

205
PARTIAL LOAD PERFORMANCE ANALYSIS OF
LIQUID SORBENT/COOLING DEHUMIDIFICATION
SYSTEMS

by

ARSHAD Y. KHAN

B.E., N.E.D. University of Engineering, Karachi, Pakistan,
1978

A MASTER'S THESIS

submitted in partial fulfillment of the
requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1983

Approved by:

W. Ball

Major Professor

**THIS BOOK
CONTAINS
NUMEROUS PAGES
WITH ILLEGIBLE
PAGE NUMBERS
THAT ARE CUT OFF,
MISSING OR OF POOR
QUALITY TEXT.**

**THIS IS AS RECEIVED
FROM THE
CUSTOMER.**

**THIS BOOK
CONTAINS
NUMEROUS PAGES
WITH THE ORIGINAL
PRINTING BEING
SKEWED
DIFFERENTLY FROM
THE TOP OF THE
PAGE TO THE
BOTTOM.**

**THIS IS AS RECEIVED
FROM THE
CUSTOMER.**

LD
2668
.TY
1983
K48
C.2

A11202 570840

ACKNOWLEDGMENTS

I would like to thank my advisor Dr. Herbert D. Ball for his constant guidance and encouragement throughout my studies. Thanks are also extended to Dr. N. Z. Azer and Dr. Charles K. Spillman for serving on my committee. Acknowledgment is also extended to Margit McCabe for careful and efficient typing.

TABLE OF CONTENTS

CHAPTER	Page
I	INTRODUCTION 1
	1.1 Refrigeration 2
	1.2 Solid Desiccant System 8
	1.3 Liquid Desiccant System 10
II	LIQUID DESICCANTS 13
	2.1 Dehumidification by Liquid Sorbents 17
	2.2 Liquid Sorbent Dehumidification Systems . . . 19
	2.3 Niagara Blower Company's "Hygrol" System . . 27
	2.4 Midland-Ross Corporation's Kathabar System . 32
III	MODELING OF COMPONENTS 37
	3.1 Mathematical Model for Absorber Analysis . . 43
	3.2 Cooling Tower 53
	3.3 Sorbent Solution Cooler 60
IV	PERFORMANCE ANALYSIS AND RESULTS 63
	4.1 Computer Model 65
	4.2 Analysis and Results 70
	4.3 Regenerator Analysis 103
V	CONCLUSIONS AND RECOMMENDATIONS 112
	5.1 Conclusions 112
	5.2 Recommendations 114
	BIBLIOGRAPHY 116
	APPENDIX A COMPUTER PROGRAM 120

LIST OF TABLES

Table 3.1:	Comparison of Equation (3.34) with the Figure 3.5	59
Table 4.1:	Performance of the liquid sorbent - cooling tower system using a 40% solution	70
Table 4.2:	Performance of the liquid sorbent - cooling tower system using a 35% solution	73
Table 4.3:	Performance of the liquid sorbent system using a 40% solution with an external source of chilled water	80
Table 4.4:	Performance of the liquid sorbent system using a 30% solution with an external source	83
Table 4.5:	Performance of the liquid sorbent system using a 40% solution for different controlling parameters.	90
Table 4.6:	Performance of the liquid sorbent system with modified configuration using a 40% solution	103
Table 4.7:	Performance of the liquid sorbent system with modified configuration using a 30% solution	105
Table 4.8:	Regeneration performance of the system for different solution requirements	108

LIST OF FIGURES

Figure 1.1	Schematic diagram of mechanical refrigeration equipment	4
1.2	Psychrometric diagram of cooling and dehumidification process by using a Refrigeration system.	5
1.3	Schematic diagram of absorption refrigeration system	6
1.4	A solid desiccant rotary bed unit	9
1.5	Psychrometric diagram of dehumidification process by using solid desiccants	11
2.1	Plot of water content vs. relative humidity of air for various desiccants	15
2.2	Temperature, vapor pressure and concentration relationship for lithium chloride	16
2.3	Psychrometric diagram of a cooling and dehumidification process by using liquid sorbent system	18
2.4	Schematic diagram of liquid sorbent dehumidification/cooling system	20
2.5	Schematic diagram of liquid desiccant system proposed by Turner	26
2.6	Psychrometric diagram of Turner's system	28
2.7	Schematic diagram of hybrid absorption system	29
2.8	Schematic diagram of Hygrol system	31
2.9	Schematic diagram of Twin-Cel system	33
2.10	Schematic diagram of Spray-Cel system	34
2.11	Schematic diagram of a liquid sorbent system	36
3.1	Two resistance concept of mass transfer	40
3.2	"Absorber" used in liquid sorbent system	45
3.3	Control volume of the Absorber	46

List of Figures (cont.)

Figure 3.4	Temperature relationship between water and air in a counterflow cooling tower	55
3.5	Cooling tower performance curve	58
4.1	Humidity ratio distribution of air in the absorber using cooling tower	76
4.2	Sorbent solution temperature distribution in the absorber using cooling tower	77
4.3	Air enthalpy distribution in the absorber using cooling tower	78
4.4	Schematic diagram of liquid sorbent system with an external source of chilled water	80
4.5	Humidity ratio profile of air in the absorber for fixed moisture content of the air at the exit.	87
4.6	Air temperature profile in the absorber for fixed moisture content of the air at the exit	88
4.7	Sorbent solution temperature profile in the absorber for fixed moisture content of the air at the exit	89
4.8	Air enthalpy profile in the absorber for fixed moisture content of the air at the exit	90
4.9	Humidity ratio distribution of air in the absorber using different controlling parameters	94
4.10	Air enthalpy distribution in the absorber using different controlling parameters	95
4.11	Air temperature distribution in the absorber using different controlling parameters	96
4.12	Sorbent solution temperature distribution in the absorber using different controlling parameters	97
4.13	Humidity ratio distribution of air in the absorber using different controlling parameters	98

List of Figures (cont.)

Figure 4.14	Air enthalpy distribution in the absorber using different controlling parameters	99
4.15	Air temperature distribution in the absorber using different controlling parameters	100
4.16	Sorbent solution temperature distribution in the absorber using different controlling parameters	101
4.17	Schematic diagram of a liquid sorbent system with modified configuration	102
4.18	Plot of moisture removal vs. regeneration temperature for different solution con- centrations	111

NOMENCLATURE

Symbol	Defination
A, A'	Nondimensional constants.
a	Transfer surface area/unit volume of absorber
B, B'	Nondimensional constants
C	Specific heat (Btu/lb-°F)
C ₁ -C ₄	Nondimensional parameters
DZ	Unit length of absorber (ft)
F ₁ -F ₈	Nondimensional constants
H	Enthalpy (Btu/lb)
ΔH	Heat of solution (Btu/lb)
h	Nondimensional constant
h _a	Heat transfer coefficient in the gas phase (Btu/hr-ft ² -°F)
h _m	Mass transfer coefficient in the gas phase (lb/hr-ft ²)
k _G	Mass transfer coefficient based on vapor pressure difference (lb/hr-mm Hg)
k _L	Mass transfer coefficient based on solution concentration difference (lb/hr)
m	Mass flow rate (lb/hr-ft ²)
P, p	Partial Pressure (mm Hg)
R	Temperature difference of water across the cooling source (°F)

Symbol	Definition
R	Range of cooling tower ($^{\circ}\text{F}$)
T, t	Temperature ($^{\circ}\text{F}$)
W	Humidity ratio of air (lb/lb)
X	Weight concentration of solution
Z_t	Absorber height (ft)
λ_s	Latent heat of vaporization (Btu/lb)
ϵ	Heat exchanger effectiveness

Subscripts

A	Component A
a	Auxiliary sources of cooling
a	Air
c	Cooling tower
ci	Inlet of the cooling source
co	Outlet of the cooling source
e	Equilibrium conditions
g	Regenerator
ho	Property of water at heater outlet
i	Absorber or regenerator inlet
o	Absorber or regenerator outlet
si	Sorbent's property at absorber or regenerator inlet
so	Sorbent's property at absorber or regenerator outlet
wb	Wet bulb

CHAPTER I

INTRODUCTION

The need to control the amount of moisture in the air is felt in all commercial industrial and institutional applications. In addition to the obvious needs for physiological comfort, uncontrolled moisture, whether too much or too little, can mar production processes and materials.

Removal of moisture from air is termed as dehumidification and has long been used for numerous purposes. Dehumidified air has found extensive use in various industrial applications. In the area of long-term storage and preservation of equipment and materials, since moisture is the most important factor that contributes to material deterioration, both directly and indirectly, most materials can be preserved for a relatively long period in a low humidity environment. In the production of foods and candies, it is necessary to control moisture content of air to maintain products' quality. More moisture in air than required enhances lumping and caking of powdered substances so dehumidified air has been used in pharmaceutical and chemical industries. Humidity control of operating equipment room, in many instances reduces the maintenance requirement of mechanical and electrical systems. Also condensation of water in instrument air lines requires very low humidity air in pneumatic control systems. Dehumidified air has

also found application in the area of grain drying. It has been shown by Rodda [1] and Danziger et al. [2] that it is more effective compared to conventional methods.

The dehumidification of air can be accomplished by three economically feasible and commercially available methods: refrigeration, liquid desiccant, solid desiccant, or a combination of these systems.

1.1 Refrigeration

Refrigeration method is the most widely used and commercially popular method of dehumidification. Systems suitable for air conditioning may be broadly classified as follows.

1. Mechanical compression
2. Absorption.

The cycle of mechanical compression method of refrigeration employs a refrigerant gas which is compressed by a reciprocating, centrifugal or rotary compressor to a higher pressure and corresponding higher temperature. Refrigerant is then allowed to condense by cooling using air or water as cooling medium. Refrigerant is then expanded by aid of an expansion valve and finally heat from the medium to be cooled is extracted in evaporator. Air can be cooled/dehumidified by passing it over evaporating coils. If the dew point temperature is higher than coil surface

temperature air will cool and dehumidify, thus reducing its dew point and dry bulb temperature near the evaporating or cooling coil temperature. A schematic of mechanical refrigeration equipment is shown in Figure 1.1. In this system the air will dehumidify only if the coolant temperature is less than air's dew point temperature and in some instances, to achieve proper moisture control the air is over-cooled to remove excess moisture content and then must be reheated to achieve required dry bulb temperature. Obviously subcooling and reheating results in waste of energy. Subcooling and reheating can be avoided by mixing the return air from the system with just enough dehumidified outside air so as to give required humidity content and then cooling coils would only be required to remove sensible cooling load. Systems requiring 100% outside air, subcooling and reheating becomes inevitable for instances mentioned above. A typical air conditioning process to accomplish cooling and dehumidification is shown on a psychrometric chart in Figure 1.2.

The absorption refrigeration system is a heat operated unit which uses a refrigerant that is alternately liberated and absorbed by the absorbent. If the compressor in a vapor compression system were replaced with a generator-absorber assembly, the result would be a basic absorption system. A schematic diagram of absorption refrigeration system is shown in Figure 1.3.

**THIS BOOK
CONTAINS
NUMEROUS PAGES
WITH DIAGRAMS
THAT ARE CROOKED
COMPARED TO THE
REST OF THE
INFORMATION ON
THE PAGE.**

**THIS IS AS
RECEIVED FROM
CUSTOMER.**

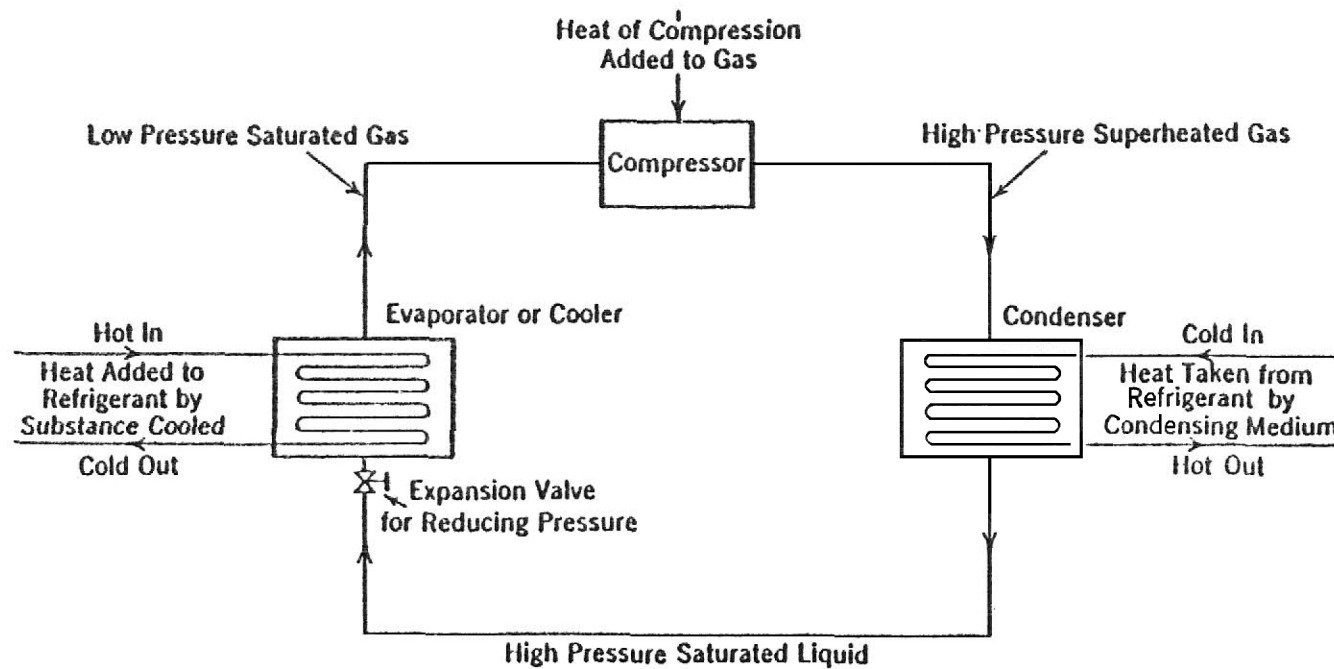


Figure 1.1: Schematic diagram of mechanical refrigeration equipment.

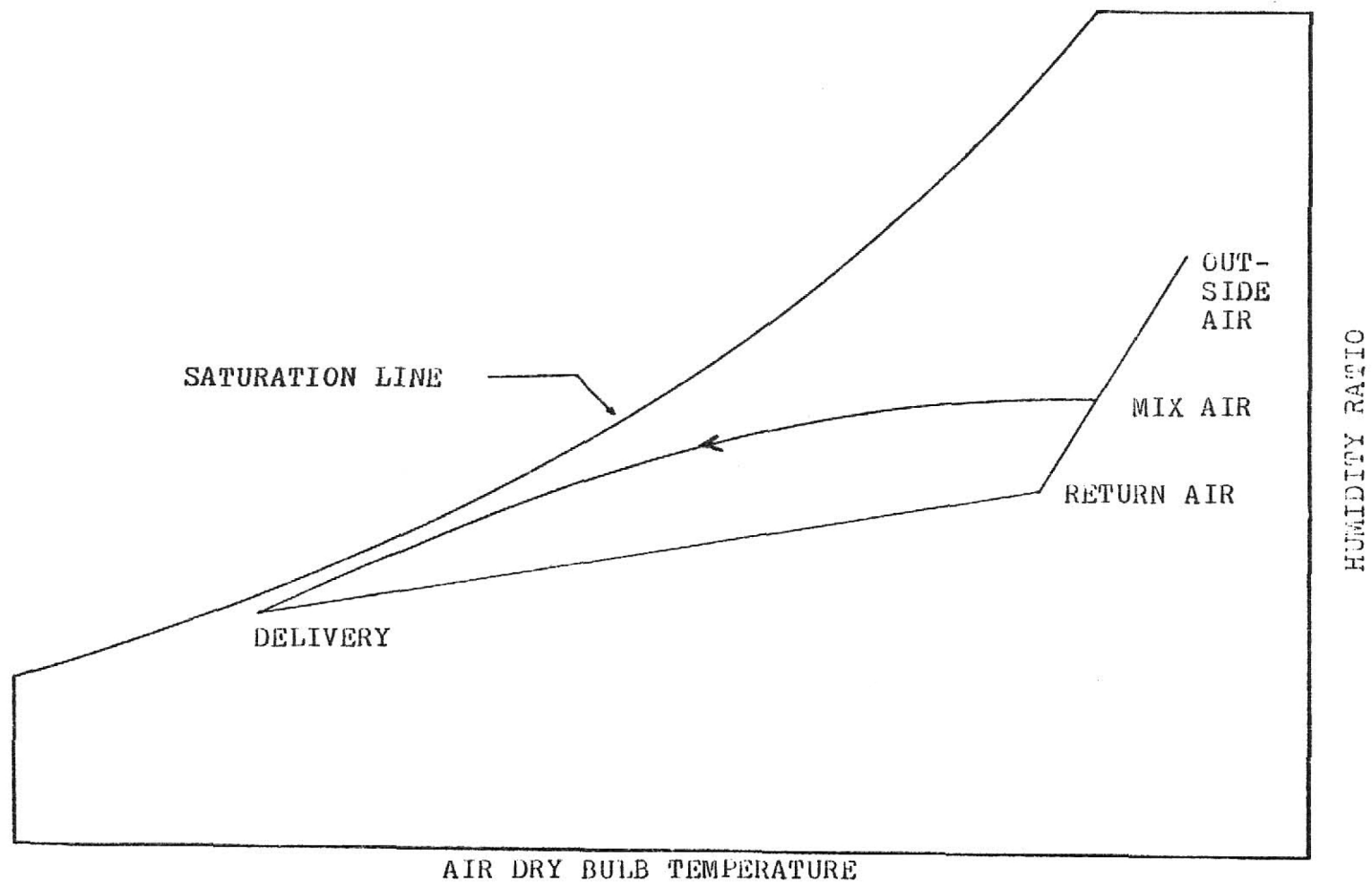


Figure 1.2: Psychrometric diagram of cooling and dehumidification process using a refrigeration system.

