressure and $P_{W,\text{inlet}}$ is the vapor pressure of water vapor at the inlet of dehumidifier. The known condition of relative humidity of air at inlet (100%) is used to solve the unknown

$P_{W,inlet}$ through the following relation of Relative humidity in terms of vapor pressure of water and saturation pressure.

$$\phi_{inlet} = \frac{P_{W,inlet}}{P_{W,Sat}} \quad (4.8)$$

$P_{W,Sat}$ is saturation pressure of water vapors at $T_{A,inlet}$

Similarly, mass fraction of water vapor $Y_{W,inlet}$ at inlet is given by

$$Y_{W,inlet} = \frac{\dot{m}_{w,inlet}}{\dot{m}_{moist\ air}} \quad (4.9)$$

The $Y_{W,inlet}$ can also be expressed as

$$Y_{W,inlet} = \frac{M_w X_{W,inlet}}{M_w X_{W,inlet} + (1 - X_{W,inlet}) M_a} \quad (4.10)$$

In a similar manner, the outlet state of air from the dehumidifier with assumption of 100% relative humidity may be described by the following relations.

$$X_{W,outlet} = \frac{P_{W,outlet}}{P_{ambient}} \quad (4.11)$$

$P_{W,outlet}$ is taken as saturation pressure of water vapor at $T_{A,outlet}$

$$X_{W,outlet} = \frac{\frac{\dot{m}_{w,outlet}}{M_w}}{\frac{\dot{m}_{da}}{M_a} + \frac{\dot{m}_{w,outlet}}{M_w}} \quad (4.12)$$

Dry mass of air $\dot{m}_{da}$ is given as

$$\dot{m}_{da} = (1 - Y_{W,inlet}) \dot{m}_{moist\ air} \quad (4.13)$$

Now, the sensible heat transfer from the moist air stream is given by following equation

$$\dot{Q}_{sensible} = \dot{m}_{da} \Delta h_{air} + \dot{m}_{w,outlet} \Delta h_{water1} + \dot{m}_{condensate} \Delta h_{water2} \quad (4.14)$$

Where Δh_{air} and Δh_{water1} are the difference of dry air enthalpies and water enthalpies based on $T_{A,inlet}$ and $T_{A,outlet}$. While Δh_{water2} based on $T_{A,inlet}$ and T_B .

Next, Log mean temperature difference is used as a driving force for heat transfer between the coil and bulk temperature of liquid column l

$$\dot{Q}_{total} = \frac{LMTD}{R_{coolant} + R_{column} + R_{coil}} \quad (4.15)$$

and

$$LMTD = \frac{(T_{\text{column}} - T_{\text{coolant,out}}) - (T_{\text{column}} - T_{\text{coolant,in}})}{\ln\left(\frac{(T_{\text{column}} - T_{\text{coolant,out}})}{(T_{\text{column}} - T_{\text{coolant,in}})}\right)} \quad (4.16)$$

Now the above-mentioned thermal resistances and their associated heat transfer coefficients are discussed

$$R_{\text{coolant}} = \frac{1}{h_{\text{coolant}}A_{\text{inner}}} \quad R_{\text{column}} = \frac{1}{h_{\text{column}}A_{\text{outer}}} \quad R_{\text{coil}} = \frac{\ln\left(\frac{r_{\text{outer}}}{r_{\text{inner}}}\right)}{2\pi L_{\text{coil}}K_{\text{coil}}} \quad (4.17)$$

Where A_{inner} and A_{outer} are the areas of coil for convection heat transfer based on its inner and outer diameters and total Length respectively.

It is noteworthy to mention that in practical applications, the coolant used inside the dehumidifier cooling coils is generally seawater for energy conservation i.e. preheating of sea water for its use in the humidifier etc. Moreover, the literature review depicts that experimental studies carried out on dehumidifier bubble column usually use coolants with high flow rates. The high mass flow rates lead to high velocity of coolant and thus turbulent flow is observed inside the curved coil cooling of bubble column. For instance, Liu et al. [82] carried out an experimental study on 9.5mm OD coil with coolant flow rate of 8L/min that corresponds to Reynolds number of 8466. Similarly, Sharqawy et al. [47] used the coolant with same high Reynolds number that depicts turbulent flow inside the coil. Moreover, recently Abdelkader et al. [119] in their study used 1.2cm OD copper coil with mass flow rate of 15L/min that provides Reynolds number of 12,598 i.e. turbulent flow.

The Critical Reynolds number in case of curved coils that distinguishes laminar and turbulent flow is given by [97]

$$Re_{critical} = 2100 \left(1 + 12 \left(\frac{D_{inner}}{D_{mean,coil}} \right)^{0.5} \right) \quad (4.18)$$

Therefore, keeping this in mind, a correlation is needed for the prediction of the convection heat transfer inside the coil with turbulent flow. In this regard, detailed studies have been carried out for the correlation that can represent the utmost picture of coolant flow inside vertical helical coils. After investigation of notable work of turbulent correlation in curved pipes in literature [98]–[100] we found the following correlation to predict convection heat transfer inside the cooling coil [112]

$$h_{coolant} = \frac{Nu_{coolant} K_{coolant}}{D_{inner}} \quad Nu_{coolant} = 0.0456 Re_{coolant}^{0.8} Pr_{coolant}^{0.4} \left(\frac{D_{inner}}{D_{mean,coil}} \right)^{0.21} \quad (4.19)$$

All thermophysical properties of coolant used are evaluated on mean temperature of coolant inlet and outlet from coil.

As discussed in detail in section 5.3.3 , the following correlation has been used. It showed the dependence of coil placement height from the sparger denoted by H_{coil} . This correlation is well suited for shallow bubble columns

$$h_{column} = C_H K_{column,Liq}^{0.5} \rho_{column,Liq}^{0.75} g^{0.25} V_{superficial}^{0.25} \mu_{column,liq}^{-0.25} C_{p,column,liq}^{0.5} \quad (4.20)$$

The properties of column liquid are evaluated at T_{column} . Superficial velocity $V_{superficial}$, a hypothetical velocity calculated from the volume flow rate of moist air $\dot{V}_{moist\ air}$

$$V_{\text{superfical}} = \frac{\dot{v}_{\text{moist air}}}{A_{\text{sparger}}} \quad A_{\text{sparger}} = n_{\text{holes}} \frac{\pi}{4} d_{\text{holes}}^2 \quad (4.21)$$

Where number of holes and their size are known values.

h_{column} correlation is derived from the theoretical background of Deckwers Correlation [75], [101], which is based on the Kolmogorov length scale η as the characteristic dimension for the heat transfer from liquid to the coil instead of coil diameter or bubble diameter used by others. C_H the correction factor for the dependency of coil height is related to the Kolmogorov length scale as follow [81]

$$C_H = 0.02 + 0.18 \left(1 - \exp \left(- \frac{H_{\text{coil}}}{300\eta} \right) \right) \quad (4.22)$$

$$\eta = \left(\frac{\nu^3}{V_{\text{superfical}} g} \right)^{0.25} \quad (4.23)$$

For steady state condition the heat transfer from moist bubble to the column liquid from T_A to T_{column} is the same as that of Equation (5.15) with the following associated Resistances.

$$R_{\text{inside,bubble}} = \frac{1}{h_{B,\text{inside}} A_{\text{outer}}} \quad R_{\text{outside,bubble}} = \frac{1}{h_{B,\text{outside}} A_{\text{outer}}} \quad (4.24)$$

The gas side resistances inside the bubble are important to analyze for the heat and mass transfer from bubble to liquid column . In this regard, a critical review have been made. Only few studies are there only with references to bubble inside resistance specially for a well-developed correlation for prediction of heat transfer and mass transfer. This complex mechanism due to bubbles motions in a pool of liquid along with bubble growth makes elusive to analyze analytically. Colombet et. al [102] performed direct numerical simulation for transport mechanism inside the gas bubble rising through stationary liquid.

The non-dimensional form of Sherwood Number with characteristic length of bubble diameter D_{bubble} is given by

$$\text{Sh} = \frac{h_m D_{\text{bubble}}}{D_c} \quad (4.25)$$

D_c is the diffusion coefficient in m^2/s

Colombet et. al [102] have developed correlation through numerical solutions and provided a normalized asymptotic Sherwood Number Sh_∞^* given by

$$\text{Sh}_\infty^* = \frac{1}{1 + \exp(-\alpha(\ln(\text{Pe}_{\text{max}}) - \beta))} \quad (4.26)$$

Whereas

$$\text{Sh}_\infty^* = \frac{\text{Sh}_\infty(\text{Re}, \text{Pe}) - \text{Sh}_\infty(\text{Re}, \text{Pe} \rightarrow 0)}{\text{Sh}_\infty(\text{Re}, \text{Pe} \rightarrow \infty) - \text{Sh}_\infty(\text{Re}, \text{Pe} \rightarrow 0)} \quad (4.27)$$

We used the definitions of $Sh_{\infty}(Re, Pe \rightarrow 0)$ and $Sh_{\infty}(Re, Pe \rightarrow \infty)$ for back transformation of Sherwood numbers from normalized form and presented the following correlation for convective mass transfer coefficient.

$$h_m = \frac{D_{bubble}}{D_c} \left(6.58 + \frac{(17.71(1 + Re_{bubble})^{0.0044} - 6.58)}{1 + \exp(-1.89(\ln(Pe_{max}) - 3.49))} \right) \quad (4.28)$$

Peclet number Pe_{max} is based on maximum velocity at bubble interface U_{max}

$$Pe_{max} = \frac{U_{max} D_{bubble}}{D_c}, \quad Re_{bubble} = \frac{\rho_l V_{rising} D_{bubble}}{\mu_l} \quad (4.29)$$

U_{max} is evaluated from the Bubble rising velocity in a pool of liquid through Bubble Reynold number

$$U_{max} = 0.5 V_{rising} \left(\frac{16 + 3.35 Re_{bubble}^{0.5} + 3 Re_{bubble}}{16 + 3.35 Re_{bubble}^{0.5} + Re_{bubble}} \right) \quad (4.30)$$

The V_{rising} is found in a correlation provided by stokes [21] as

$$V_{rising} = \frac{\left(\frac{D_{bubble}}{2} \right)^2 2g\rho_l}{9\mu_l} \quad (4.31)$$

Heat transfer coefficient inside the bubble can be related to the with above mass transfer coefficient using Lewis factor. Kloppers [103] presented the Lewis factor as the ratio of

heat transfer Stanton St_h number and mass transfer Stanton St_m number and provided a correlation as

$$Le_f = \frac{h_{conv}}{c_p h_m} \quad (4.32)$$

We found it dimensionally unsatisfactory and assumed that it was just substitutional error in Stanton numbers. The correct form of Lewis factor is given as

$$Le_f = \frac{h_{B,inside}}{\rho_l c_{p,air} h_m} \quad (4.33)$$

Lewis factor is related to Lewis number through the following equation [104]

$$Le_f = Le^{\frac{2}{3}} \quad \text{Where } Le = \frac{\alpha}{D_c} \quad (4.34)$$

Generally, [105] for wet cooling towers $Le_f = 0.89$

The evaluation of h_{inside} from equation (4.33) can provide the resistance to heat transfer inside the bubble. Now, for the heat transfer coefficient outside the bubble surface, which cover the resistance from condensation at the bubble surface and the liquid column can be modeled as direct condensation of rising bubbles.

Terasaka et al.[91] developed a correlation for direct condensation of bubbles in gas-liquid phase studied in a reactor during bubble growth. They validated their stated correlation with their experimental results. However, we found it dimensionally unsatisfactory and modified the provided correlation from substitution of basic equation presented in their work. The corrected form of the correlation is given by Equation 4.35 and confirmed by [92]

$$h_{B,Outside} = \rho_v \frac{\Delta H_v}{\Delta T} \sqrt{\frac{2\Delta H_v \Delta T}{\rho_v T_v \left(\frac{1}{\rho_v} - \frac{1}{\rho_l}\right)}} \quad (5.35)$$

Where T_v is taken as air inlet temperature in Kelvin while ρ_l and ρ_v are the densities of air bubbles and column liquid ΔT is the difference between air inlet temperature $T_{A,inlet}$ and column temperature T_{column} . Here the latent heat of vaporization ΔH_v must be evaluated at condensation temperature which is at the bubble surface T_B .

The temperature profile as shown in Figure 5-3 where the liquid column temperature is constant, and the temperature of air decreasing close to the column temperature. The bubble temperature T_B at which condensation occur must be less than air inlet temperature and greater than column and air exit temperature. With an approximation and using temperature profile of moist air inside the dehumidifier, T_B can be related to air inlet $T_{A,inlet}$ and outlet temperature $T_{A,outlet}$ through this equation

$$T_B = \frac{T_{Ainlet} + T_{Aoutet}}{2} \quad (4.36)$$

Two more relations of bubble diameter and surface tension at bubble surface completes the model with respect to number of variables and equations

$$D_{bubble} = \left\{ \frac{6\sigma d_{holes}}{(\rho_l - \rho_{air})g} \right\}^{\frac{1}{3}} \quad (4.37)$$

Where σ is the surface tension of column liquid water but at temperature T_B

Finally, the coil Heat flux can be obtained by a simple relation

$$q = \frac{Q_{\text{total}}}{A_{\text{outer}}} \quad (4.38)$$

To compare the mass transfer coefficient h_m (m/s) and its associated resistance, mass transfer resistance $R_{m,\text{bubble}}$ (s/m) is defined as

$$R_{m,\text{bubble}} = \frac{1}{h_m} \quad (4.39)$$

4.7 Solution Technique

The above 39 equations (except Equations 4.25, 4.26 and 4.27 since their result is represented by equation 4.28) have been solved in Engineering Equation Solver. Due to small values of resistances the model is very sensitive to significant figures, so residual has been increased and set to 1×10^{-7} . Equations are nonlinear due to LMTD and natural log expression. The known parameters in the above model are the inlet temperature of air and coolant with ambient pressure. The flow rate of moist air at inlet of dehumidifier and coolant flow rate are also known along with dimensions of coil and sparger. Similarly, the critical height for placement of coil above the sparger and Column temperature have been provided as inputs. Built-in thermophysical properties in EES have been used treating coolant and column as water while air inlet conditions. The vapor is treated as $\text{Steam}_{\text{IAWPS}}$ and dry air and moist air has been considered from the function provided separately for both in EES.

4.8 Validation of the Model

Proposed Model is a comprehensive analytical model that described the mechanism of heat and mass transfer inside bubble column Dehumidifier. The presented model is compared with experimental results and model of Naryan et al. [59] at the same set of input parameters.

4.8.1 Superficial Velocity v/s Total heat Flux

Parameters from Naryan et al.[59] experiments has been considered as input parameters for validation. Bubble diameter, D_{bubble} is selected as 4mm, with fixed mole fraction of water vapor at inlet of $X_{W,\text{inlet}}$ 21%, the dehumidifier sparger area A_{sparger} is considered as 0.0645m^2 . The mean coil diameter $D_{\text{mean,coil}}$ is taken as 228mm. The coolant $T_{\text{coolant,in}}$ and air inlet temperature T_{Airlet} are taken as 18°C and 50°C respectively. The outer diameter of coil D_{outer} is 6.35mm. The coolant flow rate $\dot{m}_{\text{coolant}}$ is 0.0071 kg/s . The validation result is shown in Figure 4-5.

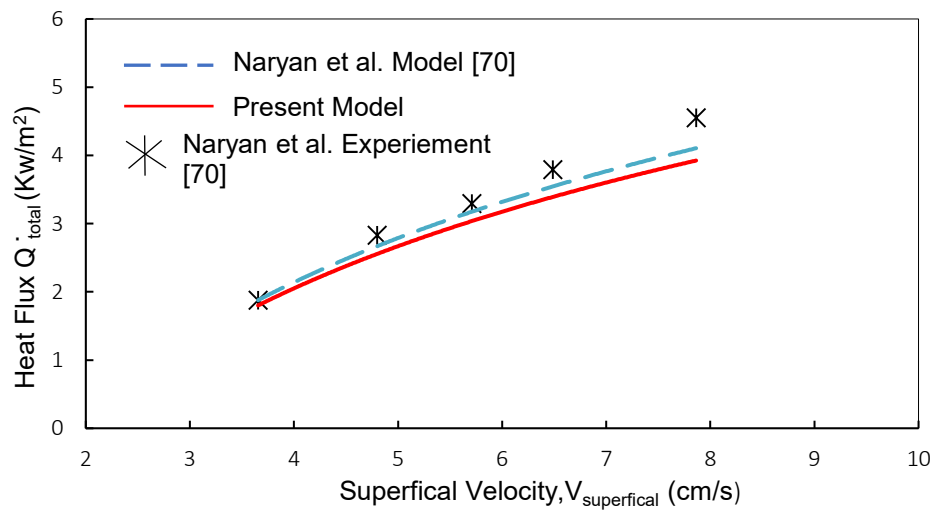


Figure 4-5 Validation Results for Heat Flux & Superficial Velocity (G.P. Naryan model and experiment with Present Model)

An excellent agreement was found between present model and the experimental values. The maximum deviation is of 11% which is for the heat flux obtained at superficial velocity of 8cm/s. There might be an experimental uncertainty in this reading since both model exhibit noticeable deviation at this specific point. Moreover, difference between models have a maximum value of 5%. Figure 4-5 shows that heat flux increases with the increase of superficial velocities. Since heat and mass transfer coefficient are higher for high superficial velocities in bubble column. Moreover, the rate of increment of heat transfer coefficient is more pronounced at low gas velocities and more gradual at high velocities. This is because of change of flow regime as discussed earlier. Flow nature is changed from homogenous bubbly region to churn turbulent regime. The effect of increase of superficial velocity in turbulent regime is reported to be lower in magnitude.

4.8.2 Inlet Mole Fraction v/s Total heat Flux

With the same set of input conditions as discussed in the foregoing validation graph with the superficial velocity. It is now fixed at 3.8cm/s. Naryan et al. [59] model underestimates the heat flux when compared with inlet vapor percentage. However, the present model overcomes this drawback and represents almost same trend with better matching with experimental values. The maximum deviation between the present model and experimental values is 8% while for the case of Naryan the maximum deviation is about 10.2%. The results are presented in Figure 4-6. From Figure 4-6 it is seen that mole fraction has strong effect on heat transfer and it is almost linear in nature. This predict that presence of non-condensable gases in bubble column dehumidifier affecting the heat transfer mechanism to much lesser degree than in film condensation situation where the effect is

quadratic in nature. This graph also explains the superiority of bubble column dehumidifier over the conventional dehumidifiers.