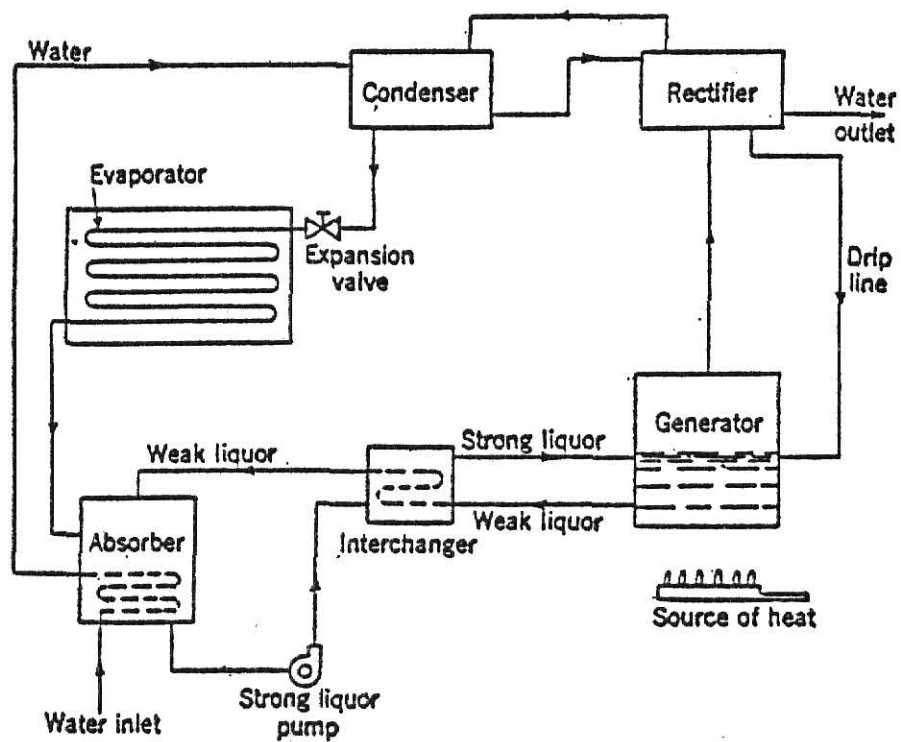


Figure 1.3: Schematic diagram of absorption refrigeration system.

The refrigerant-absorbent pairs which have found extensive commercial use in absorption refrigeration equipments are ammonia-water and water-lithium bromide. In an absorption cycle the refrigerant vapor is pumped from the evaporator pressure to the condenser pressure by first absorbing the vapor in the absorber and then liberating the vapor in the generator. Thermal energy released during absorption is rejected to the cooling coil in the absorber. To liberate the refrigerant from the absorbent thermal energy is added in the generator. From the generator the refrigerant proceeds through the condenser, expansion valve, and evaporator, as it does in the vapor compression system. To improve the efficiency of this system, liquid-to-liquid heat exchanger is employed between the absorber and regenerator to heat the refrigerant-absorbent solution pumped to the generator by the hot absorbent conveyed back to absorber. To improve the efficiency of an ammonia water system, besides the liquid-to-liquid heat exchanger between the absorber and generator, analyzer and rectifier are used to decrease the water concentration of the refrigerant vapor entering the condenser which would otherwise limit the cooling capability in the evaporator. In lithium bromide-water system since water is the refrigerant, evaporator temperature is limited to above 32°F (0°C). Chilled water is obtained from these systems, which is used to cool/dehumidify the air.

1.2 Solid Desiccant System

Solid desiccant dehumidification systems accomplish dehumidification by absorption or adsorption dependent upon the physical or chemical changes of the desiccant. Desiccants used in adsorption process remain solid in contact with air at all operating conditions, while in the absorption process desiccant change from solid to liquid in contact with wet air such desiccants are calcium chloride, phosphorous pentoxide and lithium chloride. Numerous substances may be used for adsorption desiccants including: silica gel, activated alumina, activated bauxites and activated charcoals.

Four configurations exist for solid desiccants dehumidification systems; namely single bed, dual bed, multiple bed and rotary bed units. Fixed bed units require two beds, one is regenerated and other is used for dehumidification for a continuous operation. Rotary bed configuration requires one bed for continuous operation. A portion of the bed is rotated into the regeneration chamber while the rest remains in the operation mode.

As shown in Figure 1.4, the rotary bed unit operates when humid air is brought through the hygroscopic desiccant. The moisture in the air is condensed upon the surface and drawn into the interior of the substance by capillary attraction. The rotary bed may be a disc, a drum or a wheel impregnated with any of the above mentioned adsorbents. It rotates slowly thus always subjecting a part of bed to

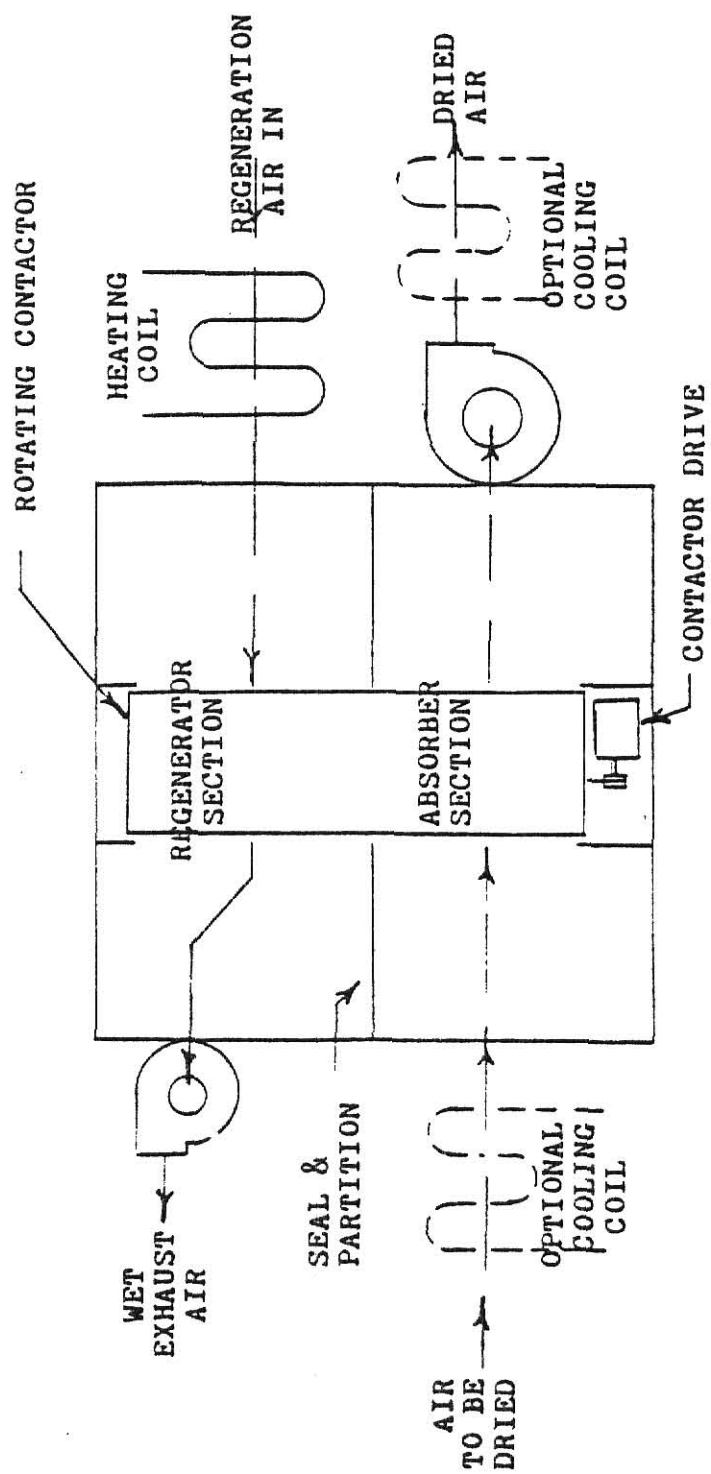


Figure 1.4: A solid desiccant rotary bed unit.

reactivation where a stream of hot air passes through the bed and drives off the accumulated moisture. By continuous rotation a cyclic adsorption and reactivation process keeps the dehumidification a continuous process. A rotary bed dehumidifier using solar energy has been analysed by Pla Barby [3].

In solid desiccant system blower power is required to circulate air both for regeneration and adsorption through relatively thick beds of solid granular material. The adsorption process is exothermic which, in addition to the transformation of latent heat of entering air to sensible heat in the leaving air, tends to further increase the temperature of leaving air. Dehumidification as achieved by a solid desiccant system is shown on a psychrometric chart in Figure 1.5, where the air at state A is heated and dehumidified to state B.

1.3 Liquid Desiccant System

A detailed description of liquid desiccant system is presented in chapter II.

All the systems described above are capable of accomplishing dehumidification. Sometimes it may be more economical to dehumidify air using mechanical refrigeration systems, at other times, dehumidification by solid or liquid desiccant systems with any necessary after cooling by natural sources may be more economical. Even a combination of sorption and mechanical refrigeration systems

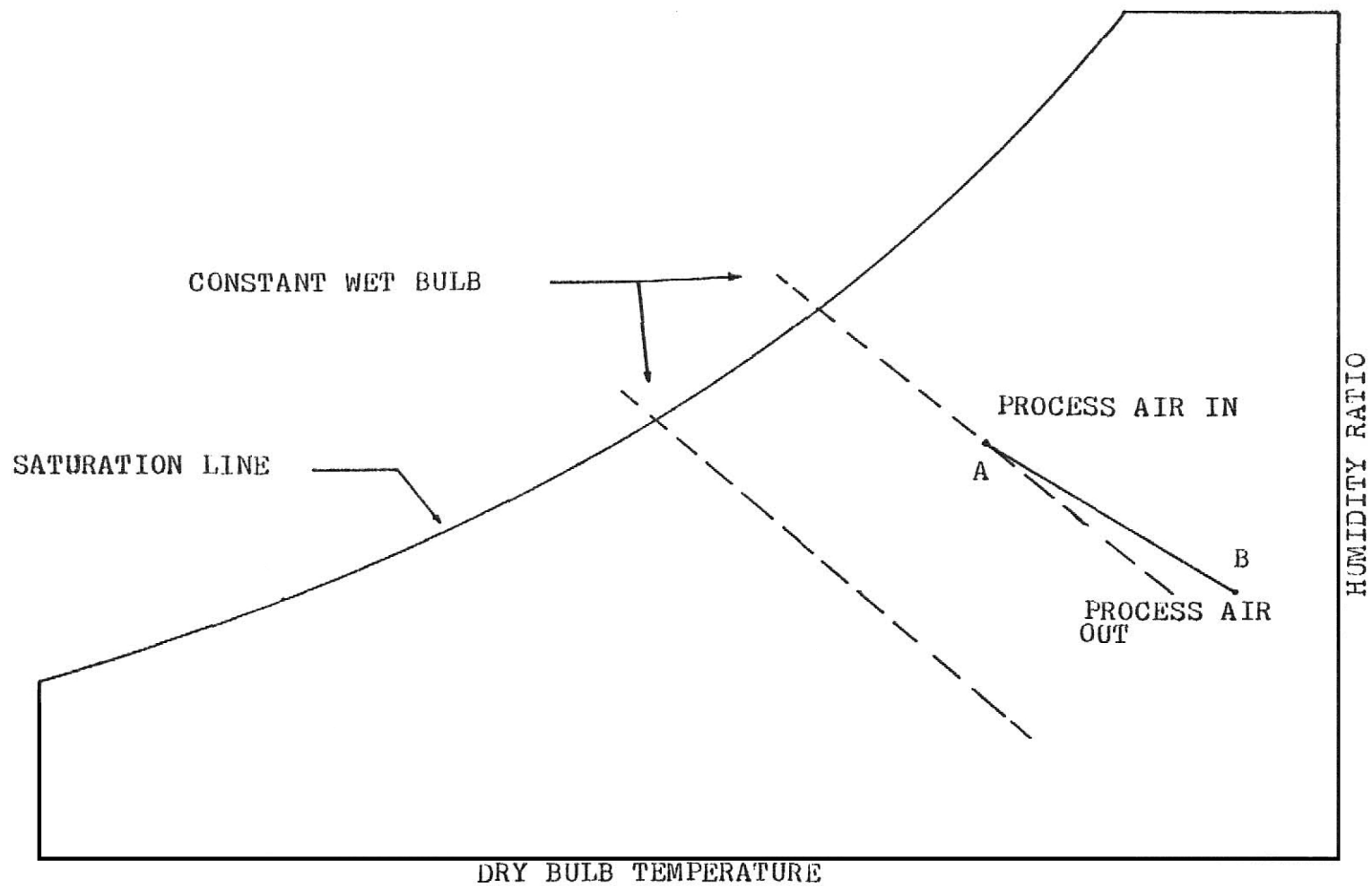


Figure 1.5: Psychrometric diagram of dehumidification process by using solid desiccants.

might turn out to be more economical for certain applications. Dehumidification systems are designed for most severe outside conditions but are seldom subjected to design conditions. Most of the times they operate under far less severe conditions. In order to estimate the annual energy consumption of these systems, before optimization, it is necessary to predict their performance, when they are subjected to partial load.

The objective of this work is to construct a mathematical model to predict the performance of liquid desiccant system when subjected to any air inlet temperature and humidity to accomplish desired air outlet temperature and humidity, to be used for out laying a procedure to determine the optimum system for all season operation.

CHAPTER II

LIQUID DESICCANTS

Cooling/dehumidification of air requires the abstraction of sensible/latent heat from the air. This can be accomplished by exposing the air directly or indirectly over a sufficient area to a medium held at necessary temperature/vapor pressure to produce sensible cooling/dehumidification. There are several basic methods of producing the necessary difference in temperature or vapor pressure between air and the medium employed to achieve cooling or dehumidification, or both simultaneously. Few of these methods were described in the previous chapter.

Few chemical compounds, when brought in direct contact with the air, have the property of removing water vapor from the air. These chemical agents are generally termed as desiccants. These agents may be generally classified as adsorbents and absorbents. Method of dehumidification by using adsorbent or solid desiccants was discussed previously. Liquid desiccants or sorbents are chemical agents which are liquid under ordinary condition and do not undergo any chemical change when water is added to them.

Such compounds are lithium chloride, calcium chloride, ethylene glycols, and various salt mixtures. Most of them can be used as solids. But by continuous adsorption

of water they are successively converted through various hydrates. They finally become saturated liquids. Amount of water taken up before becoming liquid is an individual property of these substances.

Bichowsky [4] has plotted the amount of water versus relative humidity of air in equilibrium for various substances, and is shown in Figure 2.1.

As a rough approximation true for all absorbents relative humidity of air in equilibrium with them is independent of temperature but depends on the nature of the absorbent and on the amount of water it has taken, or on concentration when it is liquid.

The choice of correct sorbent for dehumidification application requires the balancing of several factors against each other. These factors are: engineering suitability, which includes heat and mass transfer characteristics and viscosity; chemical and physiological suitability, which includes chemical stability, toxicity, corrosiveness, odor and flammability; thermodynamic suitability, which includes vapor pressure, concentration, and phase relationship characteristic. The temperature, vapor pressure and concentration relationship for lithium chloride is shown in Figure 2.2.

Another aspect necessary to be considered when a liquid sorbent is selected is crystallization, which happens when a particular concentration reaches a temperature at

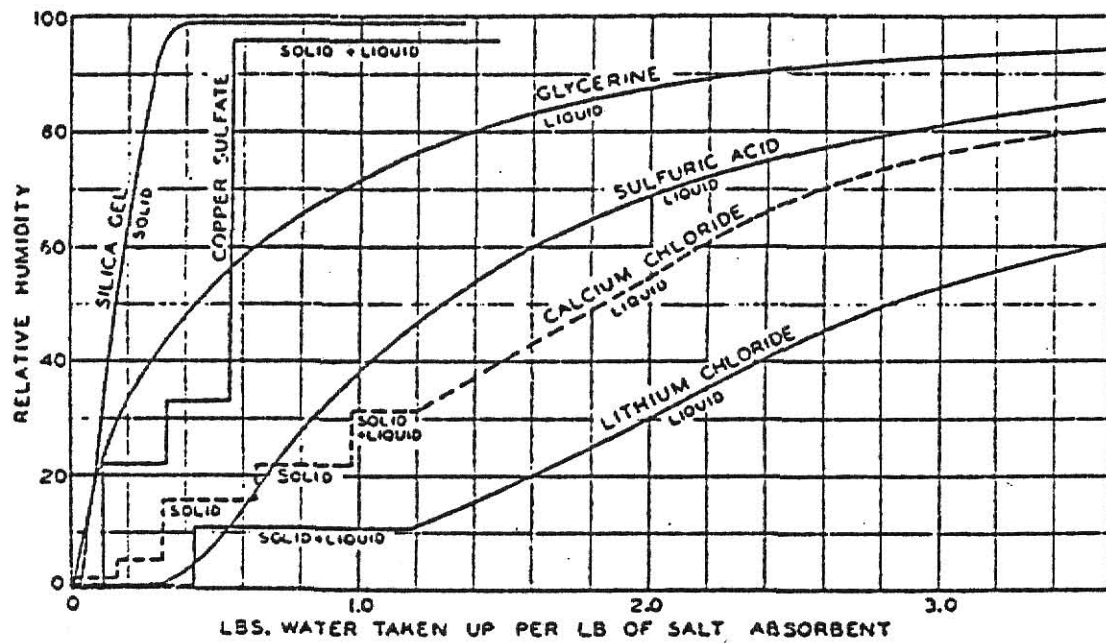


Figure 2.1: Plot of water content vs. relative humidity of air for various desiccants.

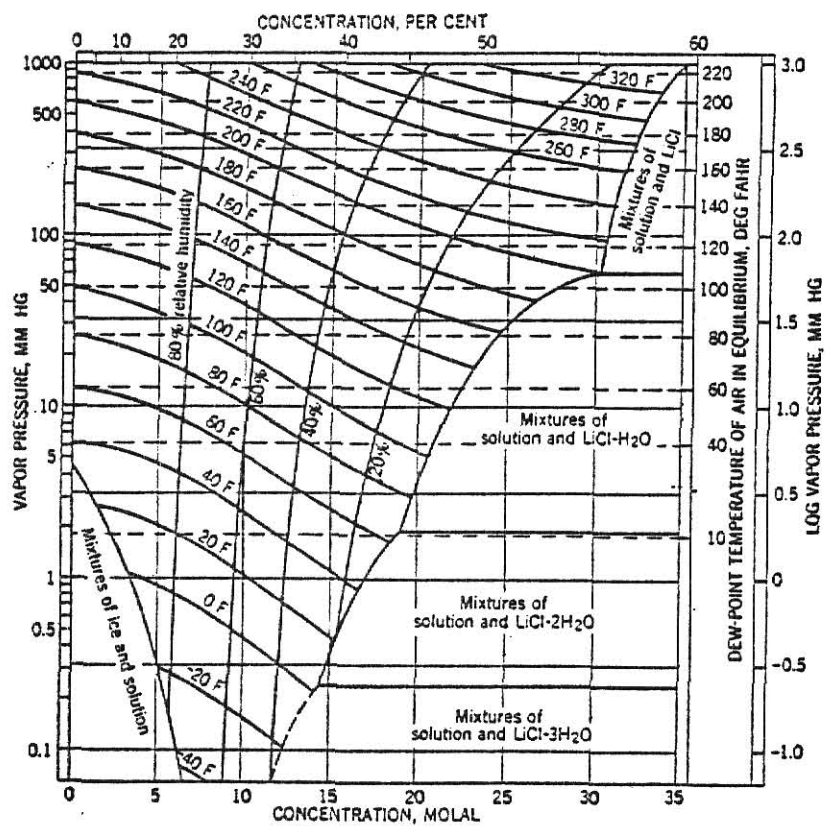


Figure 2.2: Temperature, vapor pressure and concentration relationship for lithium chloride.

which it becomes a solid crystalline mass of hydrates. Gifford [5] suggested that minimum temperature of solution in an operating cycle should be 5 to 8 degree Celcius above the crystallization temperature.

2.1 Dehumidification by Liquid Sorbents

When air is brought in contact with a liquid sorbent, the vapor pressure of air in equilibrium which depends on sorbent's temperature if less than the vapor pressure of bulk air provides a potential for mass transfer. Water molecules start migrating from bulk air to sorbent solution, thus increasing the temperature and decreasing the concentration of liquid sorbent. This operation can continue until no potential for mass transfer is present.

All liquid sorbent dehumidification/air-conditioning equipment utilizes this principle of chemical absorption.

Liquid desiccant air-conditioning/dehumidification systems are available commercially for some time. Lithium and calcium chloride brines and glycol solutions are used in equipments commercially available today. Cooling and dehumidification as accomplished by using liquid sorbents is shown on a psychrometric chart in Figure 2.3.

As it shows air at condition A is cooled and dehumidified to condition D. This can be accomplished by first sensibly cooling the air from condition A to condition E and then dehumidifying to condition D by using liquid

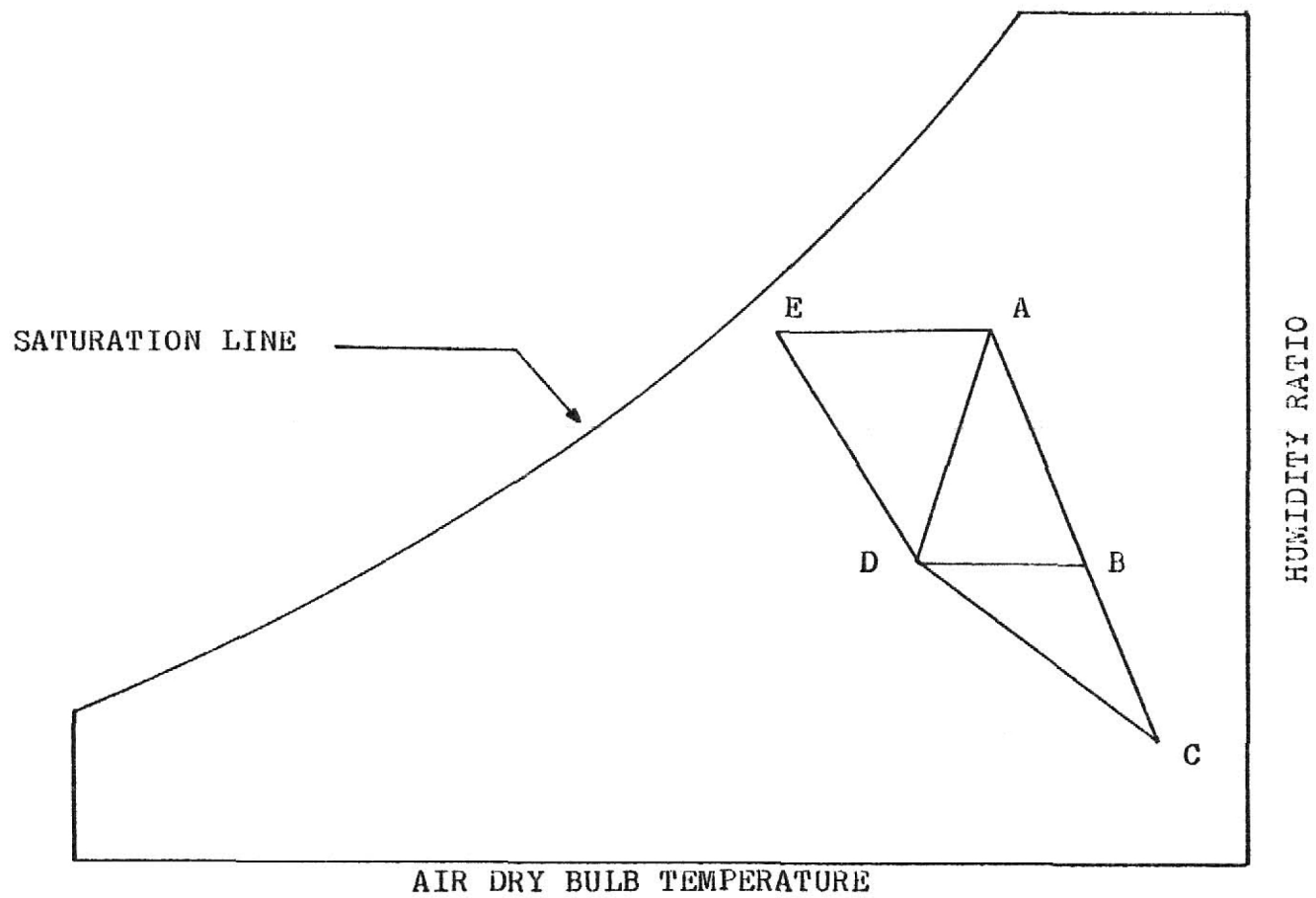


Figure 2.3: Psychrometric diagram of cooling and dehumidification process by using liquid sorbent system.

sorbent. It can also be accomplished by dehumidifying using liquid sorbent, to condition B, and then sensibly cooling it to condition D. It is necessary to mention here that to have an efficient operation, external cooling source is required to sensibly cool the air. External cooling source, depending on requirement, can be a cooling tower, a natural source or refrigeration system.

If an external cooling source is not available then using liquid sorbent air can be dehumidified to condition C and then cooled and humidified by evaporative cooling. Dehumidification to point C requires more concentrated solution which requires larger transfer surface area. Also, the moisture absorbed in the solution, for a continuous operation, has to be removed from the solution. Removal of more moisture requires the solution at higher temperature.

2.2 Liquid Sorbent Dehumidification Systems

The schematic diagram of a typical liquid sorbent dehumidification/cooling system which is shown in Figure 2.4, operates by drawing air from the bottom of the packed tower. Liquid sorbent is sprayed at the top of tower and flows in the downward direction. Counter current flow between the liquid sorbent and air ensures an efficient operation. Moisture is absorbed from the air by the solution due to the vapor pressure difference between air and the solution. The heat of absorption in this process, which

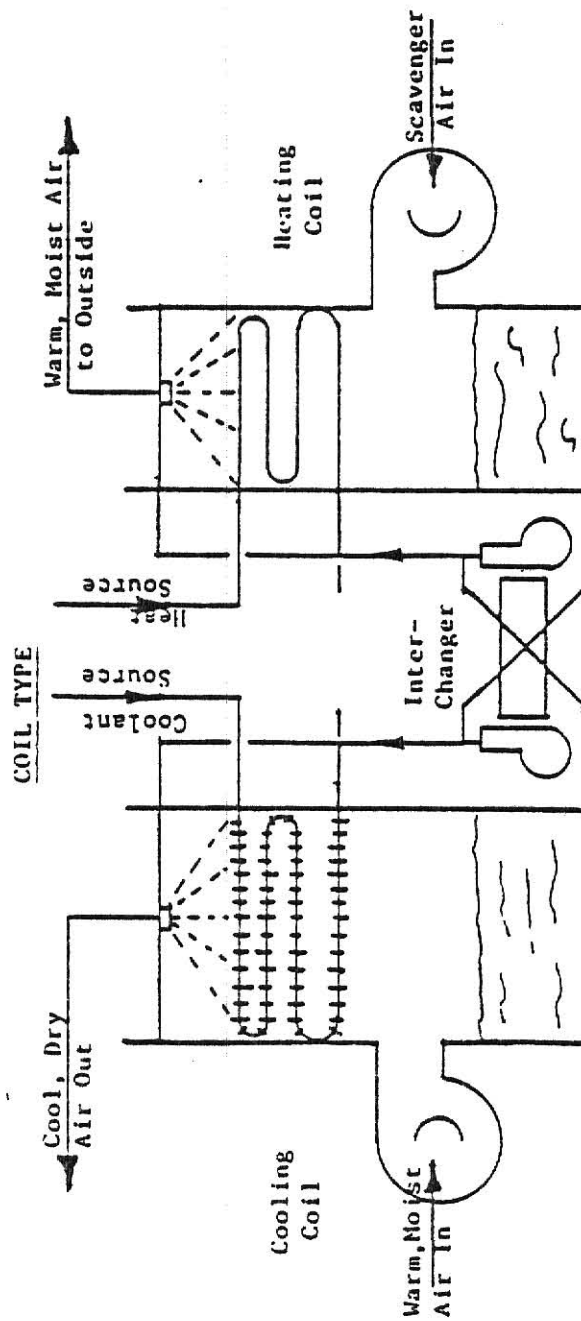


Figure 2.4: Schematic diagram of liquid sorbent dehumidification/cooling system.

includes the latent heat of water vapor during condensation and absorption and the heat of mixing raises the temperature of liquid sorbent solution. Dehumidification of air thus, by rejecting moisture to the solution decreases the concentration of sorbent solution.

The dew point temperature of air in equilibrium with sorbent solution increases with an increase in solution temperature. So it is necessary for a more efficient dehumidification operation to maintain the solution at a relatively low temperature. Cooling of sorbent solution is accomplished by either an external cooler which is an ordinary heat exchanger, or by a finned cooling coil imbedded in the dehumidification equipment with packing. The cooling coil mentioned later in some cases can be a finned coil such that besides cooling the solution, the fin surface can serve as a air-to-solution contact surface. The dew point temperature of air in equilibrium with sorbent solution also increases with a decrease in solution concentration, which makes it necessary that excess water of condensation, which dilutes the solution is removed. This is accomplished in regenerator. To ensure continuous operation a small portion of dilute solution is heated in a heater which is an ordinary heat exchanger using low pressure steam or hot water. This hot solution is sprayed over a stream of regeneration air which is outside air or building/process exhaust air

in regenerator. Regenerator is similar in construction as dehumidifier equipment. Air in equilibrium with hot solution has vapor pressure higher than the vapor pressure of bulk air which initiate transfer of moisture from solution to air stream. Concentrated solution is pumped back through another heat exchanger to heat the incoming dilute solution.

Griffiths [6] has reported that regeneration requirement is generally in the range of 1.5 to 2.5 Btu per Btu (KJ per KJ) of latent load taken care in dehumidification equipment. Sorbent concentration is commonly maintained at a fixed value by modulating regenerator heat input in response to brine operating level in the dehumidification unit. Regeneration heat source temperature hence, depends on the latent load taken care in the dehumidification unit. Griffiths [6] has also mentioned that a heat source temperature of 155 to 175°F (68 to 79°C) is required to produce air at 20% relative humidity which falls to 120 to 140°F (49 to 60°C) to produce air at 50% relative humidity. Nothing has been mentioned about the inlet and exit conditions of this air.

It would be more convenient if heat source temperature variation with amount of moisture removed from the air is known.

For the range of temperatures required for regeneration a wide range of alternative heat sources can be used. These sources in industrial applications can be

tailsteam from turbine drives, hot condensate and flash steam, process waste heat and recovered heat from oven and dryer exhaust gases. In commercial and comfort applications, heat from muffler and jacket of diesel engines, condenser heat from refrigeration equipment and high temperature heat pumps can be a few alternatives.

Solar energy can also be used for regeneration in liquid desiccant cooling/dehumidification systems. Especially in comfort applications where solar energy is in phase with cooling requirement, it can be used very efficiently to regenerate liquid sorbents.

Utilization of solar energy for refrigeration and air conditioning purposes has been a topic of extensive research. Cooling with solar energy using a liquid desiccant system was first suggested and experimentally tested by Löf [7] in 1954. His system was a conventional liquid desiccant system. Solar energy was used to regenerate the liquid sorbent. It was shown that if liquid sorbent can be heated to 140°F (60°C) could be effectively regenerated by using heated air. The average cooling capacity by using 67 m^2 of 50% efficient solar collectors was reported to be about 4 tons. It was also shown that for round the year heating and cooling performance this system was economically competitive with a conventional heating and cooling system at that time. Kapur [8] has found that in humid regions application of solar energy to a liquid desiccant system has

shown promising prospects. Since the amount of water removed from a liquid sorbent is a measure of dehumidification, that can be done by using this sorbent, only regeneration of liquid sorbents has been a topic of study for many researchers. Regeneration of calcium chloride-water solution has been studied and analyzed by Mullick and Gupta [9]. Kakabaev and Khandurdyyer [10], Kakabaev, et al., [11] and Kakabaev and Golaev [12] have studied regeneration of liquid sorbent. They considered the cases of an open collector, closed collector with air circulation, and closed collector like a solar still, and developed a model to predict the performance of each system. Collier [13] has further analyzed and simulated the performance of an open cycle absorption refrigeration system for five cities in the United States by using actual weather data. The open cycle differs from the closed cycle in that the open cycle regenerates the weak absorbent solution by evaporating water to the atmosphere rather than to a condenser. The solar collector used for the open cycle in his work is the one in which the weak sorbent solution flows as a fluid film over a flat, open, black surface. The sorbent solution is heated by insolation on the black surface and is regenerated by evaporating water to the atmosphere.

The dew point temperature of air in equilibrium with a liquid sorbent is directly proportional to the temperature of the solution and inversely proportional to it's concentration. To cool and dehumidify the air using liquid sorbent

it is necessary to cool the solution for an efficient operation. The sorbent temperature is governed by the temperature of available cooling water. Robinson [14] has analyzed a liquid sorbent system using triethylene glycol solution. Regeneration is accomplished by solar energy, when the dilute solution is allowed to flow down a tilted open corrugated trickle collector. This system uses ample well water at 64°F (17.8°C) available from the South Carolina coastal area to cool the solution and thus remove the sensible load as well as enable the solution to absorb the latent load. Obviously this system can not be used where well water is not available. Turner [15] used the concept of using part of dehumidified air to cool water and in turn use this water to cool the solution. Turner's system operates by bringing the mixture of outside air and recirculated air in contact with a liquid sorbent which is triethylene glycol-water solution in the absorber unit. The air is cooled and dehumidified in this unit, a portion of this processed air is then taken to the evaporator where it cools the water through evaporative cooling. The triethylene solution is regenerated to 140°F (60°C) by using solar energy. The cold water generated in the evaporator is used to maintain a relatively low temperature of the liquid sorbent. In this system triethylene glycol solution is fed at two points to the absorber. All the flow streams are counter flow in this system. This system is shown schematically in Figure 2.5. Cooling and dehumidification process

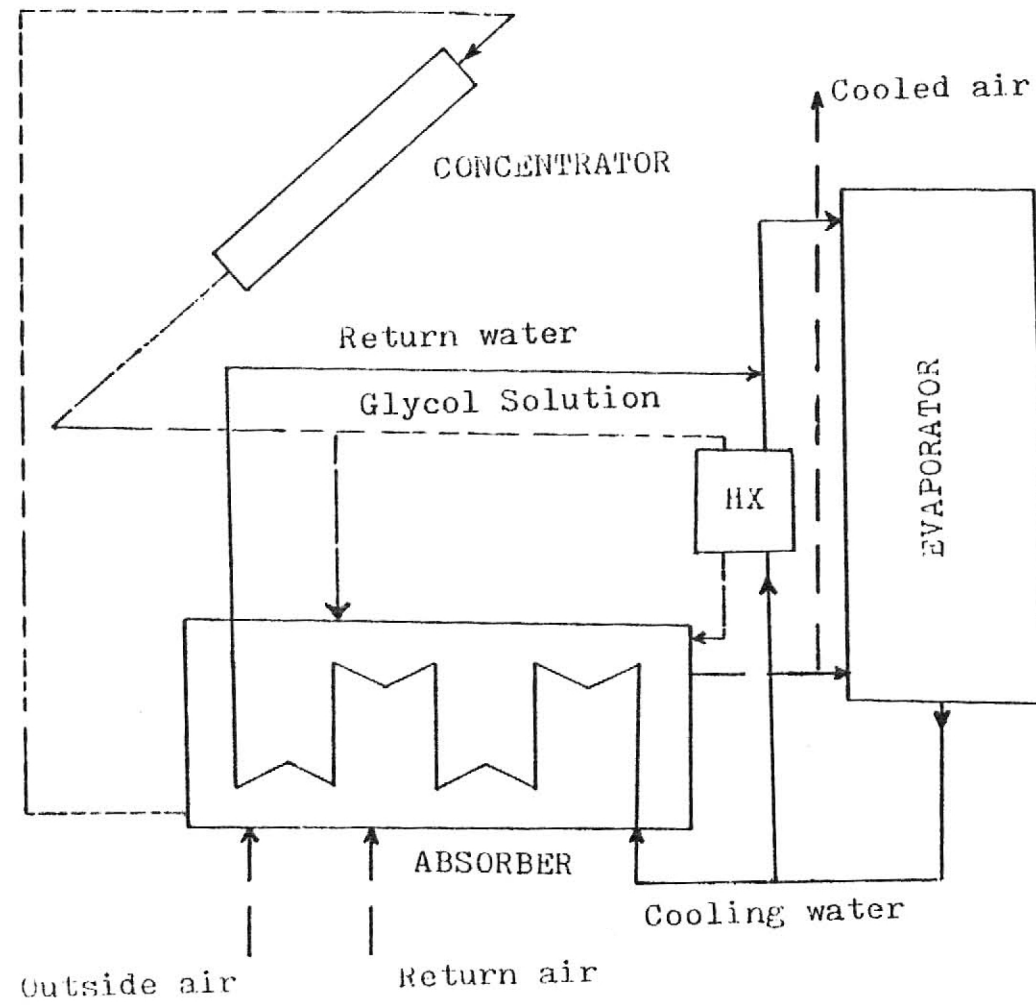


Figure 2.5: Schematic diagram of liquid desiccant system proposed by Turner.

and the corresponding air states in this system are shown on a psychrometric diagram in Figure 2.6.