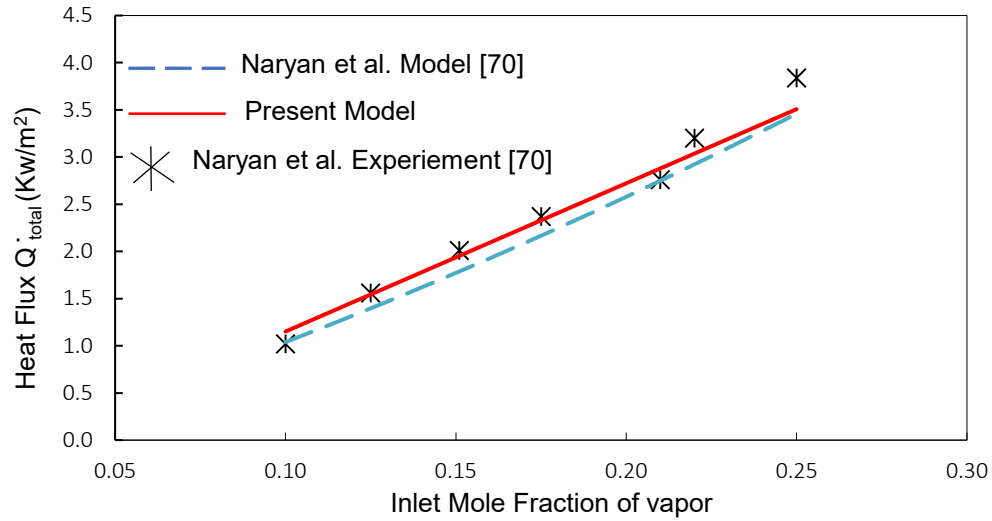


Figure 4-6 Validation Results for Heat Flux & Inlet Mole Fraction of Vapor (G.P. Naryan model & experiment with Present Model)

4.9 Evaluation of Resistances of Thermal Circuit with Operational Parameters of Bubble column Dehumidifier

After Validation, the developed model is solved for those parameters which can be controllable during the experimentation of a bubble column Dehumidifier. In general, incoming air flow rate can be controlled through a valve, air temperature can be controlled by the heater, operating pressure can be controlled using the pressure regulator, height of heat transfer internal coil can be adjusted and the bubble diameter can be obtained as per requirement through sparger size and design.

4.9.1 Effect of Incoming Air Temperature:

All resistances associated with heat transfer from moist bubble to coolant liquid are analyzed with effect of air inlet temperature T_{Ainlet} to dehumidifier. Air inlet temperature is one of the controllable operational parameters in bubble column HDH desalination systems. The air temperature range from 45 °C to 70 °C is considered for this study because this is the usual values of temperatures reported in literature in almost all experimental studies of bubble columns [32], [43], [47], [50], [59], [82], [106]. The parameters that are kept fixed in model are mass flow rate of coolant as 0.42 lpm and inlet air flow rate as 0.15 L/s. and coolant inlet temperature is 18 °C. The length of coil is selected to be 3m which is equally comparable with Naryan and Tow [50]. The other inputs values are number of holes and diameter of perforated plate given as 121 and 1.59mm respectively. The corresponding increase in column or pool liquid temperature T_{column} is obtained as 30 °C to 44 °C.

From Figure 4-7, it is clearly seen that all resistances are decreasing with the moist air temperature except the $R_{\text{outside,bubble}}$. This thermal resistance between the column liquid and condensation ($R_{\text{outside,bubble}}$) is much smaller in terms of magnitude (i.e. 10^{-3}) in comparison with others so that its effect on total heat transfer is almost negligible. Hence it appears that heat transfer inside the dehumidifier is enhanced with the increase of air temperature. This is quite satisfactory and in agreement with the finding of Tow [61] for heat flux against the air temperature.

In term of magnitudes of thermal resistances only, $R_{\text{outside,bubble}}$ is found to be the lowest with value of 1.758×10^{-4} K/kW and $R_{\text{inside,bubble}}$ is the highest as 0.14 K/kW at air temperature of 45 °C. While the mass transfer resistance R_m is greater from all thermal resistances with a value of 7.5 s/m.

Increasing moist air temperature, the thermal resistance between the column liquid and coil, R_{column} decreases slightly. The increase of air inlet temperature leads to higher column or pool liquid temperature T_{column} for a given coolant inlet temperature. As the column temperature increases, the conductivity of water (liquid column) increases with decrease of its thermal capacity and viscosity. These slight changes in properties leads to minor increment of convection heat transfer coefficient h_{column} and thus insignificant reduction in resistance. However, despite of the magnitude, high air temperature is a favorable operational condition for heat transfer through portion of column liquid to coolant coil.

Both resistances of bubble (i.e. inside and outside) vary substantially with air inlet temperature. The sharp increment of $R_{\text{outside,bubble}}$ depicts the corresponding decrease of heat transfer coefficient that relates the condensation. The driving force between the pool liquid and air bubble reduces at high temperature of moist air which decreases the condensation. Moreover, for constant relative humidity of air from humidifier outlet, the dew point temperature for condensation increases with air temperature which also reduces condensation effect. However, this thermal resistance is very low such that its contribution in overall heat transfer from the moist bubble to coolant coil is insignificant and overall heat transfer increases with air temperature.

Contrary to $R_{\text{outside,bubble}}$, thermal resistances inside the bubble $R_{\text{inside,bubble}}$ exhibit a declining trend. It pronounces that latent heat transfer play a major part in total heat transfer than Sensible portion. Mass transfer resistance R_m is significantly affected by air temperature with decreasing trend from 7.5 s/m to 6.5 s/m on 25°C increment. Similarly, the coolant resistance R_{coolant} decreases to a little extent. Due to increase in column temperature, coolant outlet temperature increases which increases the properties as conductivity, density, viscosity on which heat transfer coefficient of coolant depend.

Noteworthy to mention that optimum objective of bubble column dehumidifier is to have high heat transfer rate with minimal condensing coil area. For a small-scale bubble column dehumidifier, these resistance in sum up form may significantly affect the performance for per kW of energy. The considered range of air temperature (45 °C to 70 °C) is generally in water heated cycles study. The temperature at outlet of the humidifier is even lower for air heated cycles. Even at this small range of temperatures, the thermal resistances are high (since all are in per kW) so air temperature after humidifier should be kept high for better performance of dehumidifier.

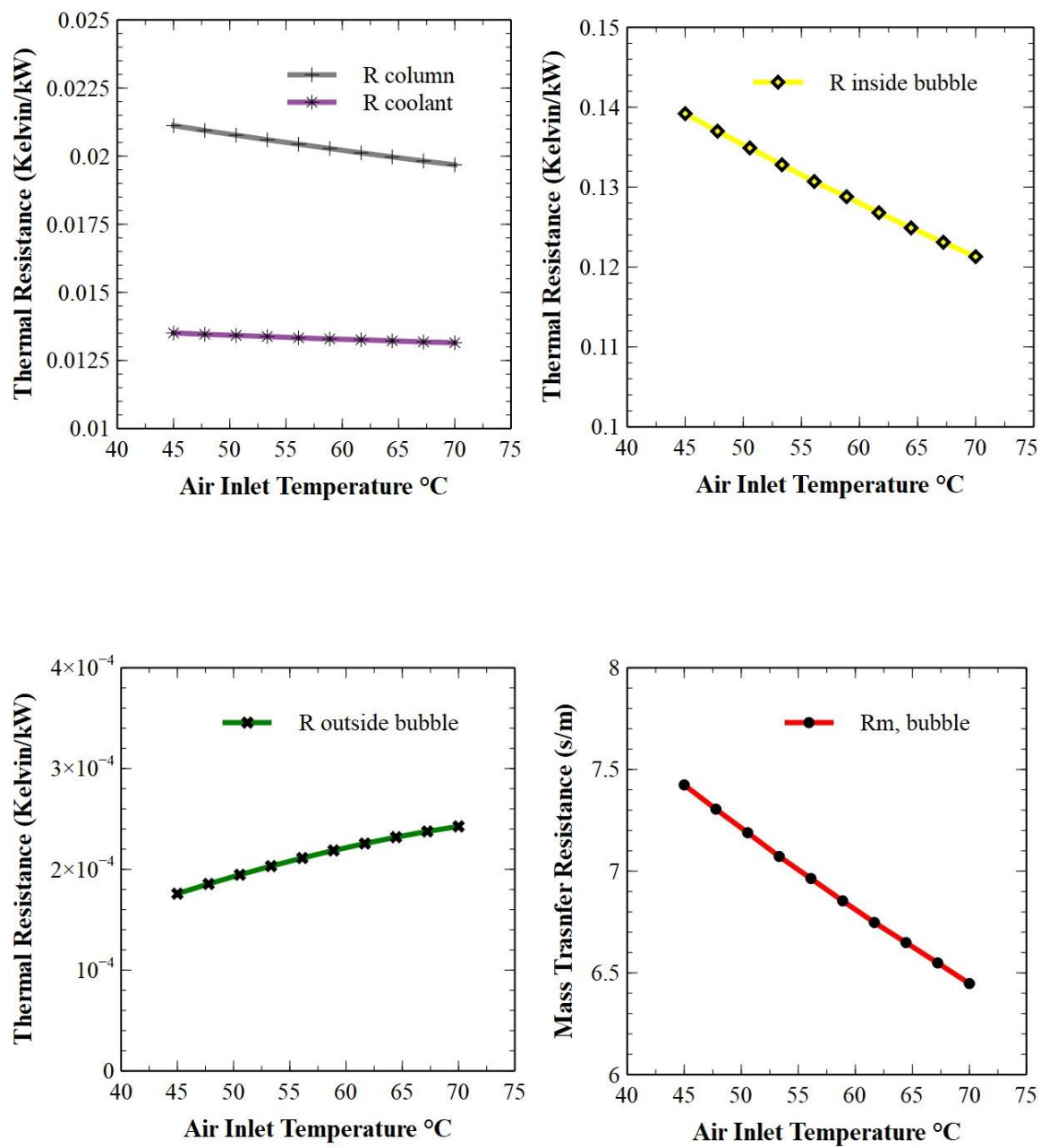


Figure 4-7 Variation of Thermal Resistance & Mass transfer Resistances inside Bubble Column Dehumidifier against Incoming Air Inlet Temperature

4.9.2 Effect of Incoming air flow rate:

Flow rate of moist air is another operational parameter that can affect the performance of the dehumidifier. All resistances have been analyzed for airflow rate with the same inputs of Figure 4-7, but air inlet temperature is set as 50 °C. To make consistency for comparison, the column temperature is set from 30 °C to 39 °C. All thermal resistances show decreasing trend with the flow of incoming air represented in Figure 4-8. The mass transfer resistance is unaffected by the flow rate and so the superficial velocity. R_{coolant} and $R_{\text{inside,bubble}}$ are merely affected by the airflow rate and exhibit a constant line. Whereas the R_{column} decreases rapidly, because the heat transfer from column liquid and coil strongly depend upon the flow regime. Superficial velocity is one of the parameters that determine whether the flow is in homogenous, developing region or in high turbulence region. For superficial velocities greater than 1.5cm/s, high bubble motion is observed [81] this leads to more heat transfer and reduced resistance from column liquid to coil.

Moreover, for all thermal resistances, higher incoming flow rate leads to high superficial velocities of the gas bubbles from the sparger. This provide more gas hold up in the bubble column dehumidifier liquid pool. Increase in the intensity of liquid pool and gas contact and ultimately high interfacial area. This interfacial contact is the reason of effective transport (mass and heat) resulting in lower resistances. High heat transfer coefficient with moist volume flow rate result in higher heat transfer rate.

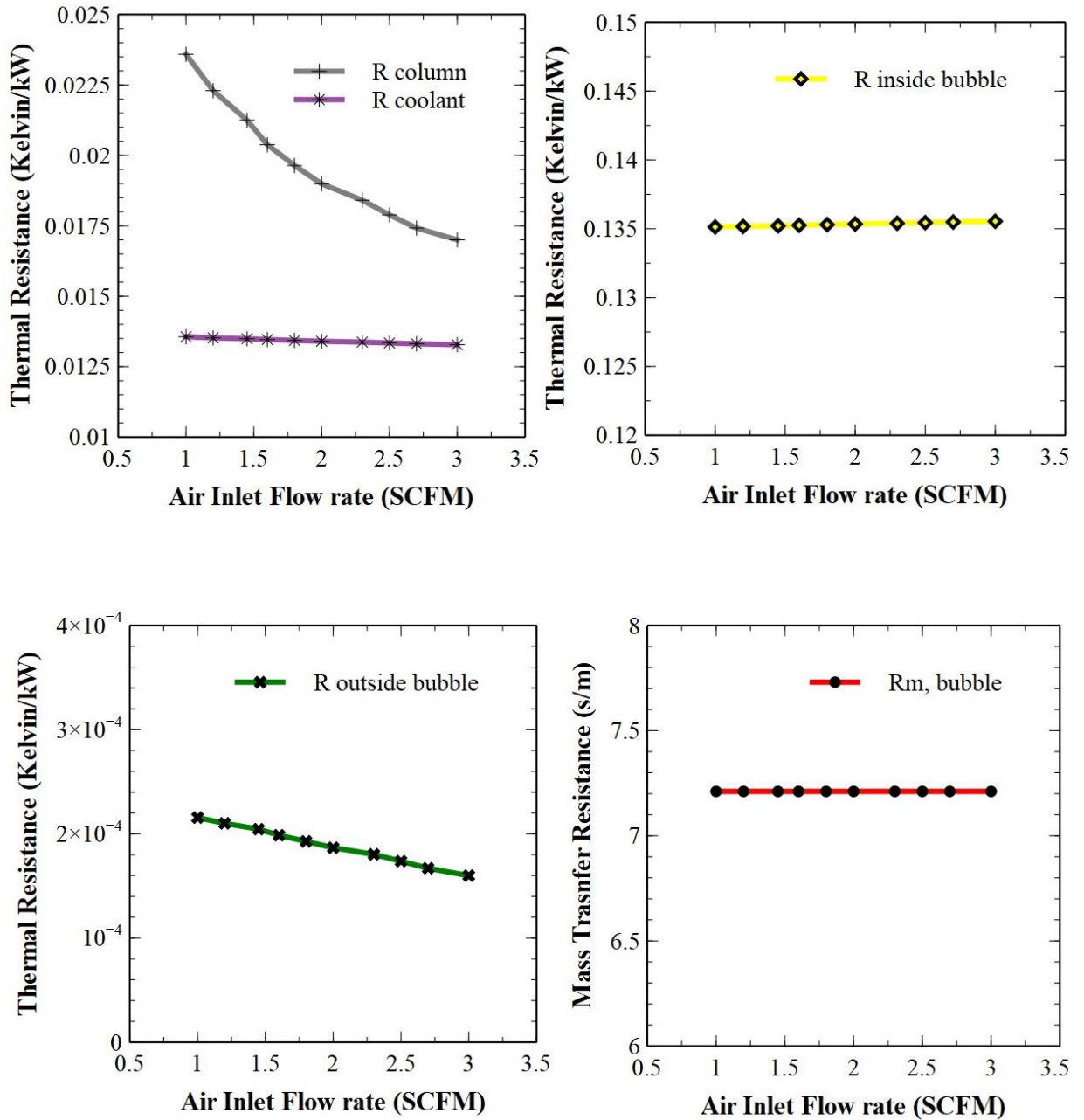


Figure 4-8 Variation of Mass Transfer & Different Heat Transfer Resistance inside Bubble Column Dehumidifier against Incoming Air Inlet flow rate

4.9.3 Effect of Bubble Diameter:

Effect of bubble diameter is investigated on associated resistances with fixed flow rate of air as 1.5 SCFM. Due to sensitivity of model a small range of bubble diameter i.e. from 3.3mm to 4.3mm is chosen. Bubble diameter is found to have significant effect on mass transfer and heat transfer relating to moist bubble depicted in Figure 4-9. One thermal resistance is found to increase with increase of bubble diameter $R_{\text{inside,bubble}}$ and other $R_{\text{outside,bubble}}$ is found to decrease. Thermal Resistance $R_{\text{outside,bubble}}$ is quite low while in comparison the $R_{\text{inside,bubble}}$ is high in magnitude which pronounce that the bubble diameter should be small for better heat transfer. This graph provides an explanation of Naryan et al [69] Figure 7. Greater bubble diameter lowers the interfacial area per unit volume of bubble. Thus, lower heat and mass transfer coefficient between the moist bubble with liquid column for a given flow rate and temperature of air.

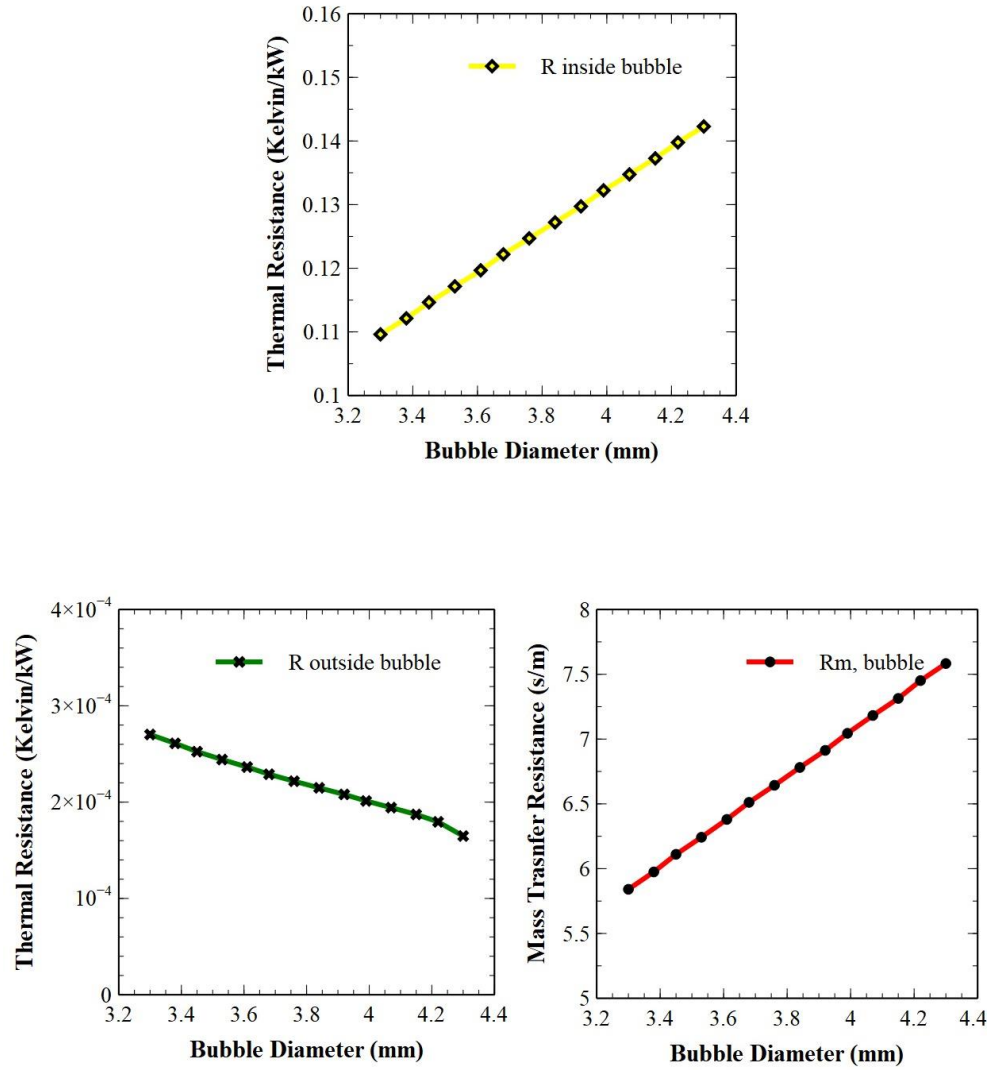


Figure 4-9 Effect of Bubble Diameter on the performance of system evaluated through Resistances

4.9.4 Column Resistance with Height of Coil:

Liquid height inside the column represents the static pressure head that must be overcome by incoming air supply. The greater the liquid height, the higher the power input from the air blower to overcome that head. Several experimental studies have been carried out to find the effect of liquid height on dehumidifier performance. It has been shown that height of liquid has no effect on effectiveness and heat transfer of dehumidifier. Hence it should

be kept minimum for better performance of the system. But at the same time care must be taken for the coolant coil so that it should be fully immersed in liquid. If the coolant coil is not immersed completely in liquid, then the energy of chiller would be wasted. To keep the liquid height lower and to make the coolant coil fully immersed, the coil may be placed near the sparger.

Hence it is critical to determine the heat transfer coefficient with effect of sparger on the coolant coil. Long column that have greater diameter to length ratio are affected by sparger because the flow is fully developed before reaching to the coolant coil. However, this is not the case with shallow bubble columns. Since for lower energy input (leading to better GOR) HDH usually has shallow bubble columns where the sparger may affect the dehumidifier performance. Hence the model is solved for Naryan et al. [69] experimental condition while varying the location of placement of coil. The superficial velocity is set to 6.52 cm/s with bubble diameter 4mm. The inlet mole fraction of water vapor is fixed at 21% with air incoming temperature as 40°C. R_{column} (so the heat transfer coefficient) is investigated by varying H_{coil} from 10mm to 65mm. The result is shown in Figure 4-10. R_{column} varies strongly with the height and becomes constant at 55mm. R_{column} exponentially decreases upto 55mm showing the heat transfer coefficient is enhanced because of the turbulence created by the sparger. After 55mm the effect of sparger is insignificant and that's the reason the heat transfer coefficient and hence the heat transfer and effectiveness become independent of height.

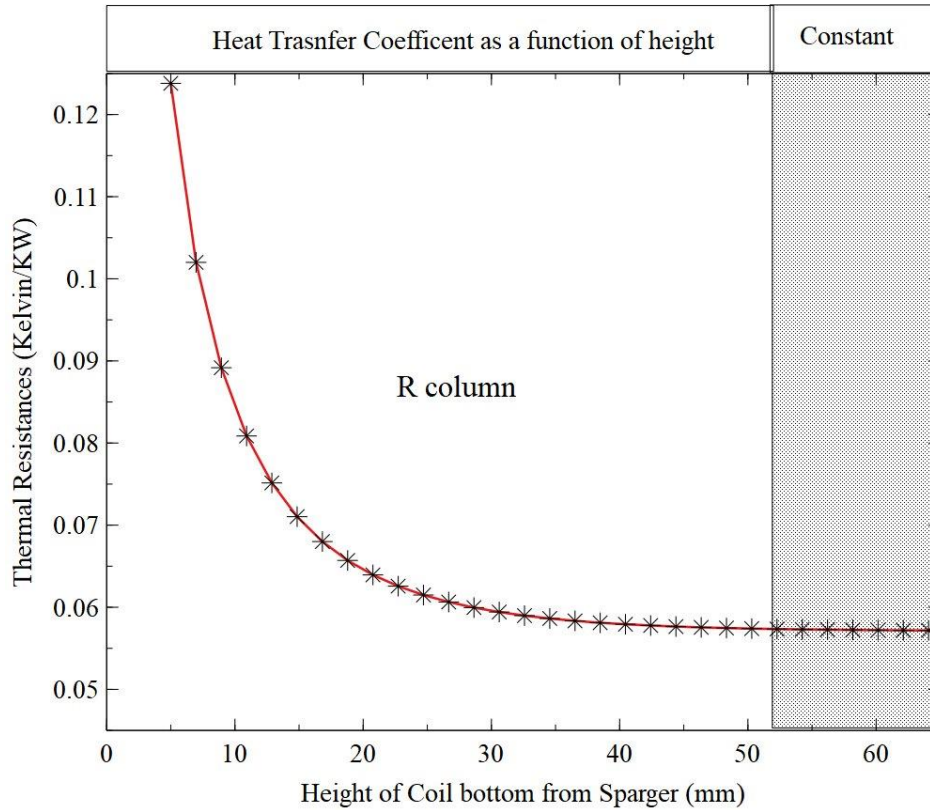


Figure 4-10 Column Liquid Resistance variation with the height taken from sparger to bottom of cooling coil from

This trend is explaining Naryan system where heat transfer becomes independent from 150mm. Here 55mm is considered the critical height. Below the critical height, the splashing and sloshing caused by sparger increases the heat transfer coefficient. Above the critical height where the bubble mostly coalesces and form liquid i.e. a bulk region provides constant heat transfer coefficient leading to a stable resistance from column liquid to coolant coil. These results of model provide analytical explanation of the experimental study conducted by Tow [61].

4.9.5 Effect of operating Pressure in Dehumidifier:

Most of the experimental works carried out on the performance of dehumidifier bubble column is on the atmospheric pressure. There are few works like Sharqawy et. al [43] who investigated the performance of a bubble column dehumidifier with varying operating pressure of air. They experimentally regulated the pressure through a regulator valve installed before the Inlet to the column. They tested the system upto 2bar. They proposed that the dehumidifier to be operated at higher pressure for better performance in view of their experimental results. Hence this proposed model is used to evaluate the detailed performance of bubble column dehumidifier including the heat and mass transfer coefficient, heat transfer (both sensible and Latent) along with the energy effectiveness. The range of pressure investigated is from 1 bar to 8 bar. The model is solved for coolant inlet temperature of 25 °C and the inlet air temperature is set to 58 °C. The mass flow rate of the coolant is set as 4.14 Liter/min and air flow rate at 1.5 SCFM.

Heat transfer coefficient inside the coolant coil & between the coil and column liquid i.e. $h_{coolant}$ & h_{column} are not shown here in Figure 4-11 but they are found to be linearly increasing with the pressure. However, their change is not substantial, and the model reveals that they are strong function of the column temperature T_{column} rather than the operating pressure. The bubble side mass and heat transfer coefficient (h_m and $h_{B,inside}$) are found to be strongly dependent on operating pressure. They both decrease exponentially but the heat transfer mechanism is dominant inside the dehumidifier if we increase the pressure. This can be justified on the basis of magnitudes, the heat transfer coefficient for inside bubble is 100 times comparatively greater than that of the mass transfer coefficient.

In another way, the latent heat transfer $\dot{Q}_{\text{latent}}$, which is purely due to the inside bubble heat transfer coefficient is higher than that of the sensible heat $\dot{Q}_{\text{sensible}}$. The variation of operating pressure controls the sensible part of the total heat transfer.

Heat transfer coefficient $h_{\text{B,Outside}}$ between the outside bubble and column liquid is found to be linearly increasing with operating pressure and it is even greater than all other heat transfer coefficient. This depicts this coefficient is responsible for increase of total heat flux and this is the most dominant among the other. Due to exponential decay of gas side coefficient, latent heat despite of higher magnitude is decreasing with the increase of pressure. The sensible heat because of the dominance of heat transfer rather than mass transfer is increasing. At Around 6.5 bar the sensible and latent heat becomes identical after which latent heat turns to decreases compared with the sensible heat. Therefore, the overall effect of operating pressure results in increase of heat transfer is what closely resembles the case of the experimentally study. [43]

The parallel flow effectiveness is also compared with the system and found to be increasing in Figure 4-11(d). For a typical operating condition and fixed dimension of bubble column, the increment of pressure from atmospheric to even 7 bar can lead the bubble column to achieve high effectiveness around 0.88.

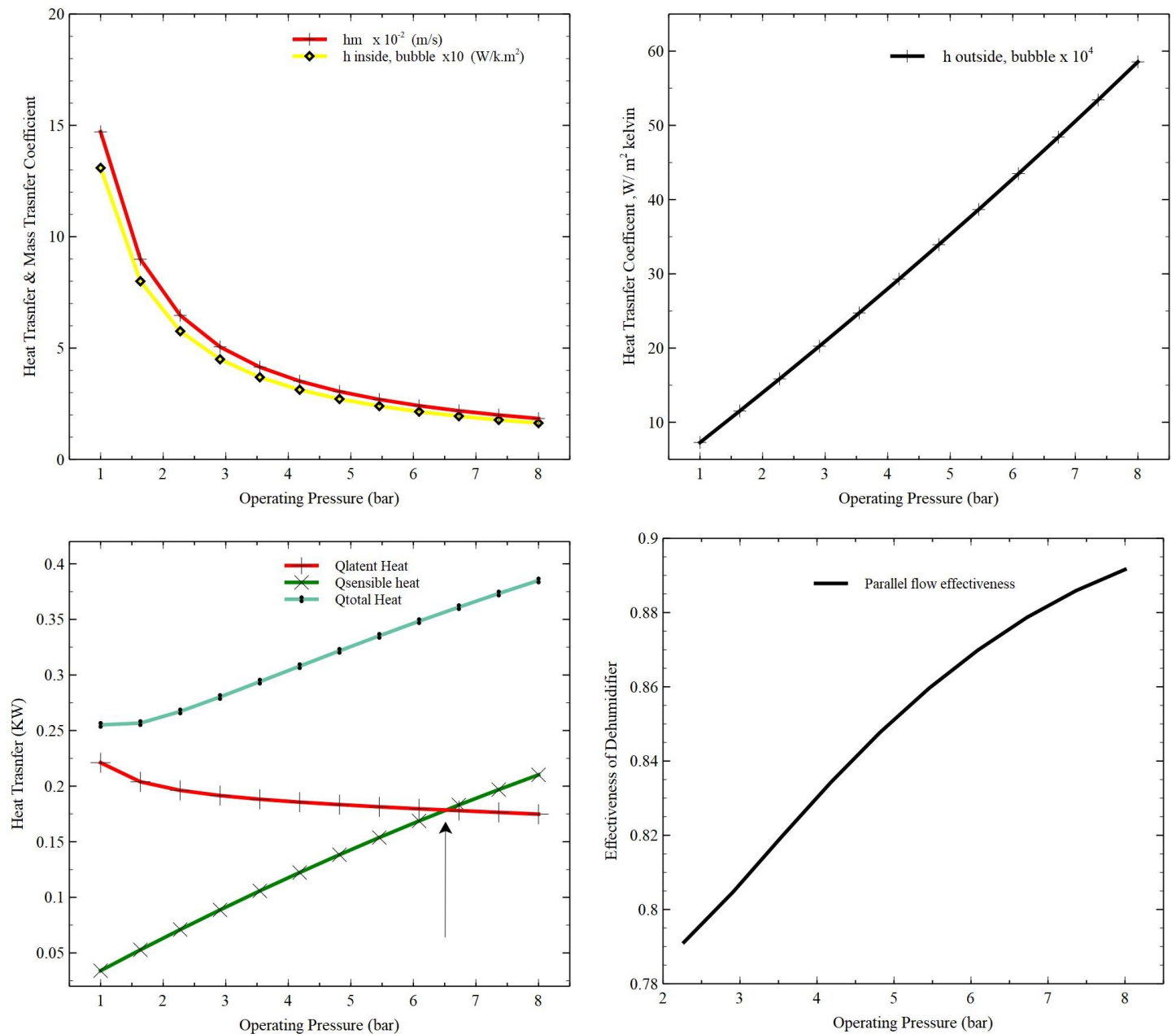


Figure 4-11 (a) Operating Pressure Effect on Coefficient of heat & mass transfer (b) Outside bubble heat transfer Linear Variation with Pressure (c) Effect of Operating Pressure on Total, latent & Sensible Heat transfer (d) Trend of Effectiveness with Operating Pressure

CHAPTER 5

EXPERIMENTAL PERFORMANCE OF BUBBLE COLUMN HUMIDIFIER & DEHUMIDIFIER

This chapter describes the experimental assessment of bubble column humidifier and dehumidifier for air heated and water heated HDH cycles. The experimental study and the comparison were made for both cycles. A parametric study was carried out for evaluating the system performance parameters such as GOR, distillate rate and effectiveness through the variation of inlet temperature, flow rate of air and humidifier water level. Due to low GOR and temperature limitations of air heated cycle, the complete system is then modified to include of double stage bubble column humidification and dehumidification. The flow chart in Figure 5-1 represents the synopsis of experimental study presented in this chapter.

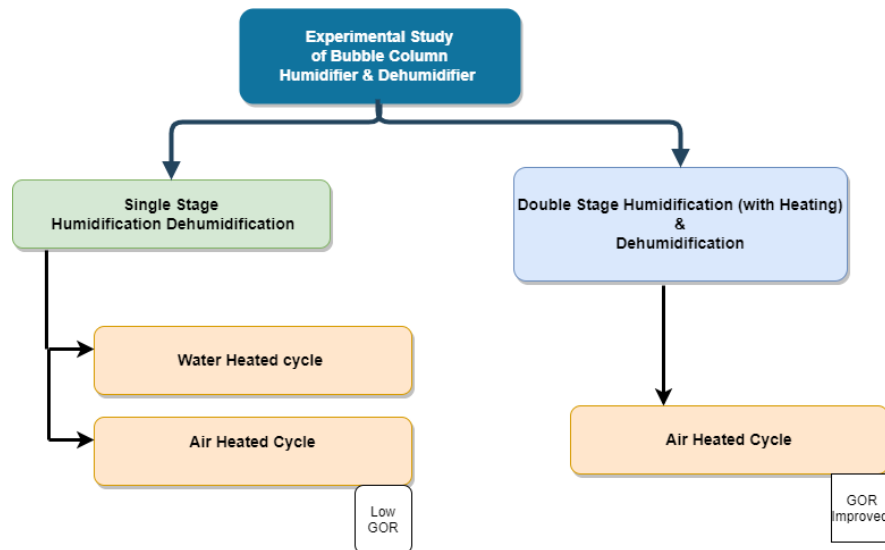


Figure 5-1 Synopsis of Experimental Study

5.1 Experimental Arrangements of Single Stage Humidification Dehumidification

A compact experimental setup is built in the lab to investigate the performance of bubble column Humidification Dehumidification system as single unit. Two columns of acrylic plastic are made in cylindrical form. One column containing copper cooling coil is considered as a Dehumidifier. The other one containing a sparger plate is considered as Humidifier as shown in Figure 5-2. Air from the supply point through a compressor is flowing from an adjustable tap to have different flow rates. Air then passes through a cylindrical heater where it is heated from the ambient temperature to required temperature. While for water heated cycle, the heaters are placed inside the Humidifier to directly heating the water. In both cases air enters the humidifier from the bottom where sparger plates creates the bubbles.