Peng [16] has suggested that instead of using processed air as in Turner's system, exhaust air can be used in the evaporator. For air conditioning applications he has claimed that this modification will reduce the size, operating and capital cost of Turner's system. Another system called Hybrid Absorption system is proposed and analyzed by Peng [15]. This system uses lithium chloride-water solution to cool and dehumidify the air in a liquid sorbent system. The same solution is used in an absorption system thus generating cold water to keep down the temperature of liquid sorbent in the open absorber. A schematic diagram of this hybrid absorption system is shown in Figure 2.7.

Technology of liquid desiccant systems is in existence for a long time. These systems are commercially available today in the market. Two of the major manufacturers in United States marketing liquid sorbent systems which can be integrated with alternative energy sources are Niagara Blower Company and Midland-Ross Corporation. The systems manufactured by them are described in the following.

2.3 Niagara Blower Company's "Hygrol" System

The schematic diagram of this system is shown in

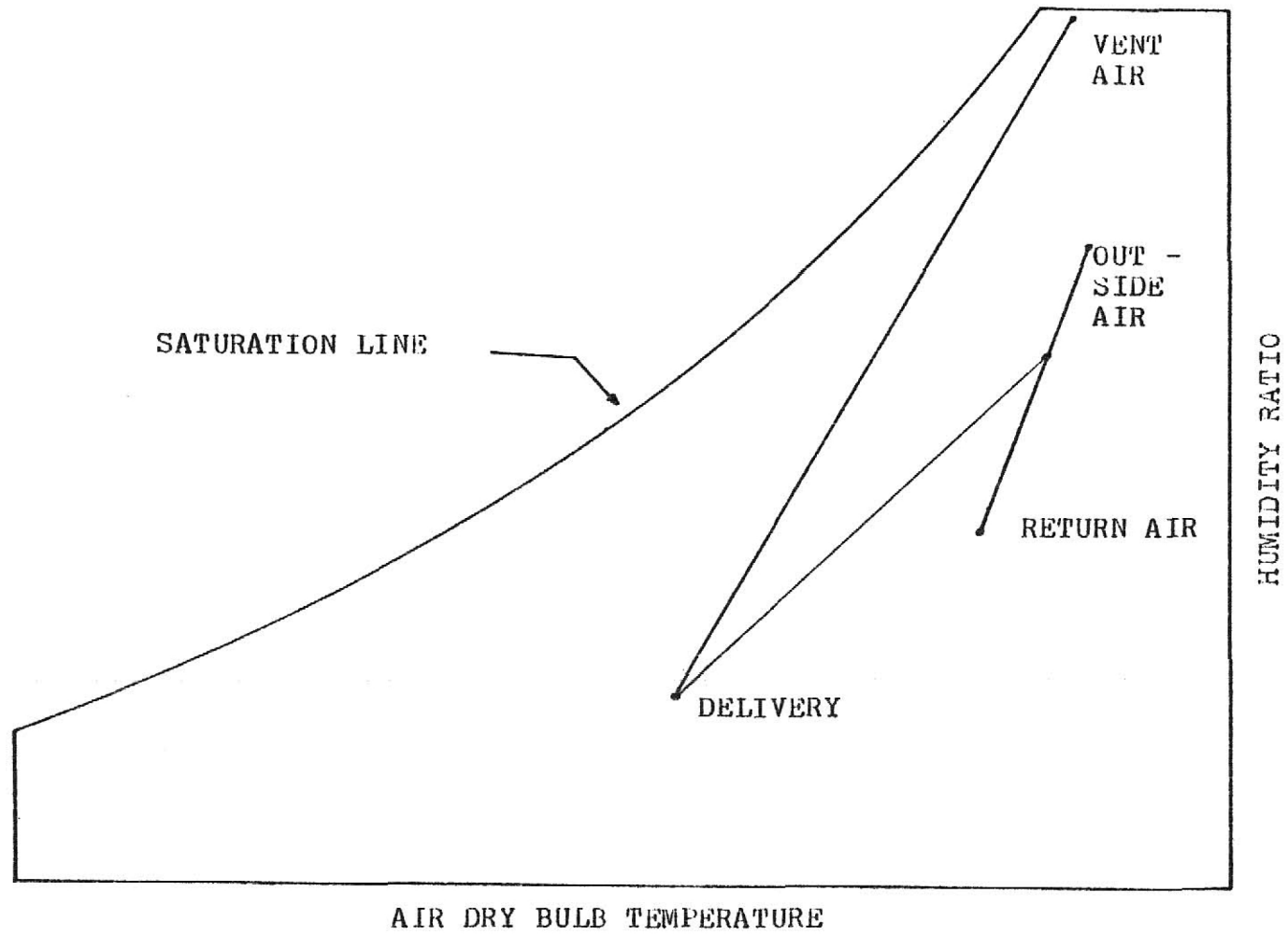


Figure 2.6: Psychometric diagram of Turner's system

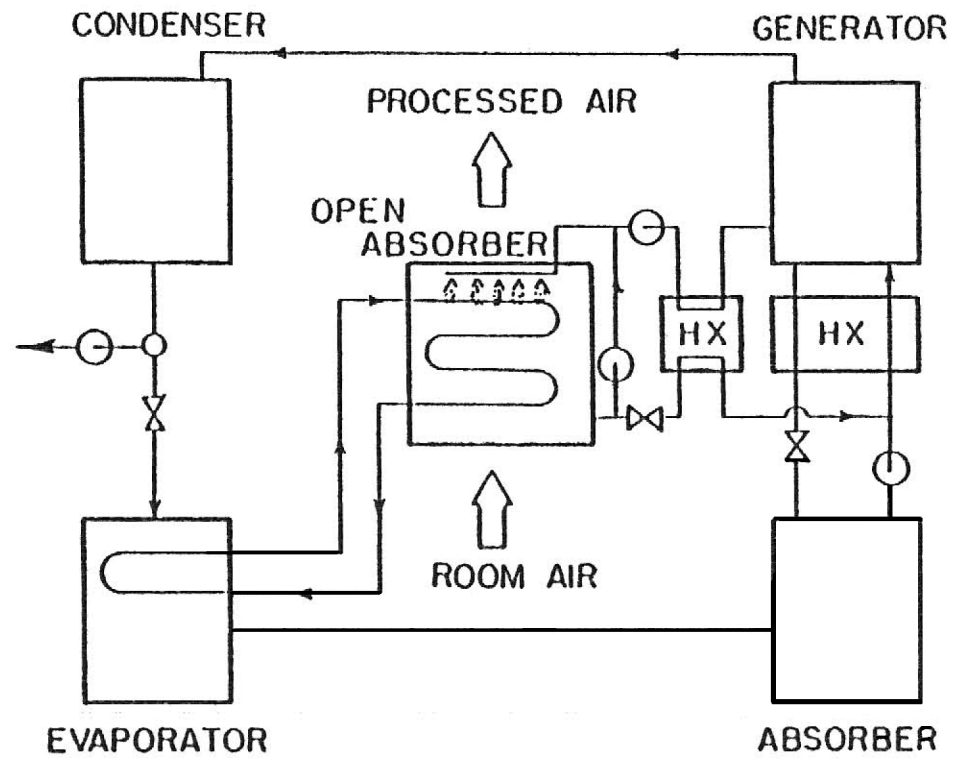


Figure 2.7: Schematic diagram of hybrid absorption system.

Figure 2.8. The system consists of two units, namely concentrator unit and conditioner unit separated by a heat exchanger. The system uses triethylene glycol solution, "Hygrol" as the sorbent. The conditioner unit is equipped with two cooling coils served by an external source of cold water. Outside air when comes into contact with concentrated triethylene solution loses moisture. This process tends to raise the temperature of the solution which is prevented by the cooling coil. Sensible cooling of the air is also furnished by this cooling coil. Air then passes through an eliminator where solution entrained in the air is trapped. Before the air is supplied to the conditioned space it can be further cooled, if required by the other cooling coil. Weak solution from the conditioner is pumped, through the heat exchanger cooling the hot concentrated solution going to the conditioner, to the concentrator. The concentrator unit is also equipped with two coils. Weak solution is sprayed into an air stream, with a heating coil to aid evaporation of the absorbed water. Air then passes through an eliminator which traps any entrained solution in the air stream. A cooling coil is then used to condense any absorbent that has evaporated along with the water vapor into the air. Further recovery of sorbent is achieved when air enters reclaimer which is a large surface area fiberglass cylinder. The surface tension difference between water and "Hygrol" causes Hygrol to stick to the cylinder surface, finally flowing down to

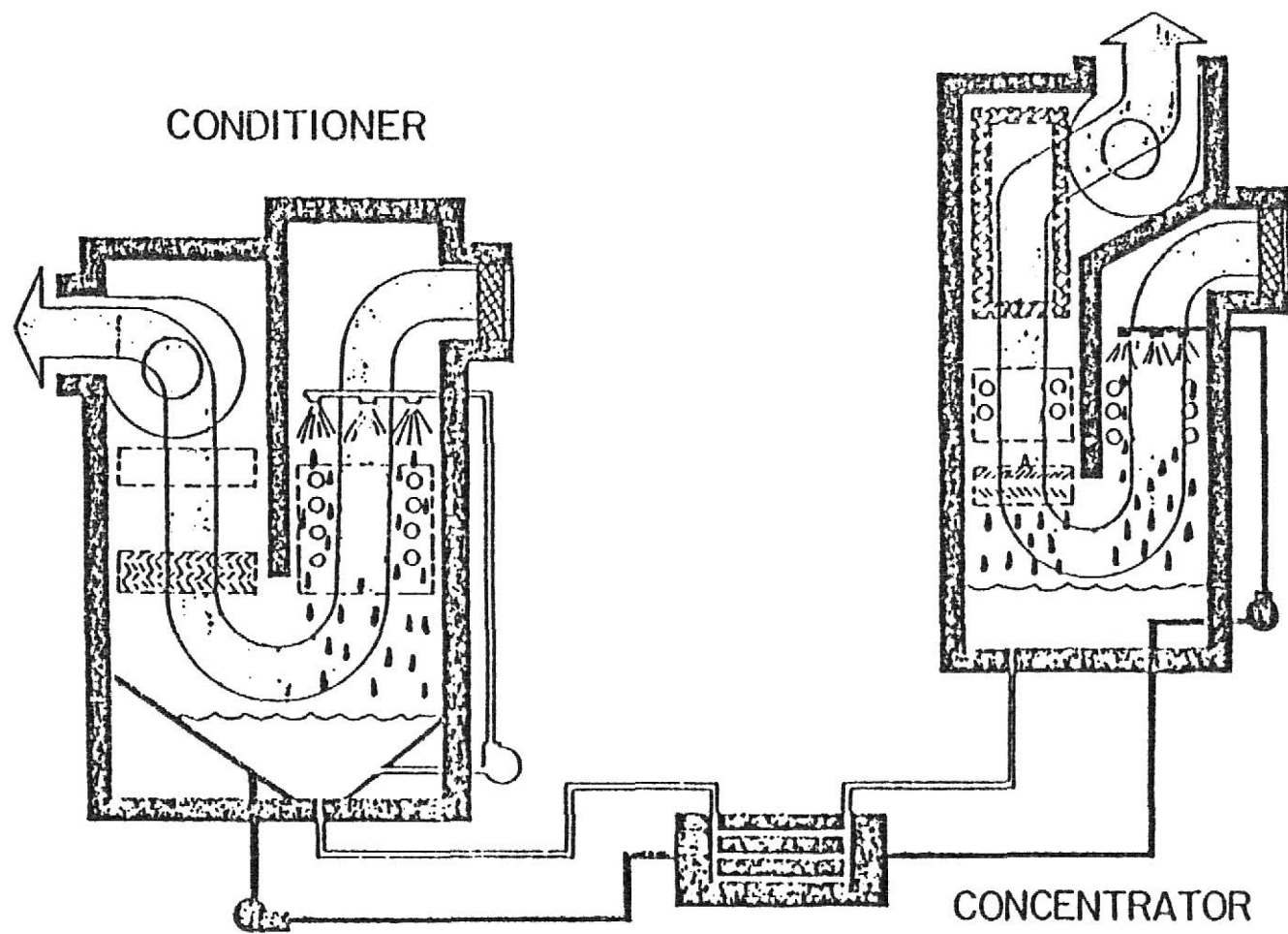


Figure 2.8: Schematic diagram of Hygrol system.

the sump. The air is exhausted to the outdoors. This system has been used by Lodwig [17] and Robinson [14] in their studies.

2.4 Midland-Ross Corporation's Kathabar System

Two systems are marketed by Midland-Ross Corporation. They are Kathabar Twin-Cel and Kathabar Spray-Cel systems.

The Twin-Cel system is shown in Figure 2.9. It is an air-to-air enthalpy recovery system. The system operates under the same principle as described for other liquid sorbent systems earlier. The only difference is that it uses the return air from the conditioned space for regeneration thus, recovering its enthalpy. Sensible cooling of the supply air is accomplished outside the absorber unit by a cooling coil. The manufacturer claims that this system recovers about 65% of the heating and cooling effect present in the return air from the conditioned space.

The other system marketed is called Spary-Cel and is shown in Figure 2.10. In this system both the absorber and regenerator unit are equipped with coils. Solution is sprayed by nozzles in the absorber and regenerator units. Air is cooled and dehumidified in the absorber while solution is regenerated by means of a heating coil in the regenerator.

The liquid sorbent employed in both these systems is "Kathene" which is a solution of water and lithium

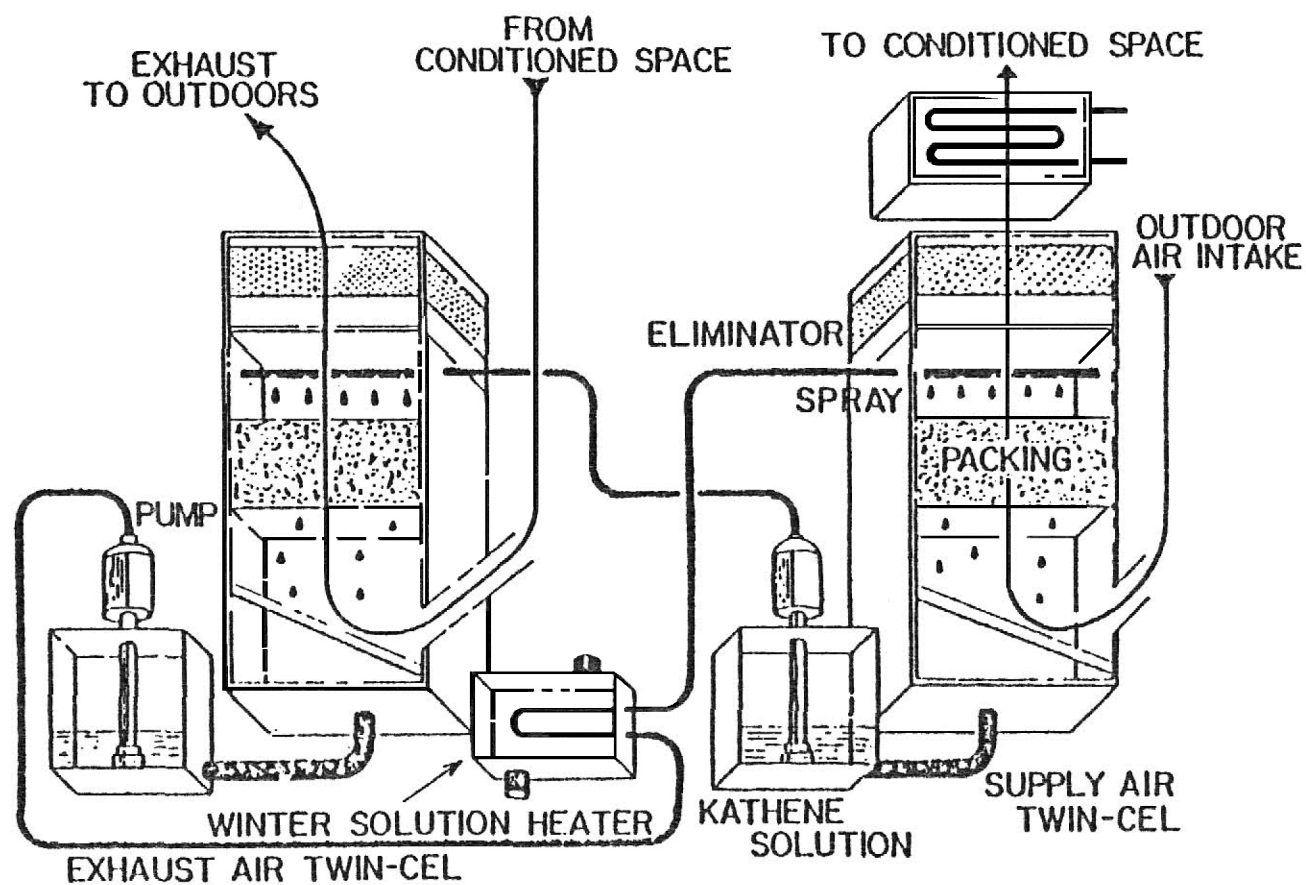


Figure 2.9: Schematic diagram of Twin-cel system.