For the air heating cycle, at a height of 1.5cm from the sparger plate there is a perforated plate holding the saline water above it. Heated air passes through water and accordingly is humidified and it raises the temperature of water and leaves the humidifier with a relative humidity that approaches 90%. After Humidification, air enters in to the dehumidifier bubble column at the bottom. It also contains a sparger plate for creating bubbles. In the dehumidifier, the moist bubble condenses when it comes in direct contact with fresh water at lower temperature. The increment of water level inside dehumidifier represents the amount distillate. Both columns are marked with scales in cm which are used for finding the height of liquid inside the columns. The dimensions of both bubble columns and cooling coil are given in Table 5-1

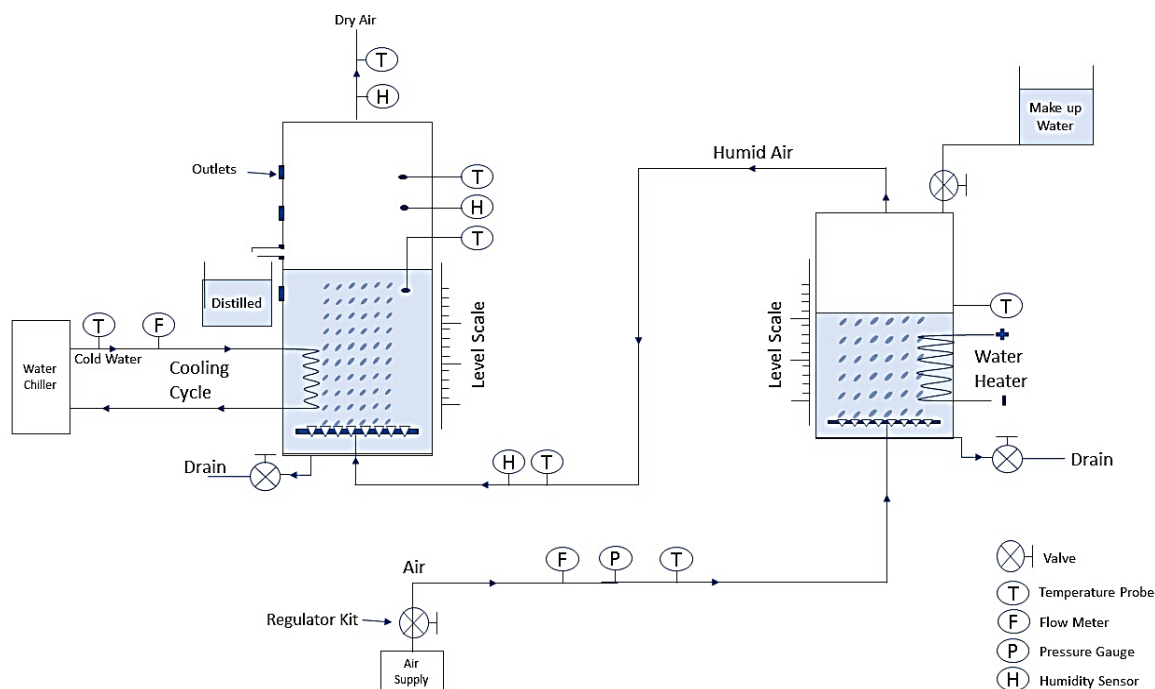


Figure 5-2 Schematic Diagram of Experimental Study of Single Stage Bubble Column HDH System

A-12 Channel Thermocouple Data Logger HH4208 is used for measuring temperature using K-type thermocouples. 8 thermocouples at different locations (Table 5-2) measure the temperature. All thermocouples used are having 0.1°C Resolution and an accuracy of $\pm (0.4\% + 0.5^{\circ}\text{C})$

Table 5-1 Geometric Parameters of Humidifier and Dehumidifier

Equipment	Dimensions		
	Internal Diameter	Outer Diameter	Height
Humidifier	11.5cm	13.5cm	23cm
Dehumidifier	7cm	9cm	23cm
Cooling Coil	-	1.2cm approx.	-

A rotameter is connected after the air supply tap to measure the flow rate. All tubing used for air connection are of 10mm in diameter. They are properly insulated with glass wool to minimize heat loss. The inlet and outlet of the cooling coil are connected to a

chiller ACCEL 500LC through which water as coolant is circulating at constant flow rate of 4.2 Liters/min. The Inlet temperature of the coolant to the dehumidifier is adjustable in order to maintain the constant columns liquid temperature inside the dehumidifier.

Table 5-2 Temperature Measuring position of Digital Thermometer

Thermocouple	Position
T ₁	Air incoming to Humidifier
T ₂	Inside humidifier measuring Saline Water Temperature
T ₃	Air Temperature Leaving Humidifier
T ₄	Temperature of Columns Liquid inside Dehumidifier
T ₅	Inlet Coolant Temperature into cooling coil
T ₆	Outlet Coolant Temperature from cooling coil
T ₇	Air leaving the Dehumidifier

Moreover, to maintain a constant level of water inside the humidifier a make-up connected to ball valve is connected to the top of column to maintain the required water level inside the humidifier before the system is reaches into steady state. The system is operated at atmospheric pressure, which is assumed to be 101.325 kPa.

For the water heated system, hot water temperature of 65°C, 75°C, and 85°C are tested. Furthermore, air flow rate values of 0.15 SCFM (standard ft³/min), 0.25 SCFM, and 0.35 SCFM are used. Three water levels in humidifier are considered, 4.5, 6, and 7.5 cm, respectively. On the other hand water level tested in the dehumidifier is kept constant at 8 cm. Cold water temperature in the dehumidifier varies from 10°C to 22°C. For air heated system, tested air temperature values are 110°C, 120°C and 130°C. Five different flow rates of air are considered from 0.5 SCFM to 1.5 SCFM. Three water levels inside the humidifier are considered; 4cm, 3cm and 2cm. All thermocouples used are of K-type with 0.1°C resolution and $\pm 0.5^\circ\text{C}$ accuracy.

5.1.1 Uncertainty analysis

The uncertainty is evaluated for results that are calculated based on experimental observations. Temperature sensors at different locations, flowmeter are used to measure the operating parameters. Level of water inside bubble column is measured through calibrated scaled rulers attached on outside surface. It is required to measure the effect of these values on the system performance parameters, distillate, effectiveness & GOR.

Standard estimation of Type B uncertainty in measurement is performed through instrument accuracies by following equation [107][108]

$$U = a/\sqrt{3} \quad (5.1)$$

Where U is uncertainty of Type B and a is accuracy of instrument .The uncertainties associated with experimental facilities are shown in below Table 5-3.

Table 5-3 Detailed description of instruments

<i>Property</i>	<i>Instrument</i>	<i>Range</i>	<i>Accuracy</i>	<i>Standard uncertainty</i>
Temperature	Digital Thermocouple RDXL 12SD, K-type	-100°C to 1300 °C	±0.5°C	0.2886°C
Air Flow rate	FL50000 Omega Flow meters	0.5 to 4.5 SCFM	(±5% FS)	0.1299 SCFM
		&	±0.225SCFM &	&
		0.1 to 1.2 SCFM	±0.05 SCFM	0.0346 SCFM
Relative Humidity	RH318 Hygro thermometer	0 to 100 % RH	±2.0% RH	1.1547 % RH
Height	Measurement Scale	0 to 10 cm	± 1 mm	0.577 mm
Time	Stop watch	0 to 1 hour	±0.1s	0.0577 sec

Moreover, engineering equation solver (EES) helps to determine the indirect uncertainty on the calculated values. For air heated cycle flow meter of range 0.5-4.5 SCFM was used while for case of water heated cycle it was 0.1 to 1.2 SCFM. The results of indirect uncertainty on calculated variables are shown in tabulated form as follow in Table 5-4.

Table 5-4 Indirect Uncertainty on Calculated Results

<i>Parameter</i>	<i>Indirect Uncertainty</i>							
	<i>Air Heated cycle</i>					<i>Water Heated Cycle</i>		
	0.5 SCFM	0.75 SCFM	1.0 SCFM	1.25 SCFM	1.5 SCFM	0.15 SCFM	0.25 SCFM	0.35 SCFM
GOR	±0.0009	±0.0008	±0.0008	±0.0009	±0.001	±0.07459	±0.06067	±0.006387
Distillate	±0.004351	±0.00400	±0.004512	±0.004611	±0.00501	-		
Humidifier Effectiveness	±0.00525	±0.00561	±0.0050	±0.00512	±0.00512	±0.001252	±0.01274	± 0.01277
Dehumidifier Effectiveness	± 0.012	± 0.011	± 0.012	± 0.016	± 0.012	± 0.01161	± 0.0182	± 0.01204
Total heat Input	±0.01202	±0.0120	±0.0110	±0.0121	±0.0122	±0.07425	±0.07425	±0.07425

5.1.2 Performance Parameters

Distillate amount (m_d)

$$m_d = \frac{\pi}{4} D^2 h \quad (5.2)$$

Where h is the height increment for each observation inside the dehumidifier.

Total Energy input (Q_{in})

$$Q_{in} = Q_{heating} + Q_{cooling} \quad (5.3)$$

$$Q_{\text{heating}} = m_{\text{air}} C_{\text{pair}} (T_1 - T_{\text{ambient}}) \quad (5.4)$$

$$Q_{\text{cooling}} = m_{\text{chiller}} C_{\text{pwater}} (T_6 - T_5) \quad (5.5)$$

T_1 air temperature leaving the heater while T_6 and T_5 are coolant outlet and inlet temperature from dehumidifier cooling coils. respectively

Gain Output Ratio (GOR)

$$\text{GOR} = \frac{m_d h_{\text{fg}}}{Q_{\text{in}}} \quad (6.6)$$

h_{fg} is the enthalpy of vaporization of water calculated at water temperature inside dehumidifier.

Effectiveness of Humidifier (ϵ_H)

$$\epsilon_H = \frac{h_1 - h_3}{h_1 - h_2} \quad (5.7)$$

h_3 is enthalpy at the temperature of air leaving the humidifier and h_2 is obtained at the maximum temperature of air that can be reached i.e. water temperature inside humidifier.

Effectiveness of Dehumidifier (ϵ_{DH})

$$\epsilon_{\text{DH}} = \frac{h_3 - h_7}{h_3 - h_4} \quad (5.8)$$

h_7 is enthalpy at the temperature of air leaving the dehumidifier and h_4 is obtained at the maximum temperature of air that can be reached i.e. water temperature inside dehumidifier.

5.2 Water Heated Cycle

5.2.1 Effect of air flow rate and humidifier water level

The influence of water level in the humidifier and the air flow rate on the system productivity, input energy, GOR, humidifier and dehumidifier effectiveness and air relative humidity for the water heated system are shown in Figures 5-3 and 5- 4. Water level in the humidifier is varied from 4.5 to 7.5 cm at constant hot water temperature of 65°C, whereas water level in dehumidifier is fixed at 8 cm and at a temperature of 10°C. Readings are taken once the steady state conditions are reached. Although increasing the air flow rate decreases air relative humidity since more air is passed within a fixed quantity of hot water, distillate production and input energy increases with increasing air flow rate (Figure 5-3a, b). It implies that higher quantities of air passing per unit time are exposed to hot water and would accordingly be loaded with more water vapor. The input energy increases since larger volume of the air requires more heating energy to keep the water at a fixed temperature. Figure 5-3c shows the effect of increasing air flow rate on the system GOR. It is important, however to state that increasing the air flow rate leads to an increase in the production at a high rate compared with the rate of increase in heat input. Therefore, GOR slightly increases at higher air flow rate as shown in Fig. 5-3c. The figure also shows the effect of water level on the performance parameters. Decreasing water, level in the humidifier increases the amount of water evaporated, hence it increases production. However, it also increases the input heat significantly, which eventually reduces the GOR. It is important to state that GOR is low due to the absence of energy recovery in this system. Most of the energy used is consumed in water cooling in dehumidifier to control its water temperature

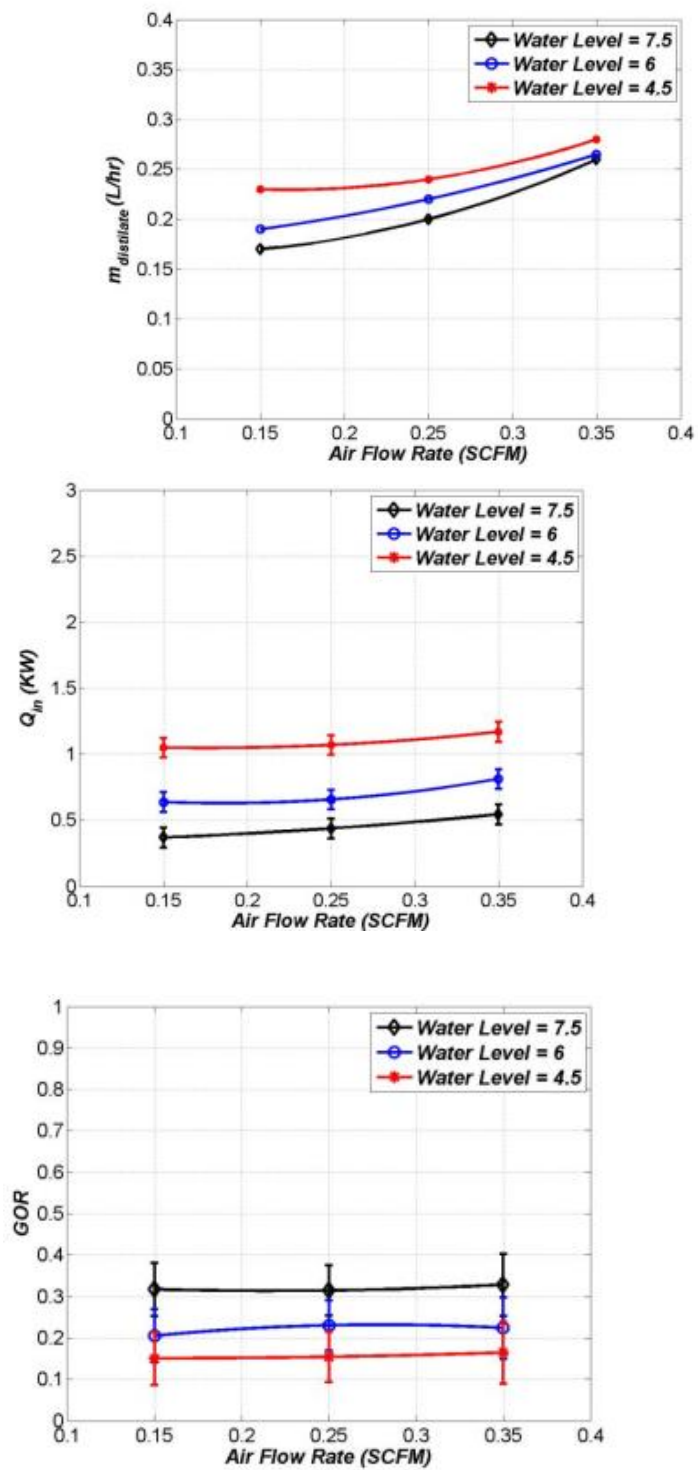
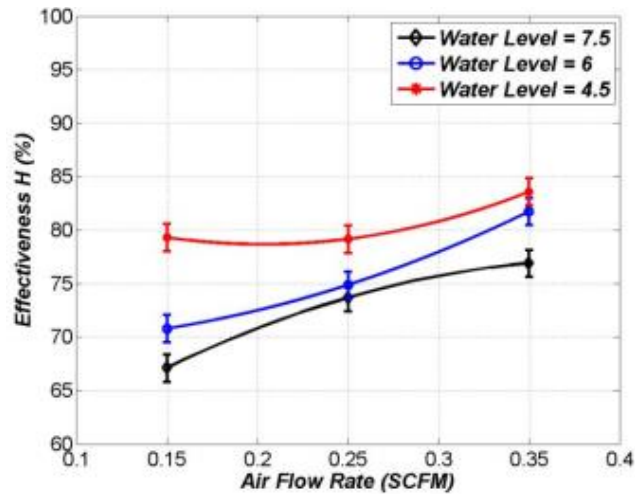


Figure 5-3 Effect of Air flowrate on a) production, b) energy required and c) GOR at different water levels in the Humidifier-Water heated system at $T_w = 65^\circ\text{C}$, $T_c = 10^\circ\text{C}$, $H_{wd} = 8\text{ cm}$

On the other hand, humidifier and dehumidifier effectiveness increase with increasing air flow rate (Figure5-4a, b) since air is loaded with more moisture and its enthalpy increases as it leaves the humidifier due to higher air temperature and humidity. Nevertheless, higher enthalpy of the air leaving the humidifier and flowing into the dehumidifier leads to better condensation of water vapor in the dehumidifier due to higher heat and mass transfer motive forces that leads to effective cooling that eventually leads to higher dehumidifier effectiveness. The decrease in the relative humidity with increasing air flow rate is attributed to more air is passing through a fixed water quantity, which limits air ability to absorb more water vapor. Moreover, water level in humidifier has an impact on the effectiveness. Decreasing water, level increases the amount of water evaporated since it increases heat and mass transfer coefficients as it becomes turbulent. Therefore, the effectiveness of the humidifier increases.



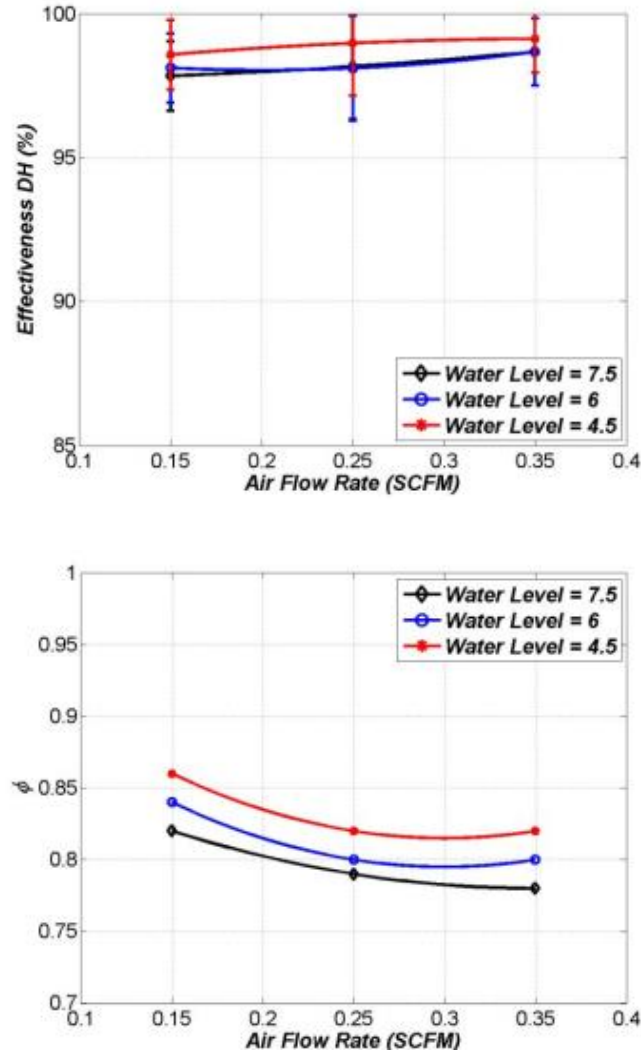
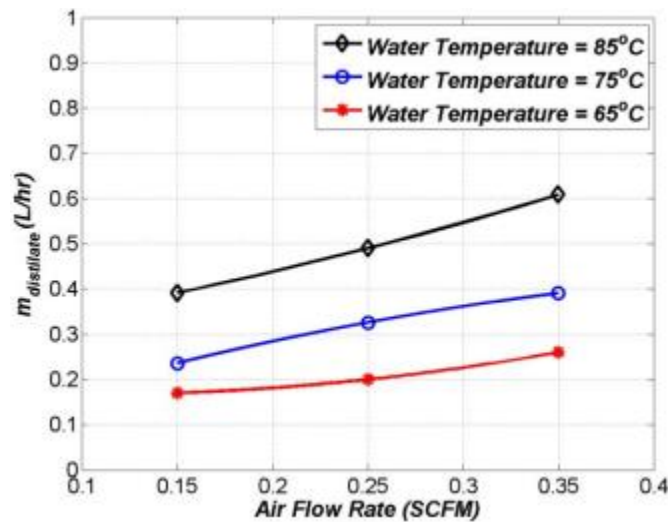


Figure 5-4 Effect of air flow rate on a) humidifier effectiveness, d) dehumidifier effectiveness, e) Relative humidity at different water levels in the humidifier for water heated system at $T_w=65^\circ\text{C}$, $T_c = 10^\circ\text{C}$, $H = 8\text{ cm}$

5.2.2 Effect of Brine Water temperature inside Humidifier

Bubble column HDH is analyzed while taking the effect of humidifier water temperature into consideration. For water heated systems, air is heated and humidified through its direct contact with hot water. This section investigates the effect of water temperature in humidifier on production, input energy, GOR, humidifier effectiveness, dehumidifier effectiveness and relative humidity at constant coolant temperature of 10°C as shown in

Figure 5-5 and 5-6. Water column height was maintained at 7.5 cm in the humidifier and 8 cm in the dehumidifier. Relative humidity increases with humidifier water temperature resulting in an increase in the distilled production as well as heat input, as show in Figure. 5-5a,b, since air can absorb more moisture when its temperature increases. Increasing water temperature would lead to higher air temperature as it leaves the humidifier, which increases the production. However, it increases required input heat in a rate that is higher than the distilled production rate. Therefore, GOR decreases with water temperature, as shown in Fig. 5-5.



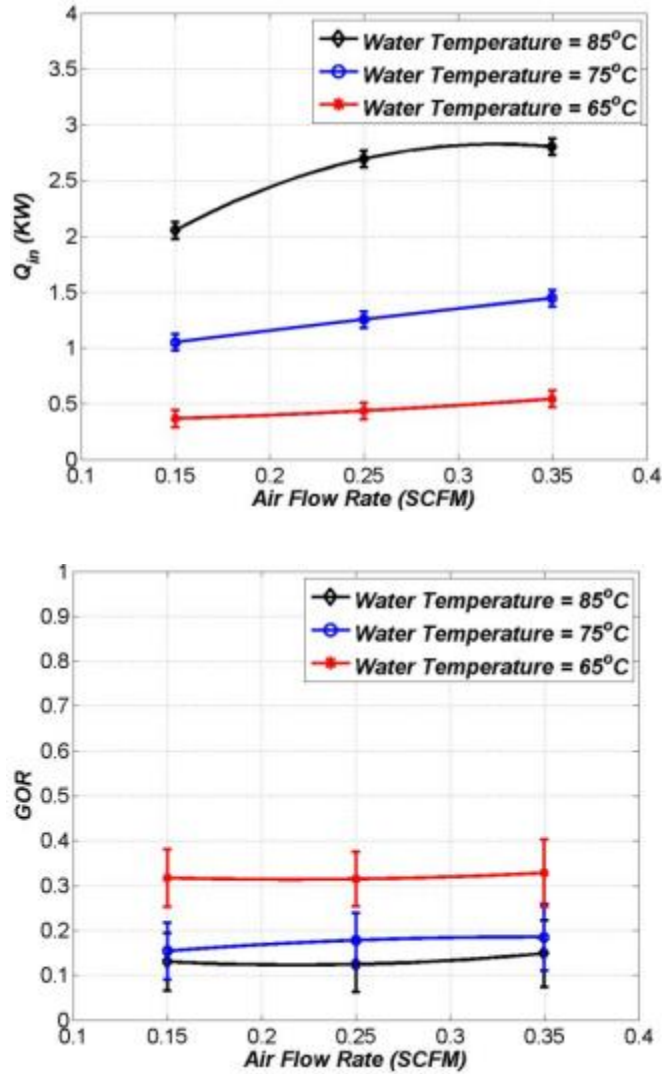


Figure 5-5 Effect of water temperature in humidifier on a) distillate, b) Q_{in} , c) GOR in the water heated system at water level in humidifier of 7.5 cm coolant temperature 10°C, and water level in dehumidifier = 8 cm

Increasing water temperature in the humidifier in a water heated HDH system increases the humidifier effectiveness as shown in Fig. 5-6a, because of the increase in air temperature and humidity ratio, leading to higher enthalpy of the air stream leaving the humidifier compared to the air inlet condition. The high enthalpy stream enters the dehumidifier that condenses the water vapor carried by the air effectively due to low water temperature inside

the tubes providing a better opportunity for condensation of water vapor on the tubes outer surface as show in Fig. 5-6b

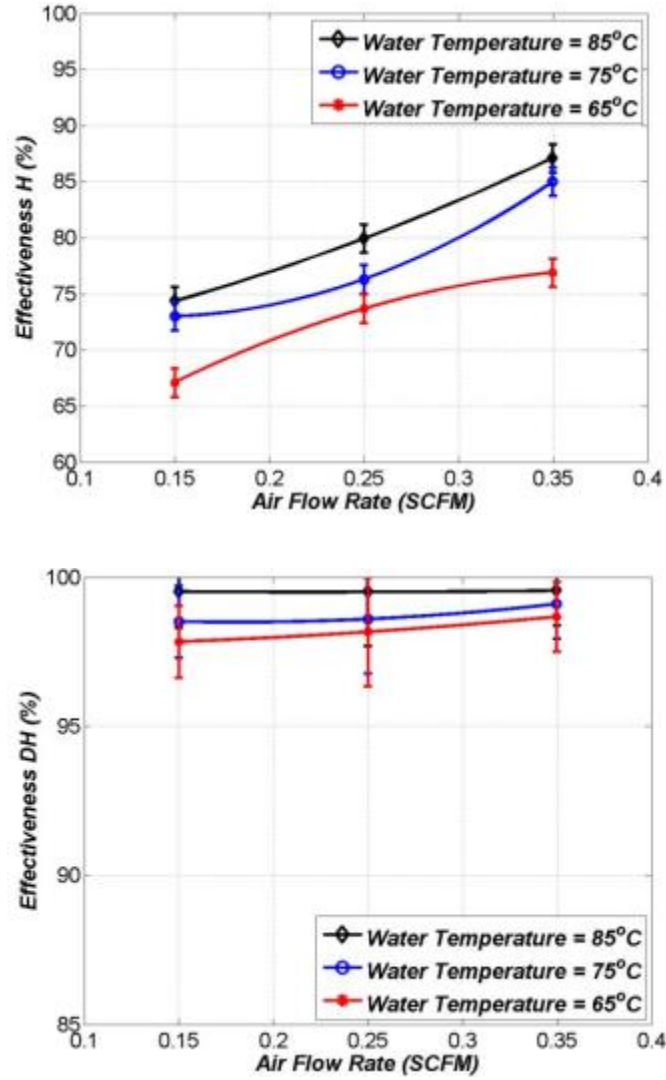


Figure 5-6 Effect of water temperature in humidifier on a) humidifier effectiveness, b) dehumidifier effectiveness, in the water heated system at water level in humidifier of 7.5 cm coolant temperature=10°C, and water level in dehumidifier =8 cm

5.3 Air Heated Cycle with Single Humidification and Dehumidification

The same experimental setup that was used for water heated cycle is also utilized to carry out a study for air heated cycle with modification such that a perforated plate of diameter of 11.5cm with 783 holes (Figure 5-8) has been placed inside the humidifier so that it can have the brine above the plate. Similarly, an air heater in form of cylindrical shape is installed before the humidifier to heat the incoming supply of air. To maintain the constant water level inside the humidifier a supply pipe of water is connected at top of the humidifier. The modified schematic is presented in Figure 5-7

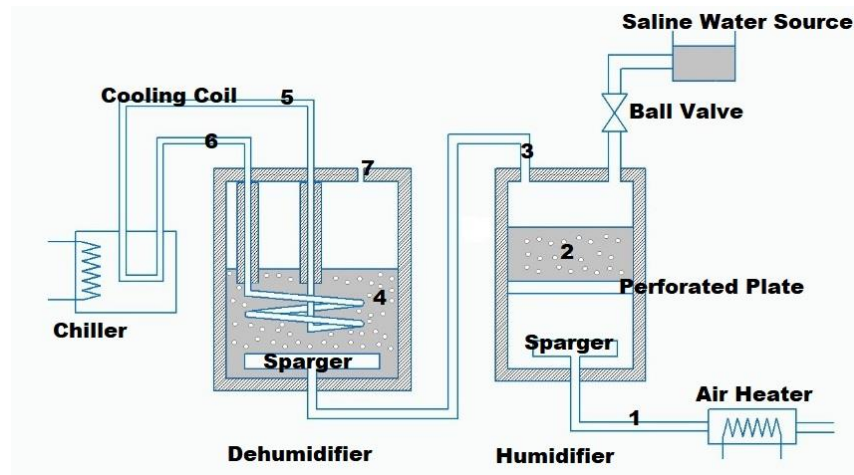


Figure 5-7 Setup Modification for Air heated Cycle Study

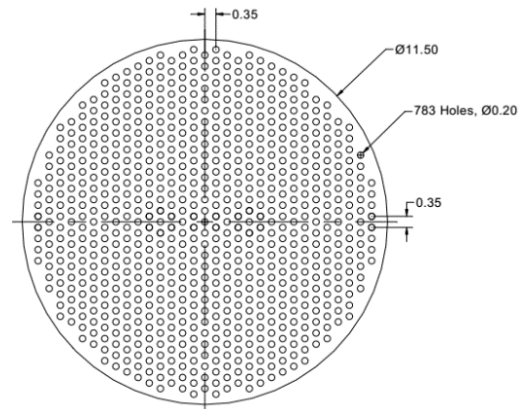
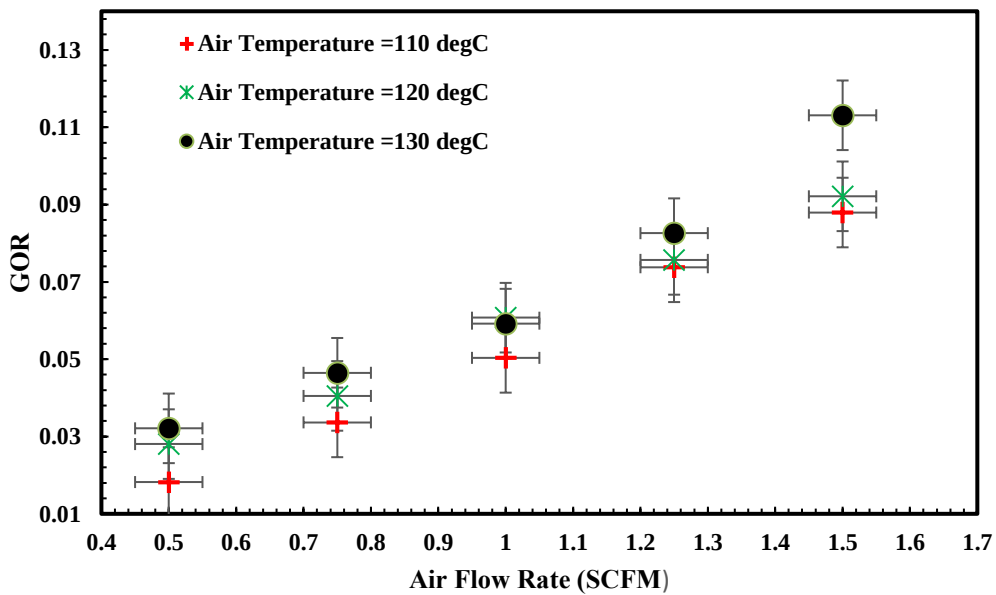
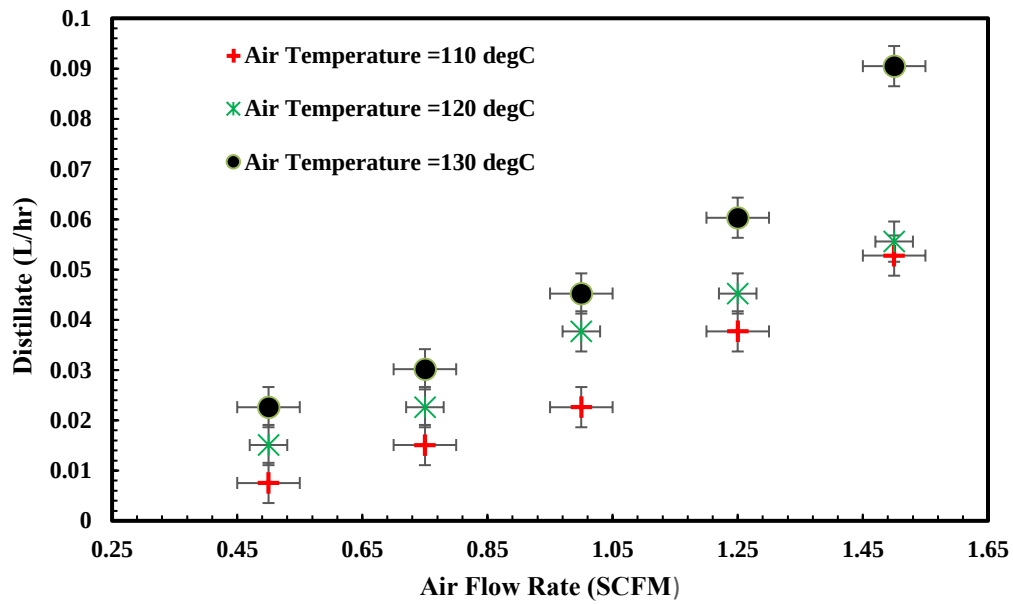


Figure 5-8 Perforated Plate Installed inside the Humidifier for air Heated Cycle

Like water heater cycle, a set of experiments was carried out for air heated cycle i.e. at five different flow rates of 0.5,0.75,1,1.25 and 1.5 SCFM at three air temperatures of 110°C,120°C and 130°C along with three different water levels of 2cm,3cm and 4cm. However, the limitation such as low exit temperature of air leaving the humidifier is a constraint to system GOR and productivity. Hence it is proposed that present system is modified with 02 stages of humidification with air re-heating to attain higher temperature after the humidifier. The following figure represents the low GOR and productivity cases that are obtained for air heated cycle.

5.3.1 Effect of Incoming Air temperature inside humidifier

The effects of air temperature and air flow rate to the humidifier in the air-heated system on the performance are presented in Figure 5-9. The high temperature of incoming air results in higher amounts of distillate productivity since air captures more water vapor at higher temperature. A maximum distillate rate of about 0.09 L/hr was achieved at air temperature of 130°C at 1.5 SCFM. This can be explained by the effects of relative humidity of air that decreases with the increase in air temperature. Increasing the temperature allows the air to hold higher amount of water at that temperature, which in turn increases the humidification of air leaving the humidifier. The higher temperature difference between air inlet to the dehumidifier and the temperature of the water column in the dehumidifier results in better condensation and thus leads to an increase in the productivity and GOR represent by trend in Figure 5-9 (b).



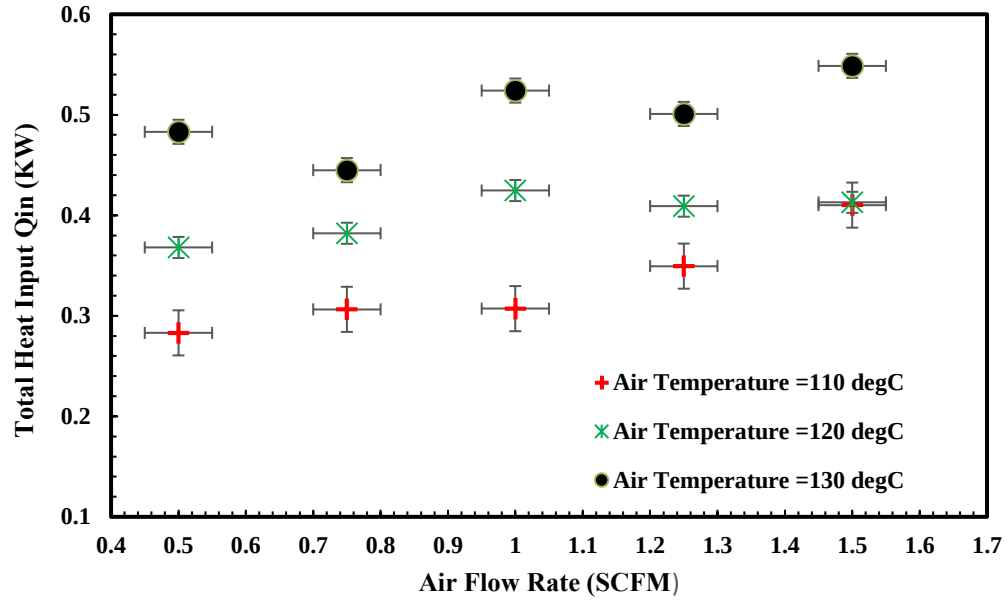
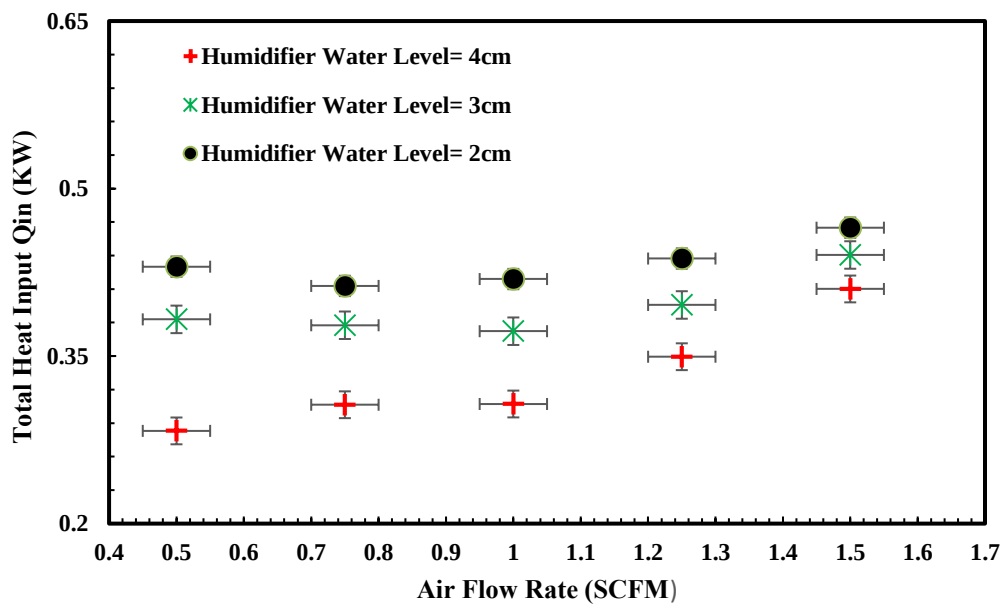
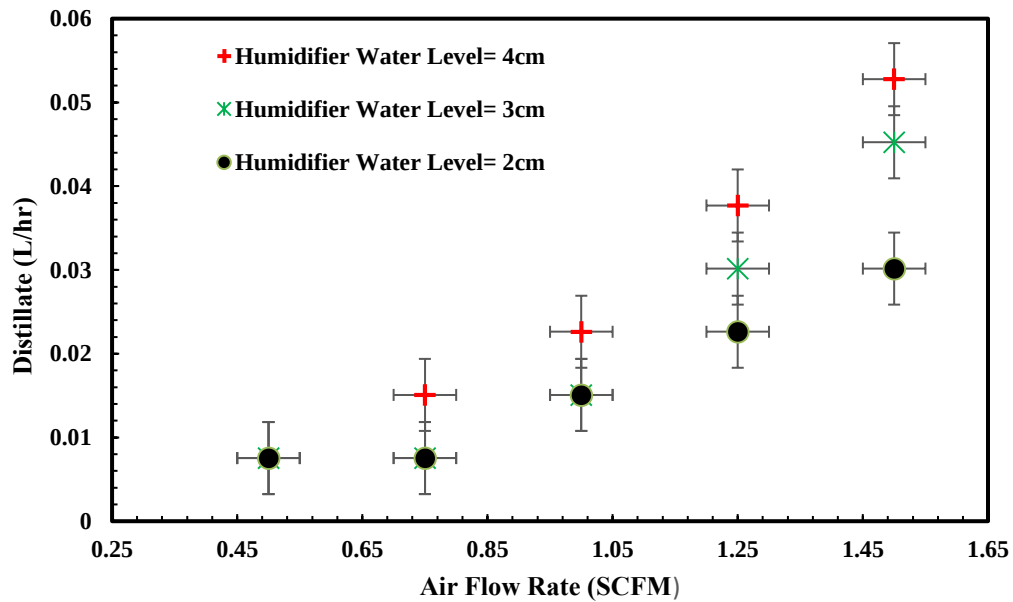