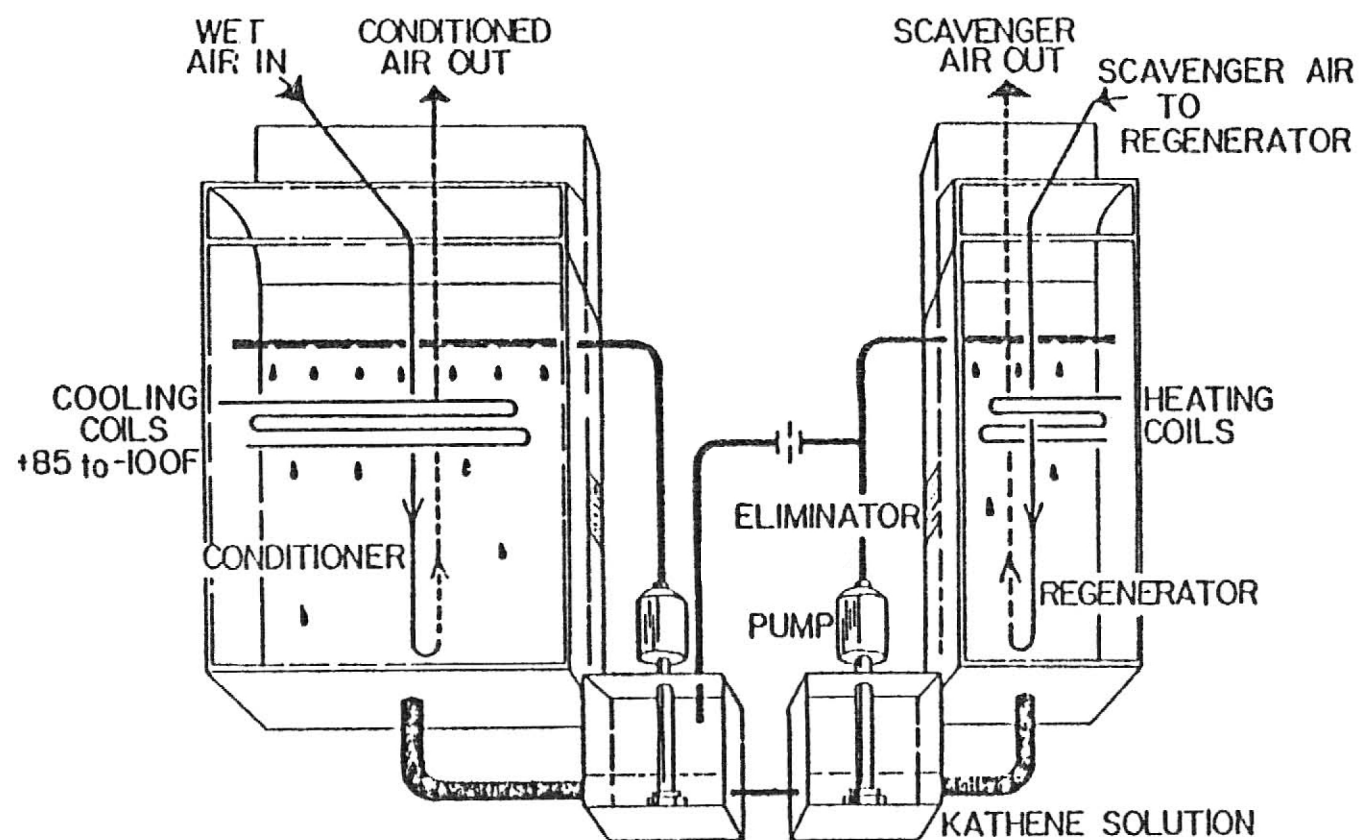


Figure 2.10: Schematic diagram of Spray-Cel system.

chloride salt. Performance of liquid sorbent systems described above when subjected to partial load is not known.

Griffiths [6] has described a liquid sorbent system which is shown in Figure 2.11. This system is called the "Packed type" liquid sorbent air conditioning system. This system differs from the commercially available systems only in that the "packed type" uses an external cooler to cool the sorbent and an external heater to heat the sorbent solution for regeneration purposes. It is this system which is subjected to detailed study in this work. In order to predict the performance of this system under partial load, it is necessary to model the heat and mass transfer behavior for individual components of this system. The next chapter is devoted to describe the necessary theoretical background, which will serve to model the individual components of this system later in that chapter.

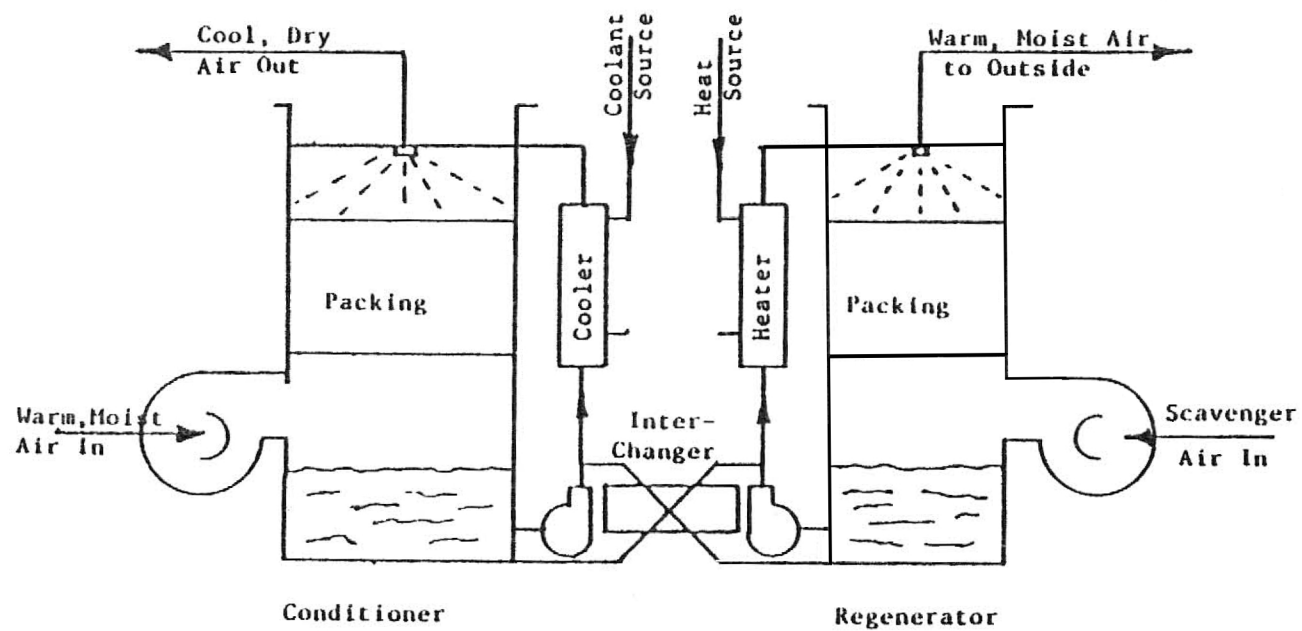


Figure 2.11: Schematic diagram of a liquid sorbent system.

CHAPTER III

MODELING OF COMPONENTS

The packed type liquid sorbent system which is modeled for part load performance analysis is shown in Figure 2.11. The major components of the system are the absorber, the regenerator, the cooling tower and the cooler. Lithium chloride water solution is used as the sorbent in this system. The model developed, however, is general in nature and can use any sorbent solution by furnishing the needed information required for calculating the physical properties of the solution. Before modeling the individual components of this system, it is necessary to describe interphase mass transfer with simultaneous heat transfer which occurs in absorber and regenerator.

Heat transfer requires a temperature potential, similarly it is essential to have a difference in the concentrations of the component in order that its mass may diffuse from a region of higher concentration to a region where the concentration is lower. This transport phenomenon is termed as mass transfer. Considering the kinetic aspect, mass transfer mechanism can be either diffusion or convection in nature. Pure diffusion is a molecular phenomenon like the conduction of heat and occurs in stagnant layers. Diffusion of molecules is more intensive at high temperatures and low pressures, furthermore light molecules diffuse more

rapidly than heavy molecules. Mass exchange by convection occurs when a fluid is artificially agitated or mixed by currents or eddies as in turbulent flow. This carries the transferred component from place with higher concentration to layers with lower concentration. Actual mass transfer phenomenon, however, does not usually take place as described above. Existence of interfacial surface renders the phenomenon more complicated as convection ceases in this layer.

Interphase mass transfer takes place between two phases, mass is transferred in one phase to the interphase surface, and then from the interphase to the bulk of another phase. Interphase surface or interface is the surface separating two phases. Such a surface might be, for example, a surface of liquid, capable of absorbing some component from the gas flowing over it, which further, penetrates from the interface into the bulk of liquid. In both the phases mass transferred either by diffusion or convection transport mechanism depends on the concentration gradient of the diffusing component.

Hence, mass transfer between two phases require a departure from equilibrium between the bulk concentrations within each phase because once equilibrium is reached within the phase there will be no mass transfer gradient and further diffusion will cease.

Liquid-gas interphase mass transfer process is analogous to heat transfer between gas and liquid phase. Same

mass flux exists on both sides of the interface in the mass transfer process, where as in the heat transfer process, though the heat transfer coefficients are different for each phase but heat flux on both sides of the interface is same. Interphase mass transfer processes involving simultaneous heat transfer are however more complex in nature to analyze. Latent heat of phase change and heat of solution in these processes make the evaluation of properties at the interface quite difficult. To get around an interphase mass transfer problem Whitman [18] suggested a two film theory. This theory assumes that there is no resistance to solute transfer across the interface separating the phases and that the only diffusional resistances are those residing in the fluids themselves. The reliability of this theory has been subjected to many tests. As a result, it is now known that there are few special situations where it is invalid. In ordinary industrial application situations, the theory holds good and is used to explain the process of interphase mass transfer.

The two film concept of mass transfer is shown in Figure 3.1. The equations of mass transfer for component A in a steady state, one dimensional mass transfer problem between a gas and liquid for each phase can be written as.

$$N_A = k_G (P_{AG} - P_{Ai}) \quad (3.1)$$

for the gas phase; and

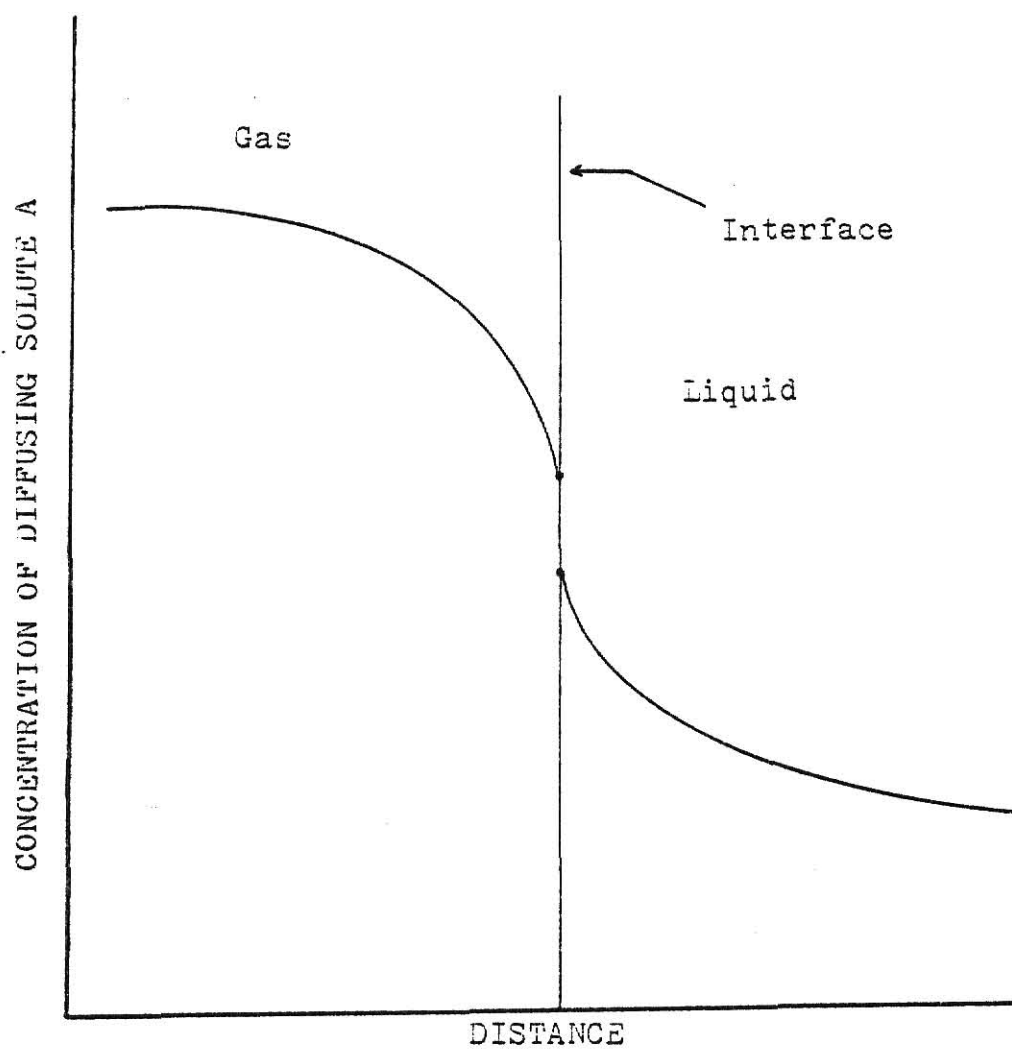


Figure 3.1: Two resistance concept of mass transfer.

$$N_A = k_L (C_{Ai} - C_{AL}) \quad (3.2)$$

for the liquid phase.

k_G and k_L are convective mass transfer coefficients in the gas and liquid phases, respectively. N_A is the number of moles transferred per unit time per unit area. $P_{AG} - P_{Ai}$, is the mass transfer potential available in the gas phase. P_{AG} is the partial pressure of component A in the bulk gas, and P_{Ai} is the partial pressure of component A at the interface between gas and liquid. Similarly $C_{Ai} - C_{AL}$ is the driving force necessary to continue the transfer of component A in the liquid phase and C is concentration of component A in the liquid phase. Since same mass flux exists on both sides of the interface, ratio of transfer coefficients can be obtained from

$$\frac{k_L}{k_G} = - \frac{P_{AG} - P_{Ai}}{C_{AL} - C_{Ai}} \quad (3.3)$$

The above relationship can be used to determine the rate of mass transfer if bulk and interface concentrations of the solute in both the fluids are known. Experimental determination of interfacial properties is usually impossible. Under these circumstances, only an overall effect, in terms of the bulk concentrations can be determined. Since the equilibrium distribution curve for the system is unique at fixed temperature and pressure, so the following relationships are used to define the overall transfer coefficients.

$$N_A = K_G (P_{AG} - P_{Ae}) \quad (3.4)$$

$$N_A = K_L (C_{Ae} - C_{AL}) \quad (3.5)$$

where P_{Ae} is the partial pressure of component A in equilibrium with bulk liquid of concentration C_{AL} , and C_{Ae} is the concentration of component A in equilibrium with P_{AG} in the gas phase. K_G and K_L are overall mass transfer coefficients in terms of gas partial pressure driving force and liquid concentration driving force, respectively. When $P_{AG} - P_{Ae}$ remains proportional to $C_{Ae} - C_{AL}$ as pressure and concentration are varied, Henry's law can be used to relate gas phase concentration through a proportionality constant h .

$$C_{Ae} = hP_{AG} \quad (3.6)$$

$$C_{AL} = hP_{Ae} \quad (3.7)$$

The relation between the overall and the individual mass transfer coefficients can be obtained by eliminating C_{Ai} and P_{Ai} from Equations (3.1) and (3.2) using Equations (3.6) and (3.7).

$$\frac{1}{K_G} = \frac{1}{k_G} + \frac{1}{h k_L} \quad (3.8)$$

$$\frac{1}{K_L} = \frac{1}{k_L} + \frac{h}{k_G} \quad (3.9)$$

The reciprocal of the overall coefficient is the overall resistance to mass transfer, which is the sum of individual film resistances, represented by the terms on right hand side of the above equations. It is evident from Equation (3.8) and (3.9) that as h is made large, the gas film resistance approaches the overall resistance, and k_G approaches K_G . Similarly, if h is made very small the liquid film resistance and overall resistance become practically equal. Since a large value of h corresponds to high solubility of a component in the solution such as water vapor absorbed in a lithium chloride water solution liquid phase resistance can be disregarded. Systems in which gas phase resistance is practically equal to overall resistance are called gas phase controlled. Systems in which reverse of above is true are called liquid phase controlled. The interphase mass transfer process which occurs in the absorber of a liquid sorbent cooling/dehumidification system is considered as gas phase controlled system in this work.

3.1 Mathematical Model for Absorber Analysis

Absorber used in liquid sorbent system can be categorized as packed towers with different kinds of packings. Absorbers for adiabatic and isothermal operations have been analyzed by Treybal [19] and Sherwood et al [20] respectively. Peng [16] has extended the method of analysis to include both the heat effects evolved in absorption process

and internal cooling or heating. He has used finned-tube surface as packing arrangement in the absorber which is stacked horizontally in the column. The absorber which is a packed tower, used in this work is as shown in Figure 3.2. The packing serves as the contact surface for air and sorbent solution. The spray nozzles on top of the packing spray fine drops of lithium chloride water solution on packing surface. The solution drips down the column through all the contacting surfaces, finally reaches the bottom sump. To maintain a continuous process, part of the solution is pumped to regenerator to keep a fixed concentration of the solution. The cycle is continued by pumping the sorbent solution through an intercooler back to spray nozzles. To analyze the absorber consider a control volume as shown in Figure 3.3, of unit cross-section area, height dz , and make the following assumptions.

1. Process in steady state
2. Liquid phase resistance is disregarded and the mass transfer process is gas phase controlled.
3. The heat released in the absorption process and the latent heat of evaporation for the solvent, both on a unit mass basis, are considered as constant.
4. Gas and liquid sorbent flow rates are constant.
5. The change of temperature and concentration over the thickness of liquid film are negligible.

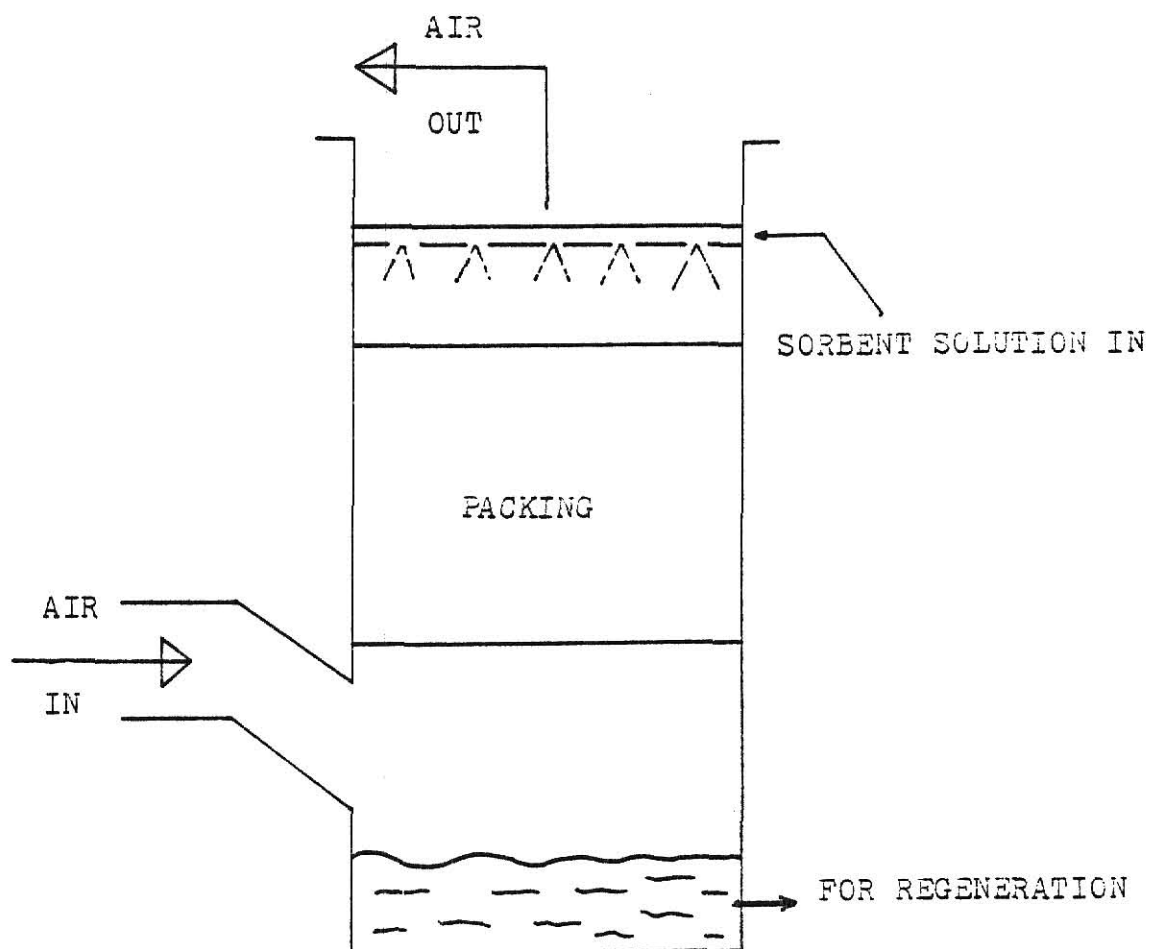


Figure 3.2: Absorber used in liquid sorbent system.