Figure 5-9 Effect of air flow rate and air temperature a) productivity, b) GOR c) Total Heat Input

5.3.2 Effect of Water level inside humidifier

The effects of water level in the humidifier and air flow rate on the performance of air heated bubble column HDH is shown in Figures 5-10. Air temperature was kept at a constant value of 110°C and level of water inside humidifier is varied as 2cm, 3cm and 4cm for every value of air flow rate. The water level inside the dehumidifier is kept at 6cm. The readings are taken once the steady state conditions are reached. Figure 5-10(a) shows that water production increases proportionally with increasing air flow rate. This means evaporation or humidification of air is strongly influenced by air flowrate. Also, the variation of humidifier water level has noticeable effect on distillate product. High water level inside the humidifier are the favorable condition for higher distillate quantity. In case of air heated cycle, the air stream is hot and accordingly it is humidified and cooled in the humidifier. The distillate is higher (around 0.05 L/hr) when saline water level in humidifier is 4cm. It is noticed that the energy required for cooling is more than energy required to heat air because of high mass flow rate of cooling water and its specific capacity. Therefore, the energy from chiller

is dominant in the total heat input. Hence, the total energy is almost linear with air flow rate. GOR increases with increasing air flow rate and with humidifier water level because of higher productivity.



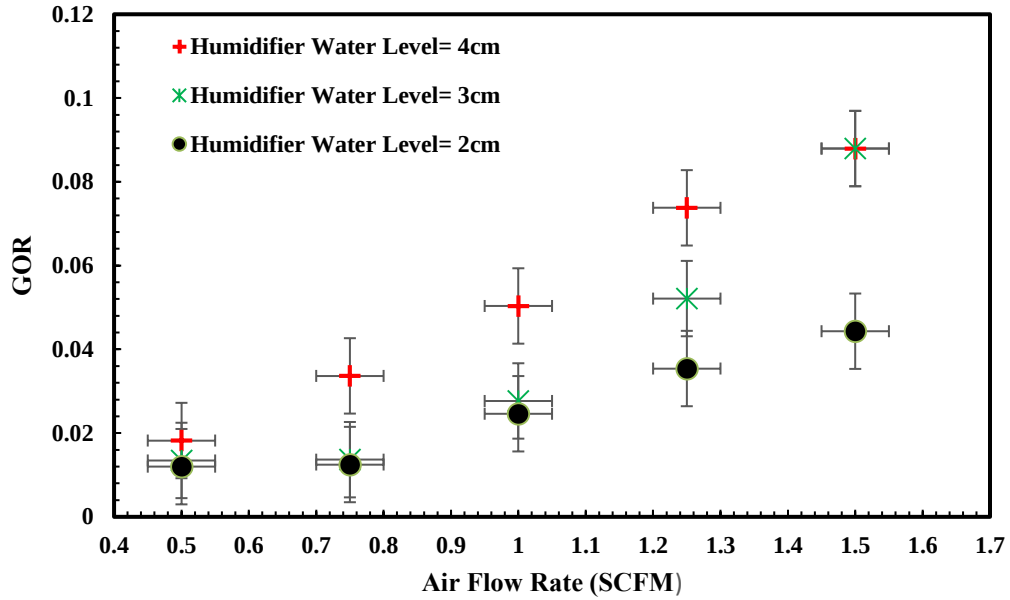


Figure 5-10 Effect of air flow rate and water level a) productivity, b) GOR c) Total Heat Input

Similarly Figure 5-11 represent the effect of incoming airflow rate and humidifier water level on the effectiveness of humidifier and dehumidifier bubble column. Humidifier effectiveness is found to be increasing with air flow rate and it is high for high water level inside humidifier. Since, at high water, more turbulence is observed so more heat and mass transfer which result in high effectiveness. For case of dehumidifier, effectiveness is found to be decreasing with air flow rate and high-water level. When water level inside humidifier is low, air leaving the humidifier would be at high temperature which make greater temperature difference between incoming air and column liquid in dehumidifier. Thus increasing of driving force result in improved heat transfer and effectiveness as shown in Figure 5-11(b).

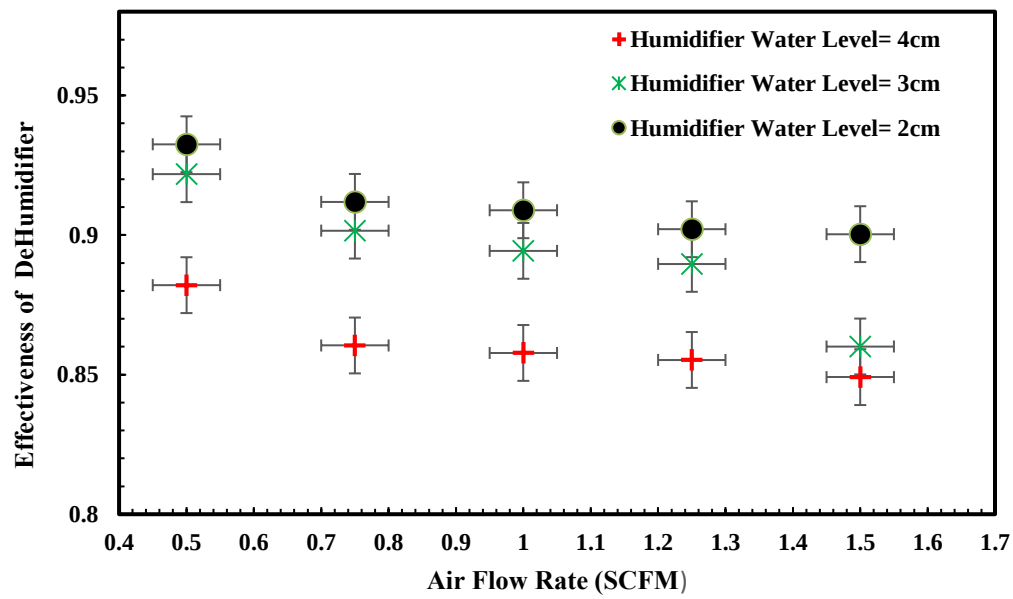
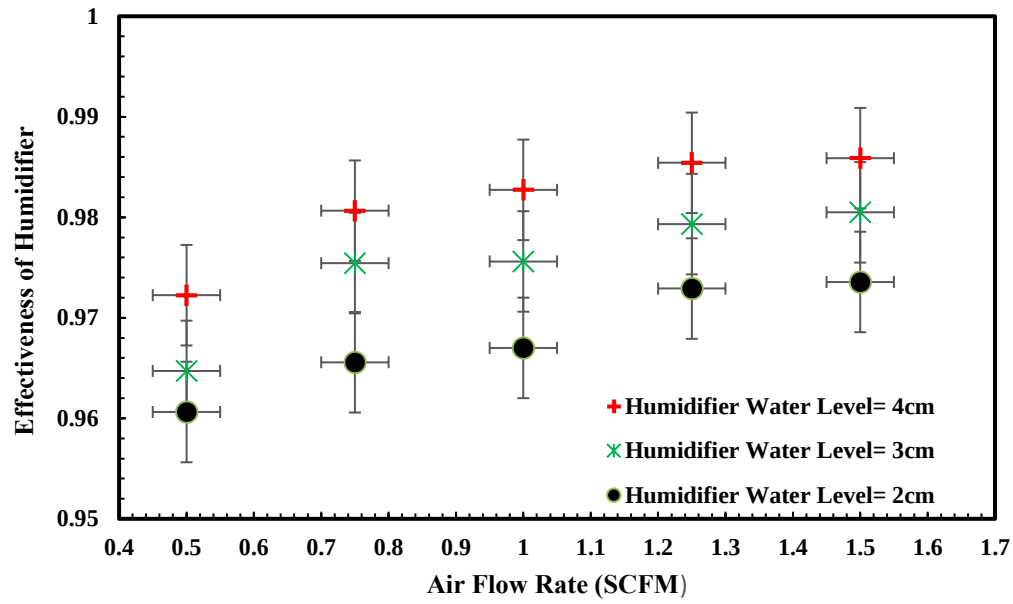


Figure 5-11 Effect of air flow rate and water level on Effectiveness of a) Humidifier, b) Dehumidifier

CHAPTER 6

EXPERIMENTAL INVESTIGATION OF AIR

HEATED CYCLE FOR TWO STAGE

HUMIDIFICATION DEHUMIDIFICATION

BUBBLE COLUMN SYSTEM

Some modifications are made to the existing setup for making it a two-stage humidification dehumidification system in order to improve air humidification process. An air heater is used to heat air leaving the blower. Air that leaves first humidifier is further heated in another air heater. Then air that leaves the second humidifier flows into the dehumidifier to condense the water vapor carried by the air in the two humidifiers as shown in Figure 6-1. The first heater is a box type heat exchanger where air passes through closed coils, which are heated hot water through electrical heaters. Whereas the second heater is simple closed air heater in cylindrical shaped. The rest of setup is the same as discussed in detail in chapter 6.

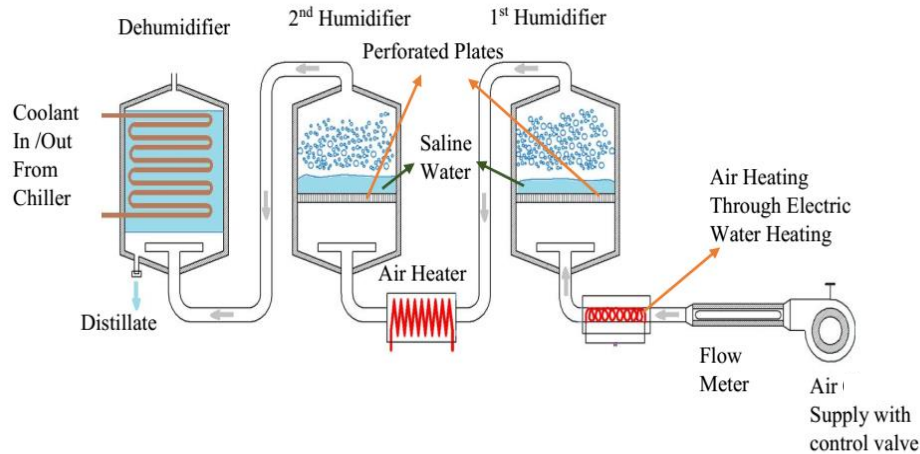


Figure 6-1 Double Stage humidification Dehumidification Air heated Cycle

For single stage humidification and dehumidification system of air heated cycle, the productivity is limited because the outlet temperature of air leaving the humidifier is not so high and limited. The modification of the system i.e. two humidifiers with reheating make possible for air to leave the final humidifier at high temperature. Thus, the air entered the dehumidifier at high temperature when compared with single stage system, and this way the productivity is increased in parallel with energy consumption. The Geometric Dimensions of the experimental setup can be summarized in Table 6-1

Table 6-1 Details of Experimental setup

<i>Bubble Columns</i>	<i>ID</i>	<i>OD</i>	<i>H</i>	<i>Liquid Height</i>	<i>Perforated Plate</i>
Dehumidifier	7cm	9cm	23cm	0-8cm	-
1 st Humidifier	9.8cm	11.5cm	20cm	0-5cm	168 holes
2 nd Humidifier	11.5cm	13.5cm	23cm	0-5cm	250 holes
<i>Heaters</i>	<i>Sizes/Dimension</i>			<i>Capacity</i>	
Air Heater 1	Box Type Exchanger LxWxH =8x5x6cm			02 Electrical heaters with 1.0 KW	
Air Heater 2	Cylindrical Shaper Direct heating			1.5 KW	
<i>Chiller</i>	Accel 500LC			Coolant rate = 15 lpm	
<i>Flow meter</i>	FL 50000 Omega			0.1-1.2 SCFM with ± 0.06 SCFM	

As all the experimental measuring instrument are same for that used for experimental setup described in chapter 6, so the uncertainties associated with experimental facilities is same and represented by previous Table 5-3

The results of indirect uncertainty on calculated variables evaluated through EES are shown in tabulated form as follow in Table 6-2.

Table 6-2. Indirect Uncertainty on Calculated Results

<i>Parameter</i>	<i>Indirect Uncertainty</i>				
	<i>Air Heated cycle</i>				
	0.5 SCFM	0.75 SCFM	1 SCFM	1.25 SCFM	1.5 SCFM
Distillate	±0.006103	±0.006233	±0.00652	±0.00661	±0.00626
GOR	±0.03123	±0.03115	±0.03111	±0.03555	±0.03666
Total heat Input	±0.01823	±0.018553	±0.01856	±0.01800	±0.01865

The performance parameters are same as discussed in section 5.1.2

6.1 Effect of Incoming Air flow Rate and its Temperature.

Performance parameters of the system such as productivity and GOR is evaluated through varying the inlet temperature of air to 1st Humidifier from 110 °C to 130 °C at five different flow rates of air from 0.5 SCFM to 1.5 SCFM. The water level inside Humidifier 1 and Humidifier 2 is maintained at 5cm. The water height inside the dehumidifier is fixed at 6cm at the beginning of each set of readings. The increase in water level inside dehumidifier represent the distillate produced while the decrease in saline water of

humidifier represent the humidification process. All readings are taken once the steady state condition is reached.

Increase of air flow rate increase the distillate. Air is carrying more water vapors from the humidifier with high mass flow rate. The higher the temperature of incoming air gives high productivity. With the increase of air temperature, the relative humidity decreases and hence the ability of air to carry more water vapor increases which can be clearly seen from the Figure 6-2 . Figure 6-2 illustrate the comparison of productivity of single stage HDH system with Double stage HDH system at two different inlet air temperature of 110°C and 130°C. Productivity is higher for case of high air temperature. Enhancement of productivity in double stage HDH system is about 22% evaluated at 130°C while it is 12% for case of 110°C. Higher the temperature leads more evaporation of water vapors in Humidifier 1 and more temperature after reheating from heater 2. This condition favors the productivity and result in high productivity. Maximum distillate is 0.2 liter/h obtained at 1.5SCFM with air temperature of 130°C for double HDH system while the corresponding distillate for single stage is 0.09 L/h. Figure 6-3 compares the total heat input for single stage and double stage HDH system. Amount of heat input is 7.5% approximately higher for double stage HDH system because of additional heat energy in heater 2. Since mass flow rate of air is low hence the additional heat input is also minute in term of magnitude (product of mass and heat capacity of air). The nature of graph is almost linear which shows that heat input system increases with the air flow rate. This is obvious as more energy is required for heating greater mass of air to achieve a fixed temperature. Figure 6-4 compares the performance of single and double stage HDH system in term of Gain Output Ratio (GOR). Figure 6-4 shows that GOR is increasing with airflow because of high productivity

obtained at high air flow rates. The GOR values are greater for case of 130°C. GOR is 0.16 at 1.5 SCFM for 130°C for double stage while it is 0.11 compared at same parameters for single stage HDH system. The average enhancement in GOR from double stage system is due to high distillate and it is about 24% higher (on average) from single stage system.

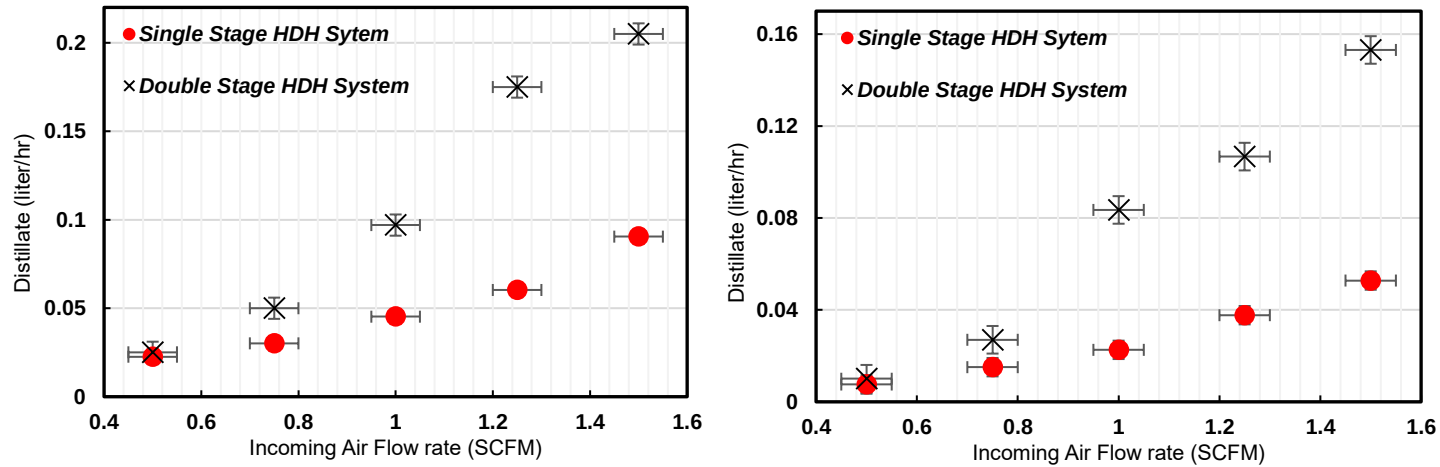


Figure 6-2 Comparison of Distillate of Double stage and Single Stage HDH System for
a) Air temperature = 130 °C b) Air temperature = 110 °C

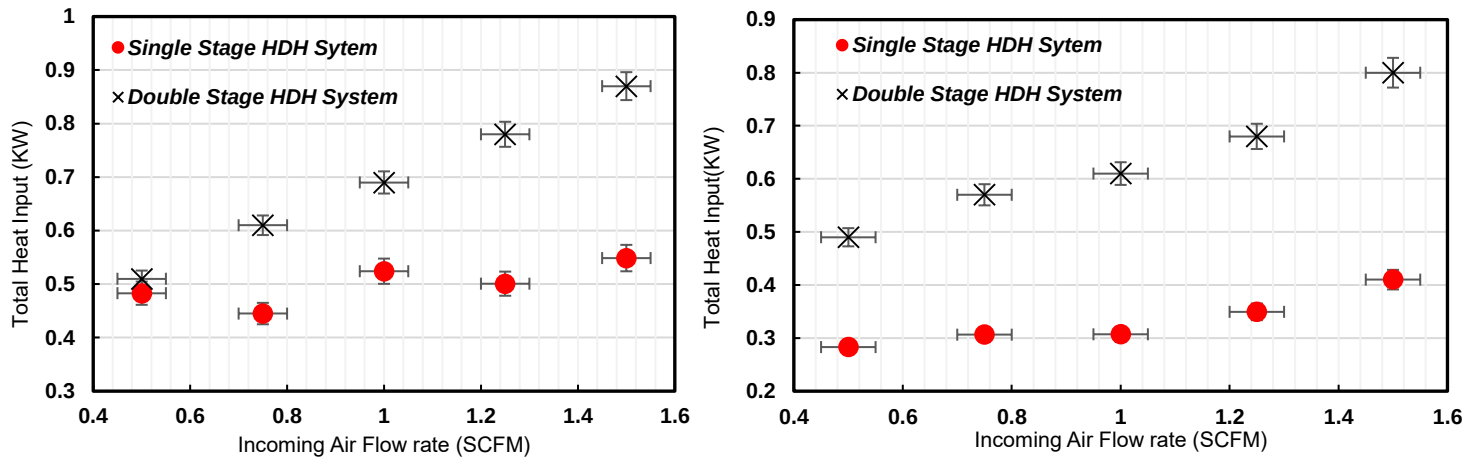


Figure 6-3 Comparison of Total Heat Input of Double stage and Single Stage HDH System for
a) Air temperature = 130 °C b) Air temperature = 110 °C

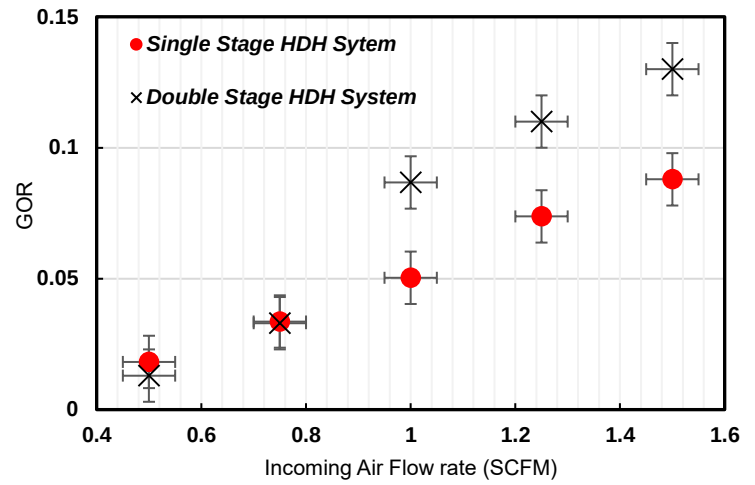
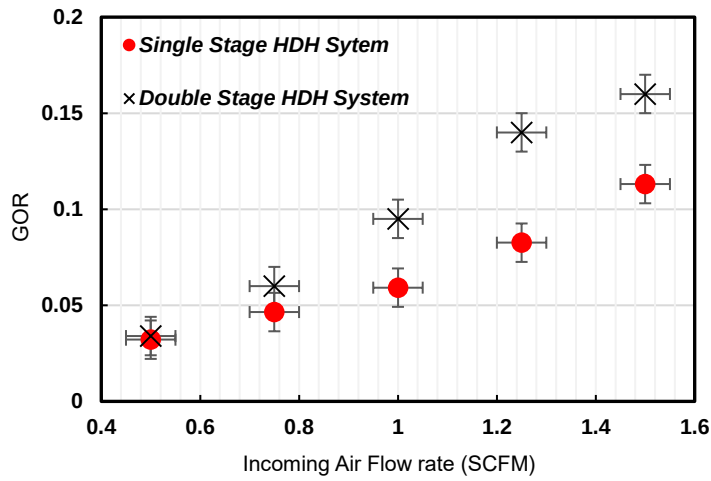


Figure 6-4 Comparison of GOR of Double stage and Single Stage HDH System for
a) Air temperature = 130 °C b) Air temperature = 110 °C

6.2 Effect of Incoming air flow rate along with brine water level in Humidifier.

To examine the effect of water level inside the humidifier, Humidifier 1's level is made constant at 5cm while the Humidifier 2' level is changed as 4cm, 3cm and 2cm while temperature of inlet air to first humidifier maintain constant at 110°C. All readings have been taken after system reached steady state conditions for 05 different airflow rates of air.

The increment of parameters such as distillate, GOR and heat input with flow rate has been discussed already in section 6. Here it is important to focus the effect of humidifier level on the mentioned parameters for both HDH systems. Figure 6-5 represent the comparison of productivity of single stage HDH system and double stage HDH system being evaluated at humidifier water level of 2cm and 4cm. Productivity of both systems is found to be high for the case of high-water level i.e. 4cm as shown in figure 6-5(a). The maximum

productivity obtained in Double Stage HDH system at 1.5SCFM is 0.3 Liter/h which is approx. 10 times higher than the fresh water quantity from single stage HDH system. On average basis, the productivity is found to enhanced by 20-25% by double stage HDH system. Figure 6-6 compares the total heat input of both HDH system (Double stage & Single Stage) at two different water level inside humidifier (4cm & 2cm) at fixed input air temperature of 110°C. Heat input exhibit almost linear trend with airflow rate since more heat is required to raise more mass of air upto fixed temperature. Since air is heated twice in double stage HDH system, hence heat input requirement is higher than single stage. But as the air flow rate is low hence there is minute difference in terms of magnitude occurred due to low value of product of mass and heat capacity of air. Double Stage HDH system consume 9.5% more than single stage HDH system.

Figure 6-7 depicts the performance of Single stage and Double Stage HDH system in term of Gain Output Ratio (GOR). As the distillate is high for double stage in parallel with minute consumption of heat input so GOR is also higher for double stage. The increment of GOR is about 32% on average basis from Single Stage system. The maximum GOR is obtained from the double stage HDH system at humidifier water level 4cm and 1.5 SCFM is 0.4 in comparison with single stage system of 0.088. The substantial difference of GOR for Double stage system is mainly due to high distillate. Reheating the air in 2nd heater enable the air to carry more vapor at exit at higher temperature from 2nd humidifier which were not happening in case of single stage HDH system.

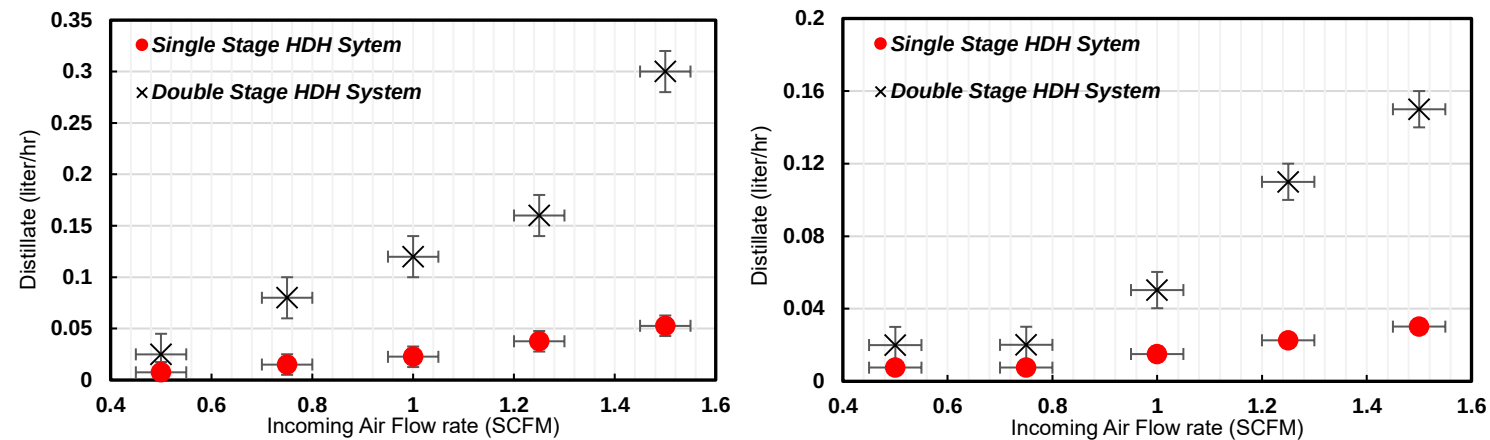


Figure 6-5 Comparison of Distillate of Double stage and Single Stage HDH System for

a) Humidifier water level = 4cm b) Humidifier water level = 2cm

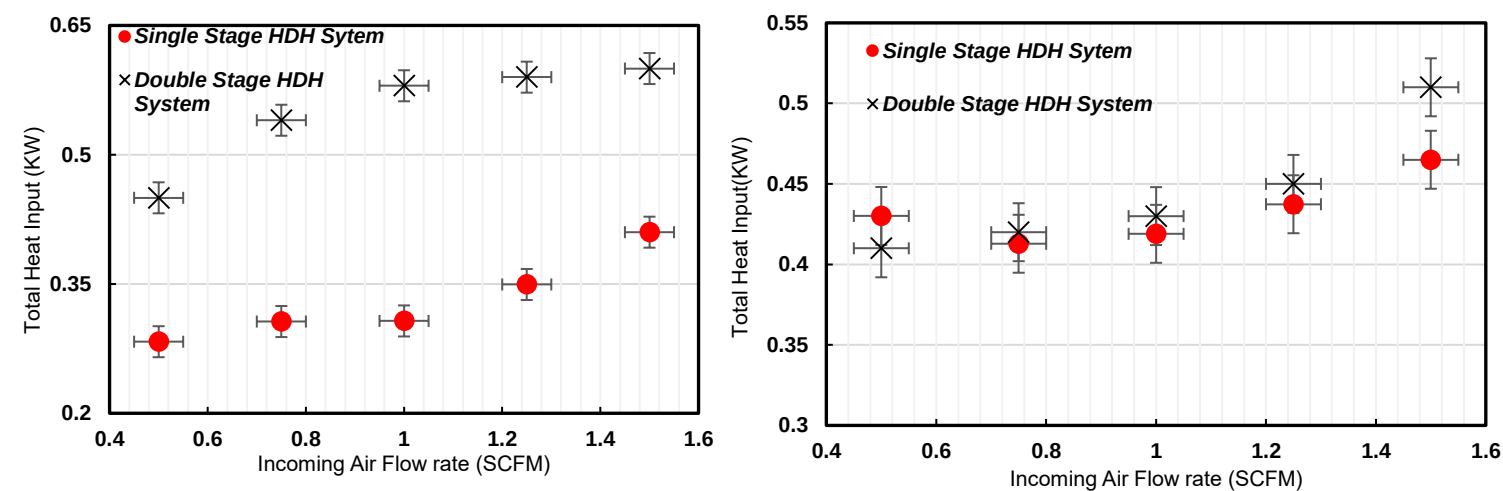


Figure 6-6 Comparison of Total Heat Input of Double stage and Single Stage HDH System for

a) Humidifier water level = 4cm b) Humidifier water level = 2cm

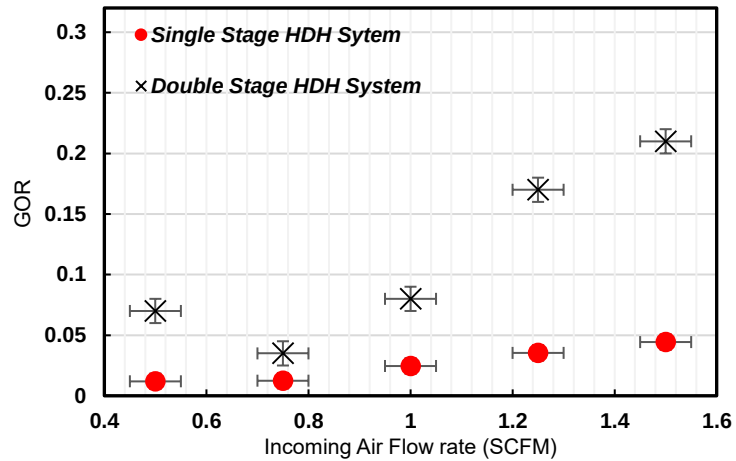
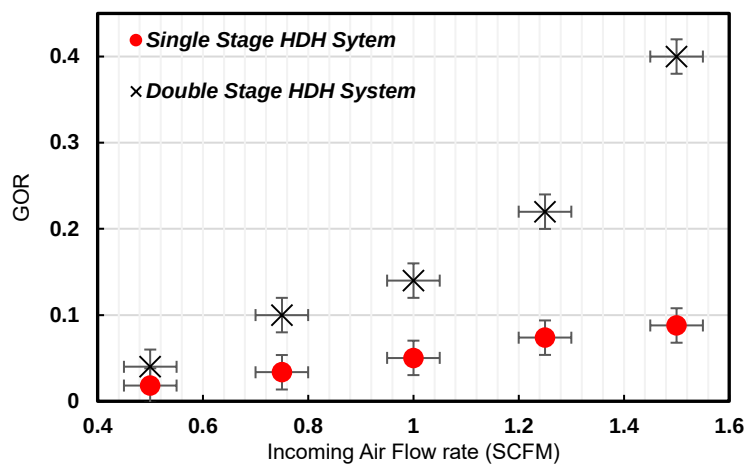


Figure 6-7 Comparison of GOR of Double stage and Single Stage HDH System for

a) Humidifier water level = 4cm b) Humidifier water level = 2cm

6.3 Maximum Performance of the system.

From the previous analysis, GOR and distillate production rate, which are important parameter in HDH desalination, are increasing with airflow rate for the above system. Hence apart from the above parametric study. The system has been tested for maximum condition to evaluate the maximum performance parameters. As per system limits, air flow rate is set 1.5 SCFM and following parameters have been shown as results

Maximum Performance parameters

- 1- *GOR of the system = 0.4*
- 2- *Distillate produced = 0.3 Liter/h*
- 3- *Total Heat input = 0.60 Kw*
- 4- *Dehumidifier effectiveness = 0.98*
- 5- *Humidifier effectiveness = 0.97*

Based on following operating conditions

- 1- *Incoming air flow rate = 1.5 SCFM*
- 2- *Max. air inlet temperature to Dehumidifier = 60°C*
- 3- *Max air inlet temperature to Humidifier = 122°C (@1.5 SCFM)*
- 4- *Water level inside the Humidifier 1 & 2 = 4cm both*
- 5- *Time of evaporation of water levels = 60min*

6.4 Comparison of Single & Double Stage HDH Bubble Column System.

Here a quick comparison of single stage HDH bubble column system with double stage HDH bubble column system is presented for air-heated cycle. For comparison, operating parameters selected are only which are possible for both systems. Hence systems are operated at 1.5 SCFM air flow rate with water level maintained 4cm and air temperature is fixed at 110°C. The double stage humidification despite of additional heating load performs well in term of productivity and GOR shown in Figure 6-8a. The GOR of double stage humidification system is 0.40 enhanced from 0.087 of Single Stage HDH system. Enhancement of productivity is substantial i.e. 0.3 Liter/hr against 0.05 Liter/hr shown in Figure 6-8b.

The major limitation of system with one humidifier is that air exit at quite lower temperature from the humidifier. Hence, we overcome this problem by using two stage heating humidification of the system. For single stage air HDH system air maximum attained 120°C after heating and leaves at max. 55 °C while for case of double humidification with reheating 2nd heater heats the air upto 140°C and this make air to exit from final humidifier at 70°C represented by Figure 6-9. The flow of modified air heated system is presented on psychometric chart in Figure 6-10.