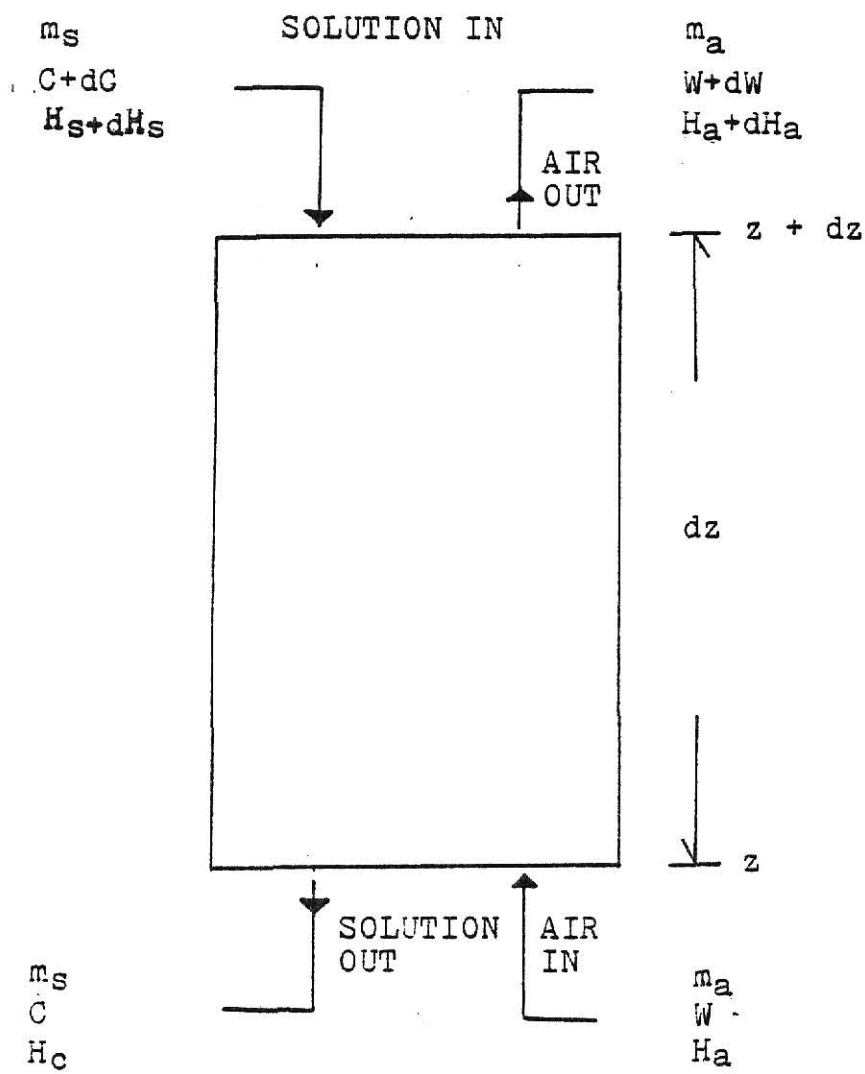


Figure 3.3: Control volume of the absorber.

6. The enthalpy of saturated liquid condensing from air is negligible.

Now defining the enthalpy of sorbent solution as

$$H_s = C_{s1} (t_s - t_o) + \Delta H \quad (3.10)$$

where ΔH is the integral heat of solution/unit mass of the solution, C_{s1} is the specific heat of the solution, t_s is the bulk temperature of the solution, and t_o is a reference datum temperature.

So equation (3.10) can be written as

$$dH_s = C_{s1} dt_s + d(\Delta H) \quad (3.11)$$

Similarly the enthalpy of air water vapor mixture can be written as

$$H_a = C_a (t_a - t_o) + W[C_s (t_a - t_o) + \lambda_s] \quad (3.12)$$

where C_a is the specific heat of dry air, C_s is the specific heat of water vapor in the air, t_a is the bulk temperature of humid air, λ_s is the latent heat of vaporization and W is the humidity ratio of air.

So equation (3.12) can be written as

$$dH_a = C_a dt_a + W C_s dt_a + [C_s (t_a - t_o) + \lambda_s] dW \quad (3.13)$$

Since both heat and mass transfer processes are assumed to be gas controlled, so the temperature of interface is essentially the temperature of the bulk liquid and the rate of heat transfer across the air film from the bulk air to the air sorbent

solution interface is related to change in air temperature by

$$m_a (C_a + WC_s) dt_a = h_a a (t_s - t_a) dz \quad (3.14)$$

where a is the transfer surface area/unit volume of tower, h_a is the heat transfer coefficient, and m_a is the air mass flow rate/unit cross section area.

Similarly for a gas phase controlled process mass transfer across the air film can be related to the change in humidity ratio as follows

$$-m_a dW = h_m a (W - W_e) dz \quad (3.15)$$

where W_e is the water vapor content of air in equilibrium with sorbent solution, and h_m is the mass transfer coefficient. Now the energy balance for the control volume can be written as

$$m_s (H_s + dH_s) + m_a H_a = m_s H_s + m_a (H_a + dH_a) \quad (3.16)$$

where m_s is the sorbent solution mass flow rate/unit cross section area.

Equation (3.16) can be reduced to

$$m_s dH_s = m_a dH_a \quad (3.17)$$

Substituting from Equations (3.11) and (3.13) for dH_s and dH_a Equation (3.17) can be written as

$$m_s [C_{s1} dt_s + d(\Delta H)] = m_a [C_a dt_a + W(C_s dt_a) + dW \{C_s (t_a - t_o) + \lambda_s\}] \quad (3.18)$$

Substituting from transfer Equations (3.14) and (3.15) for dt_a and dW Equation (3.18) can be written as

$$\begin{aligned}
 m_s [C_{sl} dt_s + d(\Delta H)] &= h_m a \left[\frac{h_a C_a}{h_m C_a} (t_s - t_o) \right. \\
 &\quad + W_e \{C_s (t_a - t_o) + \lambda_s\} \} dz \\
 &\quad - h_m a \left[\frac{h_a C_a}{h_m C_a} (t_a - t_o) \right. \\
 &\quad \left. + W \{C_s (t_a - t_o) + \lambda_s\} \} dz \quad (3.19)
 \end{aligned}$$

For transfer of water vapor, it was reported by Threlkeld [21] that the non-dimensional parameter $\frac{h_a}{h_m C_a}$, which is called Lewis number, is not far from one. In this analysis Lewis number is assumed as unity.

Now define air equilibrium enthalpy as

$$H_{ae} = C_a (t_s - t_o) + W_e \{C_s (t_s - t_o) + \lambda_s\} \quad (3.20)$$

since enthalpy of water vapor of air in equilibrium with sorbent solution is very small as compared to the total enthalpy of moist air so

$$W_e (t_a - t_o) C_s \approx W_e (t_s - t_o) C_s \quad (3.21)$$

Equation (3.19) can be reduced to

$$m_a dH_a = h_m a (H_{ae} - H_a) dz \quad (3.22)$$

Equation (3.22) after rewriting, can be integrated between inlet and outlet air states to obtain the required absorber column height

$$Z_t = \frac{m_a}{h_m a} \int_{in}^{out} \frac{dH_a}{H_{ae} - H_a} \quad (3.23)$$

Chilton and Colburn [22] first developed the concept of height of transfer unit, Equation (3.23) can be written as

$$Z_t = HTU \times NTU \quad (3.24)$$

where HTU, or the height of transfer unit is defined as

$$HTU = \frac{m_a}{h_m a} \quad (3.25)$$

The right hand side integral of Equation (3.23) is a dimensionless quantity and termed as number of transfer units, NTU. The integral between air inlet and outlet condition is a measure of the difficulty of interphase mass transfer.

The steady state conservation equations derived earlier for a control volume can be written for each flow

stream in their respective directions as follows

$$\frac{dW}{dz} = - \frac{h_m a}{m_a} (W - W_e) \quad (3.26)$$

Now define C as the weight concentration of water in solution, then according to law of conservation of mass change in C can be written as

$$\frac{dC}{dz} = \frac{h_m a}{m_s} (W - W_e) \quad (3.27)$$

Equation (3.26) in combination with Equation (3.27) state that for a steady state system amount of moisture lost by air is added to the solution when air is dehumidified. The air enthalpy equation will reduce to

$$\frac{dH_a}{dz} = \frac{h_m a}{m_a} (H_{ae} - H_a) \quad (3.28)$$

Thus, Equation (3.26) through (3.28) describe the simultaneous heat transfer with interphase mass transfer operation in a packed absorber. Since the properties of sorbent solution are function of temperature and concentration, so the process becomes coupled heat and mass transfer problem and Equations (3.26) through (3.28) become coupled differential equations.

Conditions of air stream at inlet can be used to normalize Equations (3.26) through (3.28) with the following nondimensional parameters

$$\bar{W} = \frac{W}{W_o}, \quad \bar{C} = \frac{C}{W_o}, \quad \bar{z} = \frac{z}{DZ}$$

$$\bar{W}_e = \frac{W_e}{W_o}, \quad \bar{H}_e = \frac{H_e}{H_{ao}}, \quad \bar{H}_a = \frac{H_a}{H_{ao}}$$

where W_o and H_{ao} are inlet air humidity ratio and enthalpy respectively.

The normalized equations can be written as

$$\frac{d\bar{W}}{d\bar{z}} = -C_1 (\bar{W} - \bar{W}_e) \quad (3.29)$$

$$\frac{d\bar{C}}{d\bar{z}} = C_2 (\bar{W} - \bar{W}_e) \quad (3.30)$$

$$\frac{d\bar{H}_a}{d\bar{z}} = -C_1 (\bar{H}_a - \bar{H}_{ae}) \quad (3.31)$$

where C_1 through C_2 are constants and are defined as

$$C_1 = \frac{h_m \cdot a \cdot DZ}{m_a} \quad (3.32)$$

$$C_2 = \frac{h_m \cdot a \cdot DZ}{m_s} \quad (3.33)$$

These constants C_1 and C_2 as can be seen are similar to height of transfer unit and signify the ratio of mass transfer potential to mass flow rate of respective streams in the absorber. Thus, these constants are controlling factors in the design of an absorber. The order of magnitude of these constants determine the size of an absorber.

3.2 Cooling Tower

Dehumidification of air by absorption process is highly exothermic in nature. The heat of absorption which consists of latent heat of vaporization and heat of dilution accumulates in the liquid phase. Since the equilibrium vapor pressure of air in contact with sorbent solution is directly proportional to the temperature, the rapid increase in temperature of the liquid impairs an efficient operation of the absorber. The process will even cease to proceed if temperature rise in liquid phase increases the vapor pressure of air in equilibrium with it to approach the vapor pressure of bulk air. So it is necessary to cool the solution to remove the heat of absorption. In the system which is analyzed in this work cooling of liquid sorbent is accomplished in an external counter flow heat exchanger. Cooling tower water is used as the coolant in the heat exchanger.

A cooling tower cools water by contacting it with air

and evaporating some of the water. Air may circulate by wind currents or gravity circulation, or fans may force or draw air through the tower. A large surface area for simultaneous heat and mass transfer is provided, by using different kinds of packings in the tower. Water is sprayed through nozzles on the packing surface.

Performance of a cooling tower is commonly expressed in terms of range and approach. Figure 3.4, shows the air and water temperature relationship as they pass through a counterflow cooling tower. Range of a cooling tower is defined as the reduction in temperature of the water in the cooling tower which for a steady state operation is same as the water temperature rise through the load heat exchanger. Approach is the difference between the wet bulb temperature of the entering air and the temperature of leaving water. The approach is a function of cooling tower capability. Thermal performance of a cooling tower depends on its size, entering air wet bulb temperature, and heat load imposed on the tower.

The cold water temperature from a specific cooling tower for fixed air and water flow rates, thus depends on the entering air wet bulb temperature, and the heat load which dictates the range it must accomodate.

The basic theory of heat and mass transfer in a cooling tower has been a topic for many researchers. A complete mathematical model was first developed by Merkel

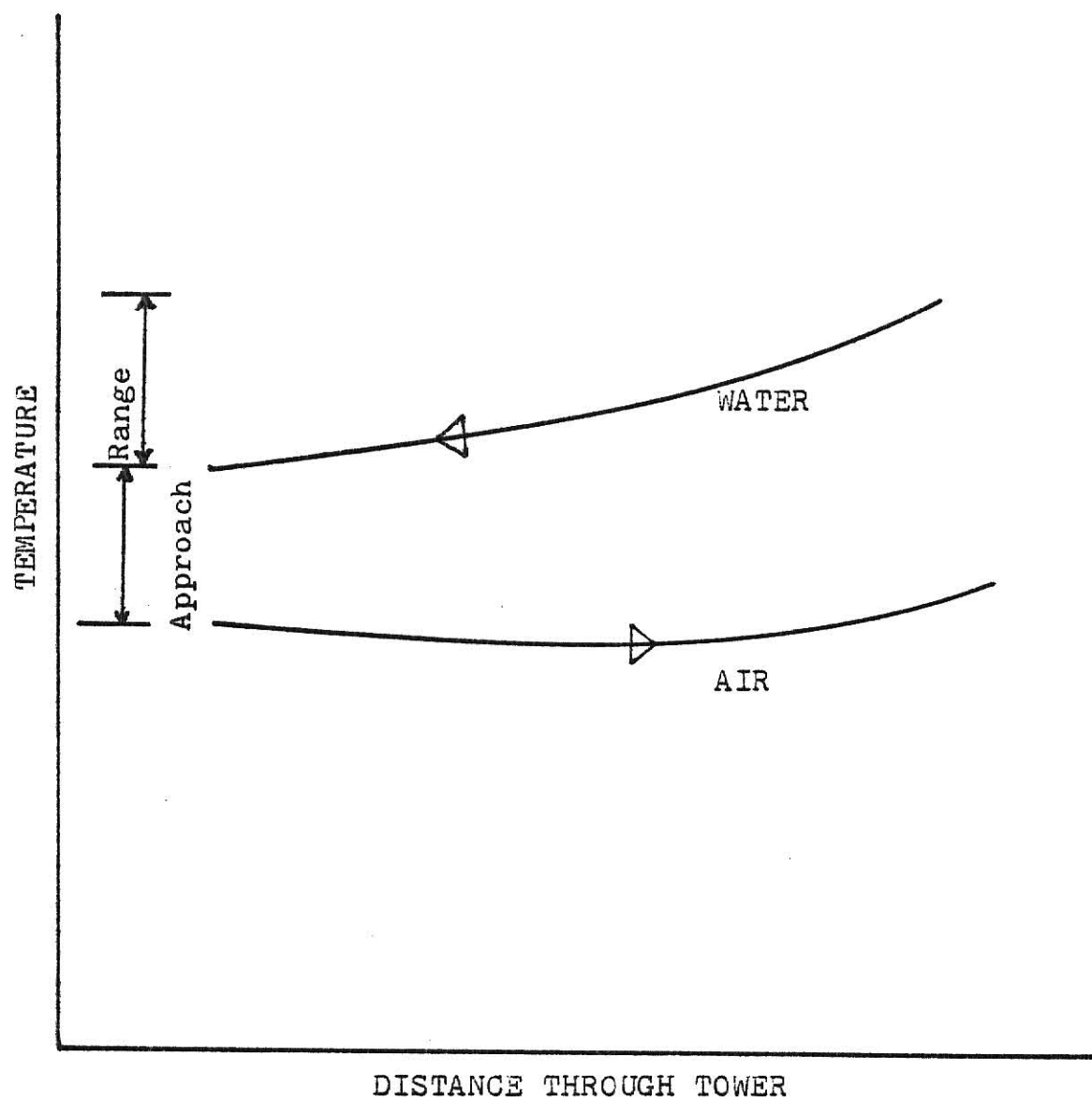


Figure 3.4: Temperature relationship between water and air in a counterflow cooling tower.

[23]. He used the concept of enthalpy potential by combining heat and water vapor transfer equations and unity Lewis number assumption in his analysis.

For partial load performance analysis of the liquid sorbent dehumidification/cooling system studied in this work, the cooling tower used is designated to the load heat exchanger, and is allowed to float with the outside air conditions. The performance of cooling tower for such a configuration as described above becomes coupled with the performance of the liquid sorbent system and ambient air conditions.

A conventional and convenient way of expressing the performance of a cooling tower is to plot the wet bulb temperature against leaving water temperature from the cooling tower for different ranges. The performance curve is a unique property of an individual cooling tower for a fixed water flow rate. When operating at any other flow rates, the same tower performance will be described by other curves. The performance curves for different cooling towers at different flow rates though are different, but exhibit similar trends. Manufacturers of cooling towers also furnish the predicted performance curves for evaluation of performance tests. These curves have a similar format as described above. So instead of using a mathematical model to predict the performance of a cooling tower at off-design conditions, it was decided to use a fitted equation for performance evaluation of a cooling tower in this study.