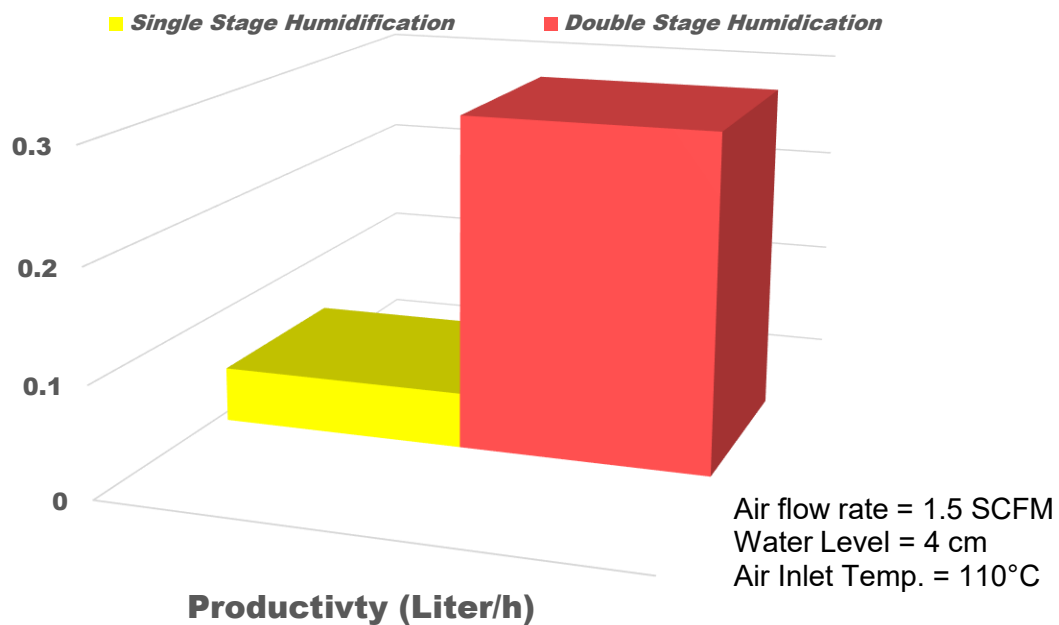
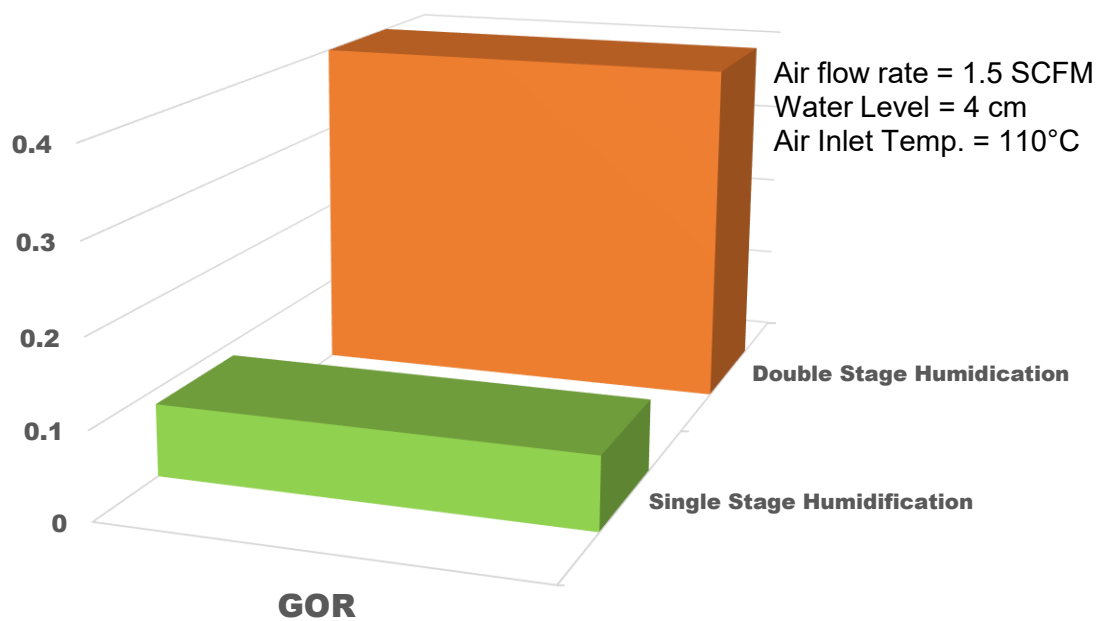


Figure 6-8 Comparison of a) GOR b) Distillate of Single Stage Humidification and Double Stage Humidification at Same operating Condition through 3D bar Graphs

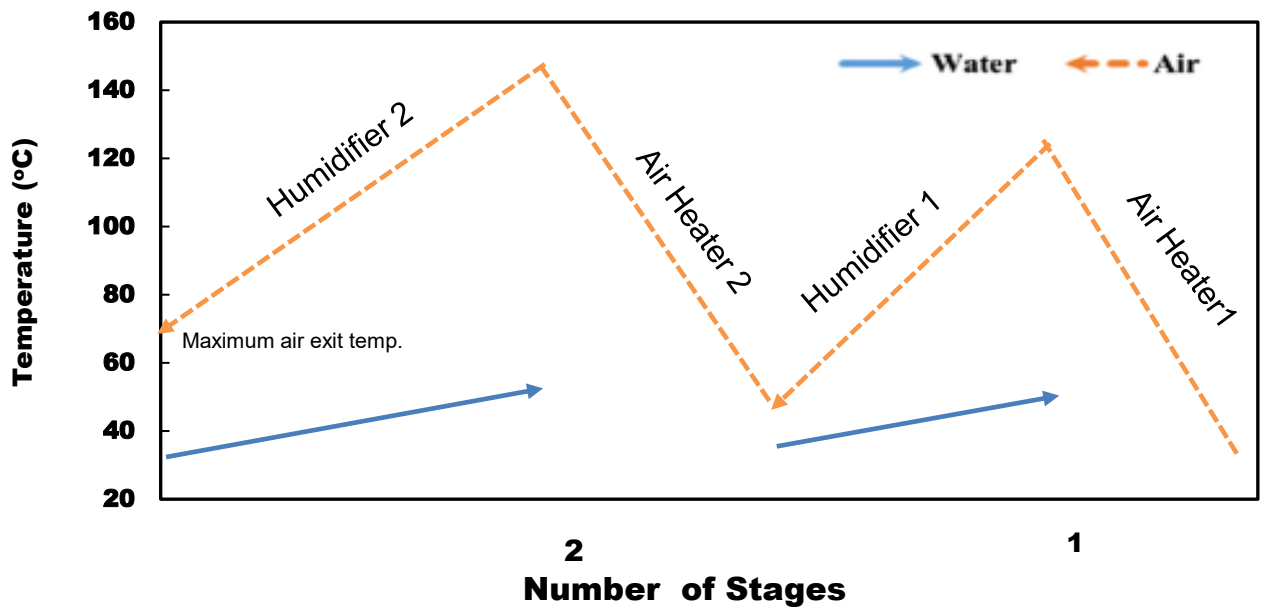
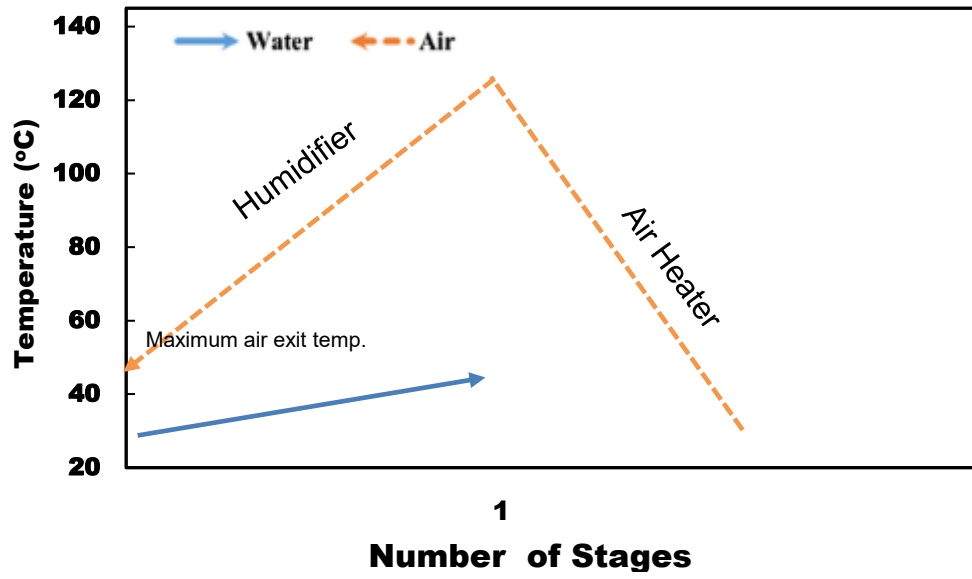


Figure 6-9 Temperature profile with limitation of single stage and increment of double stages Humidification Dehumidification air heated cycle

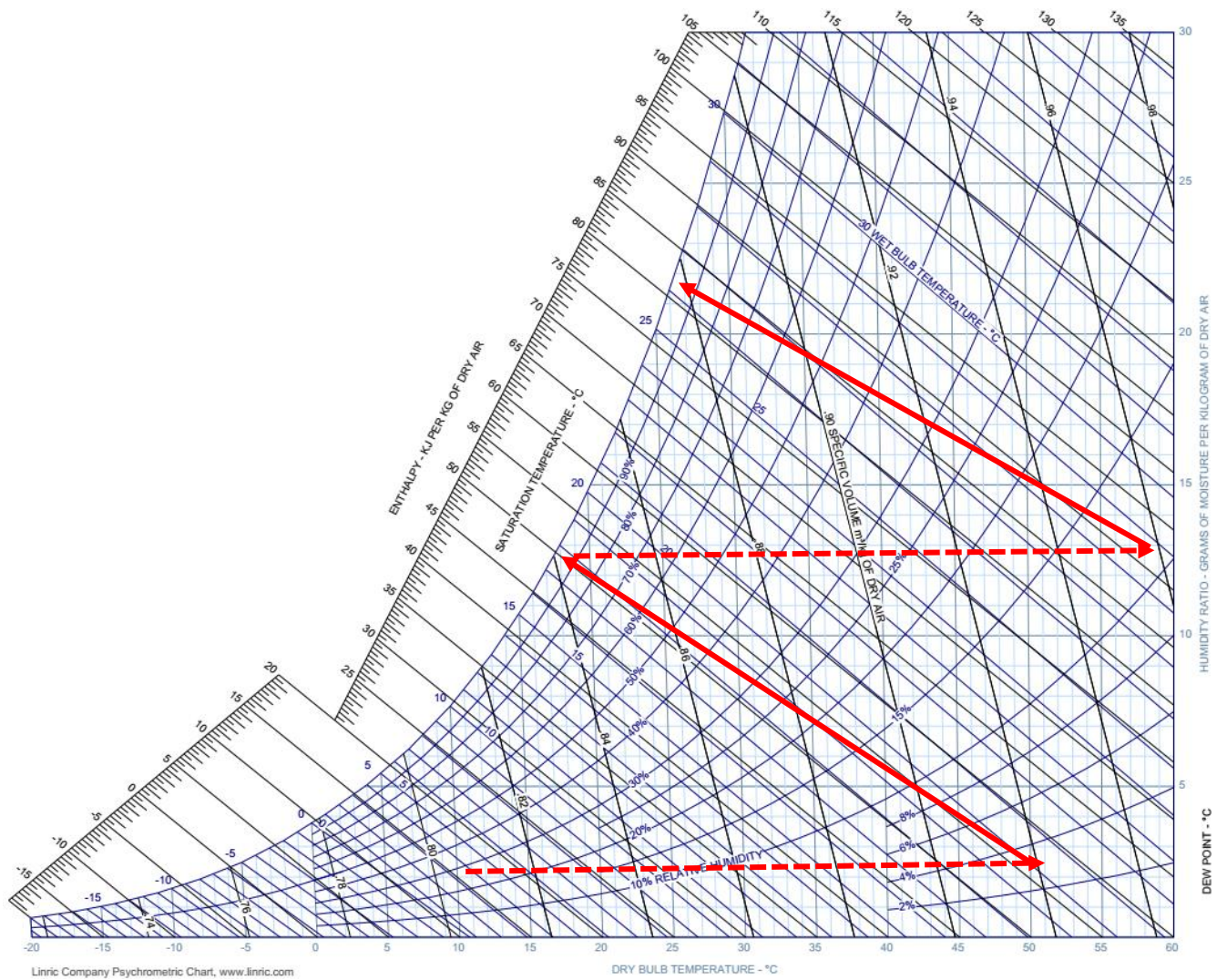


Figure 6-10 Humidification and Heating for Double stage air heated Cycle.

CHAPTER 7

CONCLUSION

In this Thesis, application of bubble column technique for Humidification Dehumidification Desalination is studied in detail. Usage of bubble column in field of thermal desalination with HDH system has been critically evaluated from literature work covering all experimental and theoretical scope. The research study presented in this thesis on Bubble Column HDH is divided in two main segments; development of modified mathematical model for bubble column dehumidifier and experimental assessment of novel configuration for air heated cycle & water heated cycle with bubble column HDH system.

A modified and comprehensive analytical model for bubble column dehumidifier is presented. The model explains the detailed mechanism of heat and mass transfer inside bubble column dehumidifier. This model preciously deals with shallow bubble column features which is actual suitable for used in HDH desalination. Heat and mass transfer coefficients of inside and outside the gas bubble is incorporated in model which were previously unconsidered and were elusive to analyze. Additionally, geometrical effects and fluid flow characteristic for short column of HDH desalination have been included in the modeling as pioneer effort. Noteworthy, the model greatly predicts the heat transfer inside/outside the condensing coil through improved correlation which were found in conflict in previous work. Proposed Model is validated with experimental

work available in literature and found in closed agreement with 11% accuracy. Five thermal resistances ($R_{B,inside}$, $R_{B,outside}$, R_{column} , R_{coil} , $R_{coolant}$.) and mass transfer resistance (R_m) have been studied against the variation of operational parameters of dehumidifier like air inlet flow rate, air temperature, operating air pressure, bubble diameter and coil placement height.

For the variation of air temperature from 45°C to 70°C, all resistances are found to be decreasing by 10-12%, except the bubble outside thermal resistance $R_{B,outside}$, because of the reduction of driving force of condensation. Mass transfer resistance R_m is greater in magnitude with average value of 7.5 s/m, while bubble outside resistance $R_{B,outside}$, is the lowest with value of 0.0002 K/kW. Thus, overall impact of air temperature is to increase the heat transfer inside bubble column dehumidifier. Unlike air flow rate, air temperature has not much significance impact on resistances. For variation of air flow rate from 1 SCFM to 3 SCFM, mass transfer R_m and gas side resistances ($R_{B,inside}$ & $R_{B,outside}$) are unaffected with constant values of 7.2 s/m and 0.135 K/kw respectively. However, column resistance R_{column} decreases significantly by 25% because of high gas hold up, leading to high heat and mass transfer.

Among all operational parameters, bubble diameter found to be significant impact on performance of the bubble column dehumidifier. With the increment of only 1mm diameter, mass transfer R_m increases by 25% from 6 s/m to 7.5 s/m. Similarly bubble inside resistance $R_{B,inside}$ increased by 36%. Thus, increased of diameter effects substantial reduction of heat transfer in bubble column dehumidifier.

The model explains the effect of sparger in shallow bubble column by identifying the critical height. The effect of sparger is significant upto certain height, where heat

transfer increase sharply. For present case the critical height is 5cm at which thermal resistance of column liquid exponential decreases from 0.12 K/kW to 0.06 K/kw i.e. almost by 50%. Above critical height, the impact of sparger is not so much noticeable and hence column resistance is nearly constant at 0.06 K/Kw irrespective of height. Similarly, operational pressure of dehumidifier is to increase the heat transfer total heat transfer found to be increase from 0.25KW to 0.37KW (i.e. 48%) for pressure range of 1-8 bar.

Experimentally, this thesis presents the comparison of air heated and water heat cycle in novel configuration of Bubble Column Humidifier dehumidifier system. Effect of water level and incoming air/ water temperature has been analyzed on system performance by varying the water level from 4cm to 2cm and incoming temperature from 110°C to 130°C. Water heated cycle found to have maximum productivity of 0.6 liter/h and GOR of 0.4 at 0.35SCFM with water temperature 85°C. Productivity increases and GOR decreases with decrease of humidifier water level while GOR increases and productivity decrease with decrease of humidifier water temperature. For air heated cycle, maximum productivity obtained is 0.09 liter/hr and GOR of 0.11 at 1.5 SCFM with air temperature of 130°C. Higher the incoming air temperature and greater the humidifier water level resulted in higher the productivity and GOR.

In comparison with water heated cycle, GOR of air heated is low and limited due to low temperature of air after humidification. Thus, experimental system is modified with double stage bubble column humidification with reheating system. Air heated cycle

investigated on this modified system found to be effective, at same operating conditions, the productivity is enhanced by 12%, GOR is increased by 24% with energy consumption increased by 7.5%. The maximum productivity and GOR achieved are 0.3 lit/h and 0.4 respectively at 1.5 SCFM flow rate and 4cm water level for double stage HDH system

CHAPTER 8

RECOMMENDATION

Following are the recommendation based on this thesis work for future research.

- There is great conflict on effect of direct hitting of bubbles on condensing coil on bubble column dehumidifier performances in literature. Some correlation can be developed to investigate this effect analytically.
- No work is done on the heat transfer correlation for inside liquid of bubble column dehumidifier to encounter the effect of liquid height and different cross-sectional area of bubble column.
- A complete model is required to analyze the Bubble column Humidifier and Dehumidifier system in combined form.
- Correlation and additional modeling equations are needed to predict the heat transfer mechanism above the column liquid surface.
- The effect of sparger holes size, pitches and their placement height can be investigated experimentally to obtain effect on system performance.
- The GOR of air heated cycle is experimentally low, there is wide range of improvement and modification that can be done to experimental setup to achieve higher GOR.

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“Analytical modeling of Heat & Mass transfer of Shallow Bubble Column Dehumidifier” (progress)

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