Since the performance curves for different cooling towers exhibit similar trends, so by simply changing the constant coefficients of the fitted equation, any cooling tower commercially available can be modelled for off design performance evaluation. Thus, a fitted equation to model a cooling tower does not impose any limitations on the partial load analysis of liquid sorbent system.

The performance curve of the cooling tower selected for the system studied in this work is given in ASHRAE Handbook of Equipment [24]. The design flow rate of water for this cooling tower is 3 gpm for nominal ton (0.054 l/s per kW). This cooling tower at design conditions is able to cool the water from 95°F (35°C) to 85°F (29.4°C) at 78°F (25.6°C) wet bulb temperature. In the analysis throughout this work 100% design flow rate of water is used. The performance curve for this cooling tower is shown in Figure 3.5. The equation developed to predict the performance of this cooling tower can be written as

$$t_{co} = 95 - 154e^{-0.04R} \times e^{\frac{-74}{110-t_{wb}}} \quad (3.34)$$

where t_{co} is the cold water temperature leaving the cooling tower, R is the range of the cooling tower, and t_{wb} is the ambient air wet bulb temperature.

Comparison between the values found from Figure 3.5 and the calculated values from Equation (3.34) is presented in Table 3.1.

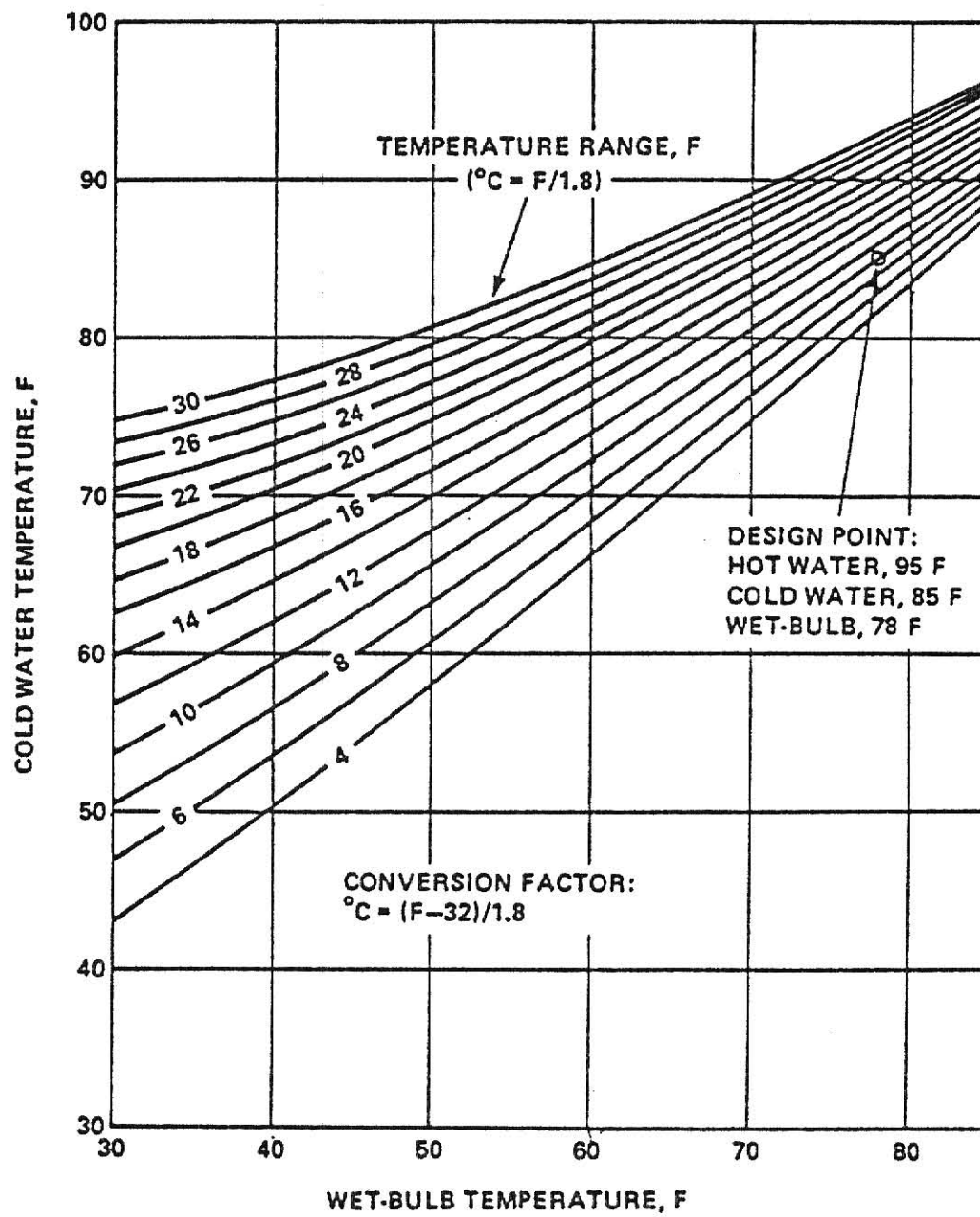


Figure 3.5: Cooling tower performance curve.

TABLE 3.1: Comparison of Equation (3.34) with Figure (3.5).

R °F	T _{wb} °F	T _{co} °F Eq (3.34)	T _{co} °F Fig (3.5)
4	30	43	43
4	60	65	66
4	80	84	84
10	30	54	54
10	60	72	72
10	80	86	86
20	30	68	67
20	60	79	79
20	80	89	91
30	30	77	75
30	60	84	85
30	80	91	94

3.3 Sorbent Solution Cooler

Sorbent solution cooler, which is an ordinary heat exchanger is employed to cool the sorbent solution before it enters the absorber. For an efficient heat transfer the cooler is assumed to be counter flow in nature. Sorbent solution in the cooler is cooled by cooling tower water, and energy rejected by the sorbent solution in the cooler is discarded to the cooling tower.

So the energy balance across the cooler can be written as

$$m_w (H_{wi} - H_{wo}) + m_s (H_{si} - H_{so}) = 0 \quad (3.35)$$

where m_w is the water mass flow rate, H_{wi} and H_{wo} are the enthalpy of water entering and leaving the cooling tower respectively. Similarly H_{si} and H_{so} are the enthalpy of sorbent solution entering and leaving the absorber, respectively.

The effectiveness of a heat exchanger is defined as

$$\epsilon = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} \quad (3.36)$$

To model the cooler for liquid sorbent solution having the minimum heat capacity of the two fluids, heat transfer and effectiveness equations for the counter flow heat exchanger can be written as

$$\epsilon = \frac{(t_{so} - t_{si})}{(t_{so} - t_{co})} \quad (3.37)$$

and

$$q = UA \frac{(t_{si} - t_{co}) - (t_{so} - t_{ci})}{\ln [(t_{si} - t_{co}) / (t_{so} - t_{ci})]} \quad (3.38)$$

where q is the rate of heat transfer, U is the overall heat transfer coefficient. A is the heat transfer area, and t_{ci} is the temperature of water entering the cooling tower. t_{si} and t_{so} are the temperature of the sorbent solution entering and leaving the absorber respectively.

Using Equations (3.37) and (3.38) an expression for the effectiveness of the cooler can be written as

$$\epsilon = \frac{1 - e^D}{\frac{C_S}{C_W} - e^D} \quad (3.39)$$

and

$$D = \frac{UA}{C_S} \left(1 - \frac{C_S}{C_W}\right) \quad (3.40)$$

where $\frac{UA}{C_S}$ is the NTU, C_S and C_W are the total heat capacities of sorbent solution and water, respectively.

Initial calculations which were performed to determine the mass flow ratio of different fluids involved in the system have revealed that sorbent solution is in fact the minimum heat capacity fluid in the cooler.

3.4 Method of Solution

Since the properties of lithium chloride water solution, which is used as liquid sorbent in this analysis, change with temperature and solution concentration a differential approach was used to develop the model. Therefore, for the analysis of liquid sorbent system considered, the absorber was divided into a number of control volumes, and Equations (3.29) and (3.31) were written in a finite difference form for each control volume in the flow direction. In each control volume, all the properties of fluid streams were assumed to be constant.

To analyze the partial load performance of this liquid sorbent cooling/dehumidification system, a computer program was written. Numerous computer runs were performed to determine appropriate combination of controlling parameters.

A detailed description of method of analysis, with different functional relationships used in the computer program is presented in Chapter IV.

CHAPTER IV

PERFORMANCE ANALYSIS AND RESULTS

Components of a liquid sorbent cooling/dehumidification system were modeled in the previous chapter to describe the performance of this system. Controlling parameters used to describe the performance of the absorbers were C_1 and C_2 , which are given by Equations (3.32) and (3.33).

Another parameter which controls the performance of sorbent solution cooler can be defined as

$$C_3 = \frac{m_s}{m_w} = \frac{\text{sorbent mass flow rate}}{\text{water mass flow rate}} \quad (4.1)$$

A similar parameter for the absorber can also be defined as

$$C_4 = \frac{m_s}{m_a} = \frac{\text{sorbent mass flow rate}}{\text{air mass flow rate}} \quad (4.2)$$

Hence, these four nondimensional parameters govern the performance of the liquid sorbent cooling/dehumidification system analyzed here. The nondimensional parameter C_1 is the product of C_2 and C_4 . Because the cooling tower is modeled for a specific water mass flow rate, limitations are imposed on the value of C_3 also. To conduct the parametric analyses, the liquid sorbent dehumidification system

was required to process relatively hot and humid air at 95°F (35°C) dry bulb and 79.9°F (26.6°C) wet bulb conditions. Instead of assuming an arbitrary absorber column height which can vary from zero to infinity it was decided to fix the exit air humidity ratio of the exit process air to be 0.0094 lb/lb of dry air (0.0094 kg/kg) for the above mentioned process air inlet conditions. Numerous computer runs were performed to get optimum values of the controlling factors to accomplish this performance. A constant similar to C_1 was used by Peng [16] and after taking into account the flooding limit, the value of this constant was reported to lie between 1 and 6. Since these controlling parameters are inter-related to each other, variation in the values of these constants have to be in the limits imposed on them. It was found that if enough sorbent solution is available (higher value of C_4), dehumidification will proceed at a faster rate for higher value of C_1 . The reason for this behavior is that higher C_1 would mean either higher value of mass transfer coefficient or more transfer surface area, both of which will increase the water vapor transfer from air to sorbent solution if the driving force for mass transfer is present. The mass transfer driving force, which is the difference in water vapor pressure between air and sorbent solution, is inversely proportional to the sorbent solution temperature. Thus to have less temperature rise of the sorbent solution, more sorbent solution is required which would mean a higher value of C_4 .

The controlling factor C_1 , when having a high value would mean a small equipment size to accomplish a certain performance, while it would require larger equipment for lower values of C_1 .

To accomplish the performance mentioned earlier the minimum possible value of C_4 was found to be 1. The corresponding values of C_1 , C_2 and C_3 were 2.5, 2.5 and 0.7, respectively. For values of C_4 less than 1 sorbent solution can not accomplish the required dehumidification as temperature rise of solution is such that the vapor pressure of sorbent solution approaches the vapor pressure of air before the required dehumidification is achieved.

4.1 Computer Model

To proceed with the partial load performance analysis of the liquid sorbtion dehumidification/cooling system, the only information available is the air inlet conditions.

An iterative procedure was adopted to determine the process air wet bulb temperature at the absorber inlet when its dry bulb temperature and humidity ratio are known.

The iteration scheme starts by first assuming the wet bulb temperature equal to the dry bulb temperature of the ambient air, and then decreasing it in next iterations, till it converges to actual wet bulb temperature. The saturation pressure of air p_{ws} is determined by using the equation of Keenan et al., [25] which can be written as

$$\log_e \left(\frac{p_{ws}}{217.99} \right) = (0.01/T) (374.136 - t) \sum_{i=1}^8 F_i (0.65 - 0.01t)^{i-1} \quad (4.3)$$

where t and T are the iterated temperatures in degrees Celcius and Kelvins respectively. The values of F_1 through F_8 are given as follows:

$$F_1 = -741.9242 \quad (4.4)$$

$$F_2 = -29.721 \quad (4.5)$$

$$F_3 = -11.55286 \quad (4.6)$$

$$F_4 = -0.8685635 \quad (4.7)$$

$$F_5 = 0.1094098 \quad (4.8)$$

$$F_6 = 0.439993 \quad (4.9)$$

$$F_7 = 0.2520658 \quad (4.10)$$

$$F_8 = 0.05218684 \quad (4.11)$$

and the humidity ratio of saturated air W_s is given by

$$W_s = 0.62198 \left(\frac{p_{ws}}{14.696 - p_{ws}} \right) \quad (4.12)$$

and the humidity ratio of air when saturated at iterated temperature can be written as

$$W = \left(\frac{1093 - 0.556 t^*}{1093 + 0.444 t_{ai} - t^*} \right) W_s - 0.24 (t_{ai} - t^*) \quad (4.13)$$

where t^* is the iterated temperature in Fahrenheit, and t_{ai} is the inlet air dry bulb temperature. The iterated temperature equals the wet bulb temperature when the humidity ratio W from Equation (4.13) converges to the inlet air humidity ratio.

The temperature and concentration of sorbent solution changes continuously as it passes through the absorber. Uemura [26] has presented the functional relationships of lithium chloride water solution. These relationships were used to determine the characteristic properties of lithium chloride water solution which has been used as the sorbent in this analysis.

The vapor pressure p , of aqueous solution of lithium chloride is given by

$$\log p = A' - B'/(t_{sc} + 230) \quad (4.14)$$

where t_{sc} is the temperature of lithium chloride water solution in degrees Celsius, and

$$A' = 7.5713 + 3.7964 \times 10^{-2} X + 9.0133 \times 10^{-4} X^2 \quad (4.15)$$

$$B' = 1.5821 \times 10^3 + 1.0364 \times 10X - 1.4462 \times 10^{-1} X^2 \quad (4.16)$$

where X is the weight concentration of lithium chloride water solution.

The specific heat of the lithium chloride water solution is given by

$$C_{sl} = A + Bt_{sc} + Ct_{sc}^2 \quad (4.17)$$

where A, B, and C, are constants and are given by

$$A = 1.0020 - 1.2505 \times 10^{-2}X + 0.7575 \times 10^{-4}X^2 \quad (4.18)$$

$$B = -5.554 \times 10^{-4} - 1.5178 \times 10^{-5}X + 6.8248 \times 10^{-7}X^2 \quad (4.19)$$

$$C = 5.2266 \times 10^{-6} + 3.6623 \times 10^{-8}X - 3.8345 \times 10^{-9}X^2 \quad (4.20)$$

In the operation of a liquid sorbent system moisture is absorbed by solution from air as dehumidification proceeds, thus heat is given off and this is known as heat of dilution.

The heat of dilution is given by

$$\Delta H = 0.01 (X)^{2.5} \quad (4.21)$$

As mentioned earlier in this chapter, the liquid sorbent dehumidification/cooling system was required to dehumidify relatively hot and humid air at 95°F (35°C) dry bulb, and 79.9°F (26.6°C) wet bulb temperature to a humidity ratio of 0.0094 lb/lb (kg/kg). The system was required to accomplish this performance because it is a typical industrial application. An actual system located in Southern United States was mentioned by Griffiths [6] to accomplish this same performance.

To achieve this performance, the absorber was sized

by using an indicator in the computer program. As the desired dehumidification was attained, the indicator was fixed to that value and thus by having constant controlling parameters and indicator value, the physical dimensions of the system were fixed. This system was then subjected to different air inlet conditions to analyze the performance at partial load.

The absorber and regenerator are operated in a counter flow manner and the properties of the fluids at either end are not known, so an iterative procedure was used to analyze the performance. After specifying the values of controlling parameters and the NTU for sorbent solution cooler, the calculations proceed by first guessing the exit temperature and concentration of liquid sorbent solution. The enthalpy of air in equilibrium with lithium chloride water solution is calculated by first determining the vapor pressure of the solution at that temperature and then using this vapor pressure to calculate the humidity ratio of air in equilibrium. The change in enthalpy and humidity ratio of air over the control volume is calculated by using the model presented in previous chapter, similarly the change in the concentration and the temperature of sorbent solution in the flow direction is calculated over the control volume. The absorber is divided into a discrete number of control volumes, and the calculations for heat and mass transfer continue until the predetermined indicator

value is reached. The performance of sorbent solution cooler is then analyzed. Heat transfer in the cooler is determined by using the energy, and effectiveness equations desired for the cooler. The cooling water temperatures are thus obtained. The performance of cooling tower is then analyzed by using cooling tower model presented earlier. The capability of cooling tower is checked by comparing the cold water temperature calculated from cooling requirements in the solution cooler. The calculation cycle continues by changing the exit sorbent temperature and concentration until the comparison between cold water temperature from the cooling tower and that required by the sorbent cooler agrees within a specified convergence criterion.

4.2 Analysis and Results

The performance of a liquid sorbent system using 40% lithium chloride water solution by weight with a cooling tower floating with ambient air conditions was analyzed for various air inlet dry bulb temperature and humidity ratio. The cooling/dehumidification performance of this system is given in Table 4.1. It is to be noted that in all the performance tables presented in this chapter the temperature of air, sorbent solution and water is in degrees Fahrenheit. The enthalpy of air is in Btu/lb, and the concentration of sorbent solution is in % by weight.

It can be seen from this table that when the system is subjected to less severe conditions more dehumidification is obtained and also that the wet bulb temperature decreases for these conditions, as colder water is available from the cooling tower. Furthermore, for the same

TABLE 4.1: Performance of the liquid sorbent-cooling tower system using a 40% solution.

Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.9	.0188	.0094	43.5	33.1
Tsi	Tso	Tco	R	Tao	Xso
98.9	107.2	85.6	8.4	94.9	39.63
Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.2	.0180	.0090	42.7	32.5
Tsi	Tso	Tco	R	Tao	Xso
88.2	105.9	85.0	8.2	94.1	39.64
Tai	Twb	Wi	Wo	Hai	Hao
95.0	75.9	.0150	.0078	39.3	30.2
Tsi	Tso	Tco	R	Tao	Xso
84.8	100.6	81.9	7.3	90.3	39.71
Tai	Twb	Wi	Wo	Hai	Hao
95.0	72.3	.0120	.0065	36.0	27.9
Tsi	Tso	Tco	R	Tao	Xso
80.9	94.9	78.4	6.4	86.3	39.78
Tai	Twb	Wi	Wo	Hai	Hao
90.0	78.8	.0188	.0090	42.3	32.3
Tsi	Tso	Tco	R	Tao	Xso
87.7	105.4	84.5	8.2	93.1	39.61
Tai	Twb	Wi	Wo	Hai	Hao
90.0	74.5	.0150	.0074	38.1	29.3
Tsi	Tso	Tco	R	Tao	Xso
83.3	98.6	80.5	7.1	88.2	39.70

Tai	Twb	Wi	Wo	Hai	Hao
90.0	70.9	.0120	.0062	34.8	27.0
Tsi	Tso	Tco	R	Tao	Xso
79.4	92.8	76.9	6.2	84.1	39.77
Tai	Twb	Wi	Wo	Hai	Hao
90.0	68.4	.0100	.0055	32.6	25.4
Tsi	Tso	Tco	R	Tao	Xso
76.3	88.6	74.1	5.7	81.0	39.82
Tai	Twb	Wi	Wo	Hai	Hao
85.0	73.1	.0150	.0071	36.9	28.5
Tsi	Tso	Tco	R	Tao	Xso
81.9	96.6	79.2	6.8	86.2	39.69
Tai	Twb	Wi	Wo	Hai	Hao
85.0	69.5	.0120	.0060	33.6	26.2
Tsi	Tso	Tco	R	Tao	Xso
77.9	90.7	75.6	5.9	82.1	39.76
Tai	Twb	Wi	Wo	Hai	Hao
80.0	71.7	.0150	.0068	35.6	27.7
Tsi	Tso	Tco	R	Tao	Xso
80.5	94.6	78.0	6.5	84.2	39.68
Tai	Twb	Wi	Wo	Hai	Hao
80.0	67.8	.0120	.0056	32.4	25.3
Tsi	Tso	Tco	R	Tao	Xso
76.1	88.4	73.8	5.7	79.7	39.75

inlet humidity ratio, more dehumidification can be accomplished for a lower dry bulb temperature. For most applications the dehumidified air is required at fixed humidity ratio to take care of a specified latent load. It is obvious from Table 4.1, that this system will not be able to accomplish fixed process air humidity ratio when subjected to partial loads. It can also be seen from this table that the dry bulb temperature of the process air after dehumidification decreases with a decrease in humidity ratio of outside air for a fixed dry bulb temperature.

The performance of this liquid sorbent system, using 35% lithium chloride water solution was also analyzed. The performance of this system when subjected to the same air inlet conditions is presented in Table 4.2.

It is obvious from Table 4.2, that less dehumidification is accomplished when 35% sorbent solution is used in the absorber compared to when 40% sorbent solution is used. This happens because the vapor pressure of sorbent solution increases with a decrease in concentration at a fixed temperature, which decreases the mass transfer potential. The plots of dimensionless air humidity ratio $(\frac{W_o}{W_i})$ distribution, dimensionless sorbent solution temperature $(\frac{T_{si}}{T_{so}})$ distribution and dimensionless air enthalpy $(\frac{H_{ao}}{H_{ai}})$ distribution in the absorber at design conditions for the system using 35% and 40% concentrated solution are shown in Figure (4.1) through (4.3).

The cooling tower was the source of cold water to

TABLE 4.2: Performance of the liquid sorbent-cooling tower system using a 30% solution.

Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.9	.0188	.0114	43.5	34.8
Tsi	Tso	Tco	R	Tao	Xso
87.8	102.0	85.2	6.9	93.0	34.74
Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.2	.0180	.0110	42.7	34.2
Tsi	Tso	Tco	R	Tao	Xso
87.0	100.8	84.4	6.7	92.1	34.76
Tai	Twb	Wi	Wo	Hai	Hao
95.0	75.9	.0150	.0095	39.3	31.7
Tsi	Tso	Tco	R	Tao	Xso
83.6	96.0	81.3	6.0	88.5	34.81
Tai	Twb	Wi	Wo	Hai	Hao
95.0	72.3	.0120	.0080	36.0	29.1
Tsi	Tso	Tco	R	Tao	Xso
79.7	90.8	77.7	5.4	84.5	34.86
Tai	Twb	Wi	Wo	Hai	Hao
90.0	78.8	.0188	.0109	42.3	33.8
Tsi	Tso	Tco	R	Tao	Xso
86.5	100.3	83.9	6.7	91.0	34.73
Tai	Twb	Wi	Wo	Hai	Hao
90.0	74.5	.0150	.0091	38.1	30.7
Tsi	Tso	Tco	R	Tao	Xso
82.2	94.2	80.0	5.8	86.5	34.79

Tai	Twb	Wi	Wo	Hai	Hao
90.0	70.9	.0120	.0076	34.8	28.1
Tsi	Tso	Tco	R	Tao	Xso
78.1	88.8	76.1	5.2	82.4	34.85
Tai	Twb	Wi	Wo	Hai	Hao
90.0	68.4	.0100	.0068	32.6	26.5
Tsi	Tso	Tco	R	Tao	Xso
75.4	85.1	73.6	4.7	79.6	34.89
Tai	Twb	Wi	Wo	Hai	Hao
85.0	73.1	.0150	.0087	36.9	29.8
Tsi	Tso	Tco	R	Tao	Xso
80.7	92.3	78.5	5.6	84.4	34.78
Tai	Twb	Wi	Wo	Hai	Hao
85.0	69.5	.0120	.0073	33.6	27.2
Tsi	Tso	Tco	R	Tao	Xso
76.5	86.8	74.6	5.0	80.2	34.84
Tai	Twb	Wi	Wo	Hai	Hao
80.0	71.7	.0150	.0083	35.6	28.9
Tsi	Tso	Tco	R	Tao	Xso
79.3	90.4	77.2	5.4	82.4	34.77
Tai	Twb	Wi	Wo	Hai	Hao
80.0	67.8	.0120	.0069	32.4	26.3
Tsi	Tso	Tco	R	Tao	Xso
74.8	84.7	72.9	4.8	78.0	34.82

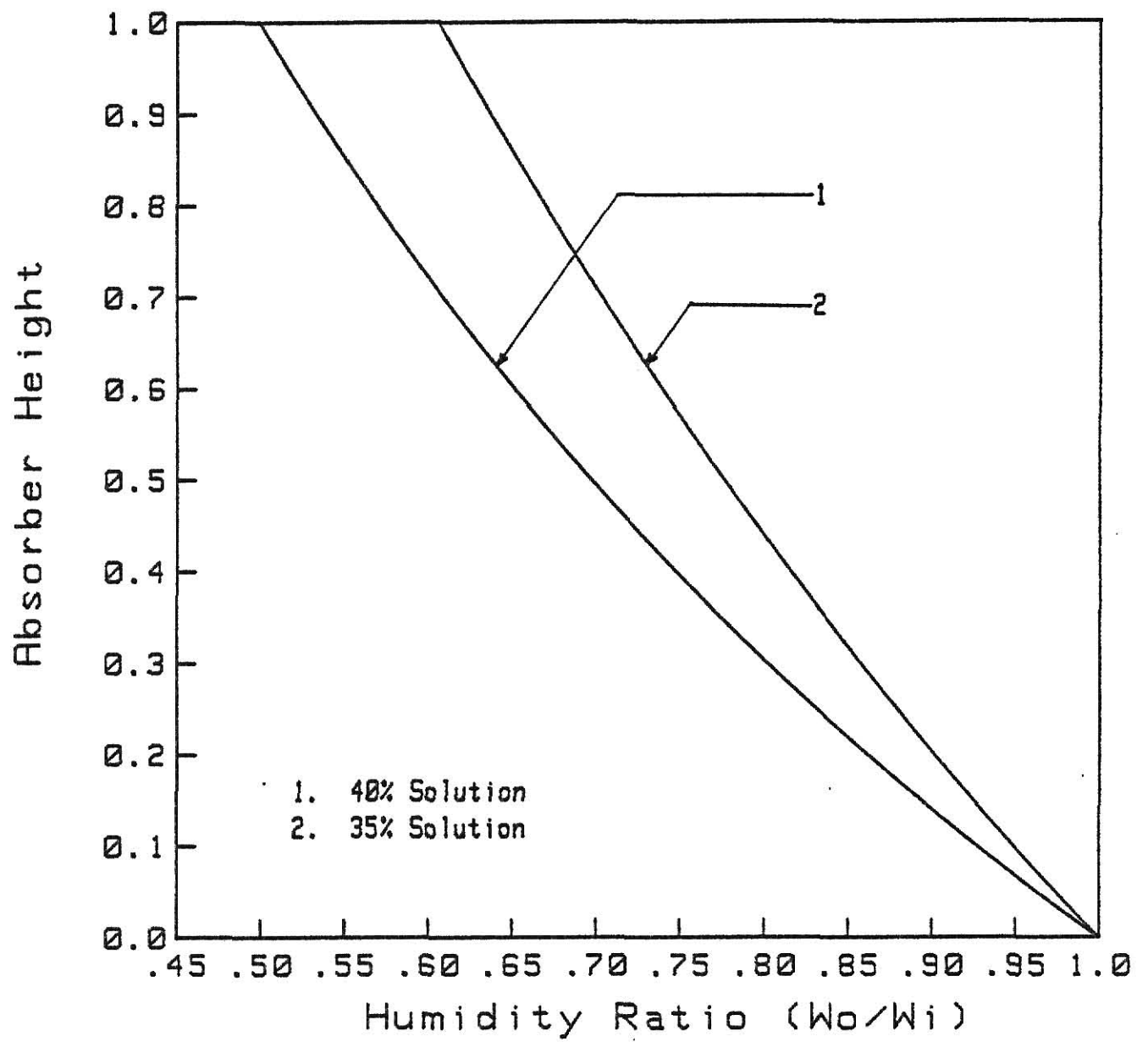


Figure 4.1: Humidity ratio distribution of air in the absorber using cooling tower.

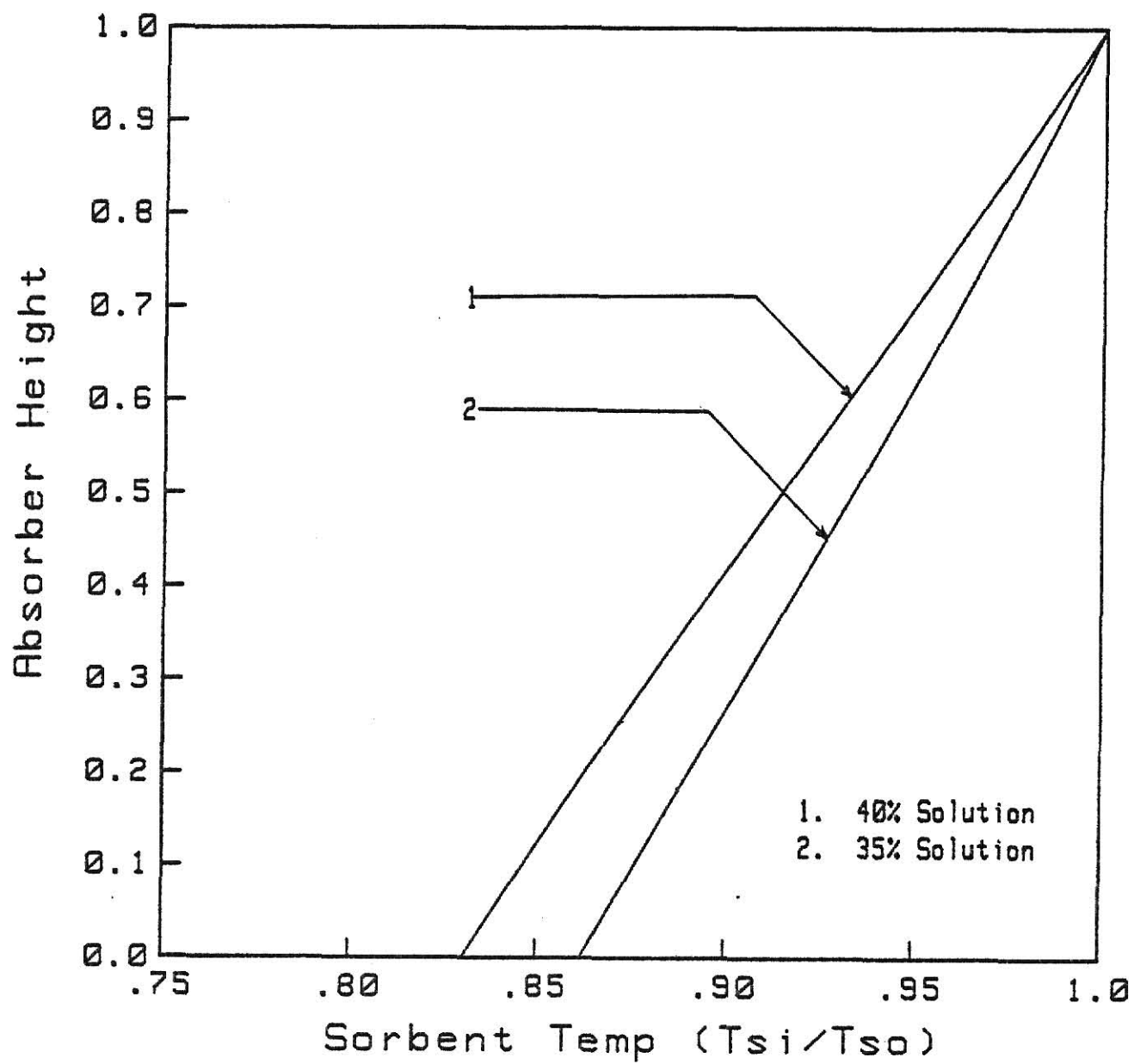


Figure 4.2: Sorbent solution temperature distribution in the absorber using cooling tower.

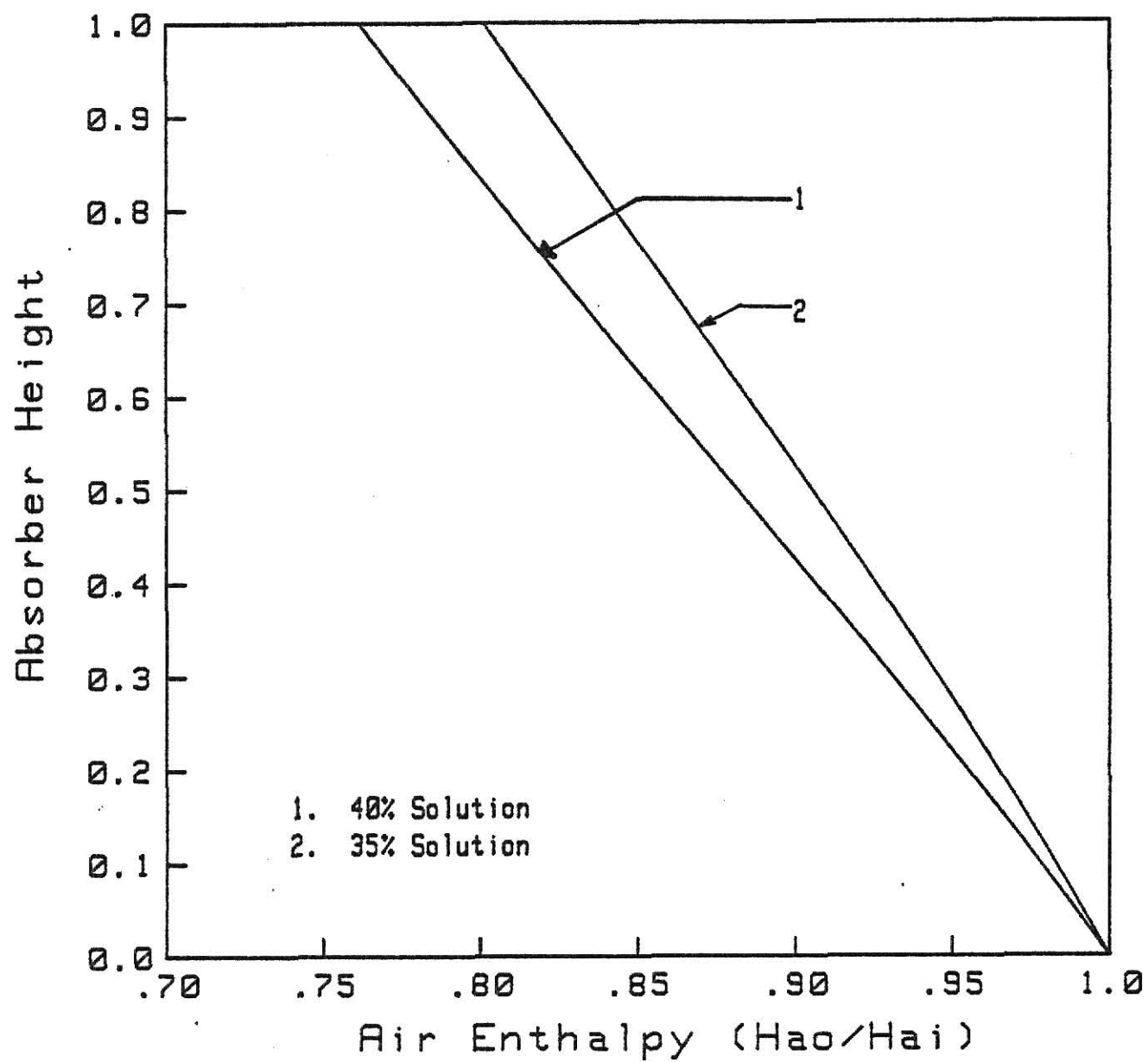


Figure 4.3: Air enthalpy distribution in the absorber using cooling tower.

cool the sorbent solution in the liquid sorbent systems analyzed above. If the cooling tower in this system is replaced by an external source of chilled water (from mechanical or absorption refrigeration system) the system would be able to furnish process air at fixed humidity ratio. The schematic diagram of this system is shown in Figure 4.4. The performance of this system with the same controlling parameters, which were used in the previous system, under partial load was analyzed. The system was required to dehumidify the process air to a humidity ratio of 0.0055 lb/lb (kg/kg) using 40% solution of lithium chloride. The cooling/dehumidification performance of this system for different air conditions is presented in Table 4.3. It can be seen that the minimum cold water temperature required by the system is 55.7°F (13.2°C), with outside air at a dry bulb temperature of 95°F (35°C) and a humidity ratio of 0.0188 lb/lb (kg/kg). For constant dry bulb temperatures of the outside air, temperature of cooling water increases with a decrease in the humidity ratio of this air. Furthermore for a constant humidity ratio with a decreasing dry bulb temperature of outside air, the system requires cold water at higher temperatures.

This behavior can be explained as follows: In the absorber, the mass transfer takes place between the lithium chloride solution and air and the driving force is the difference in the water vapor pressure between them. When the humidity ratio of the air decreases at the inlet less

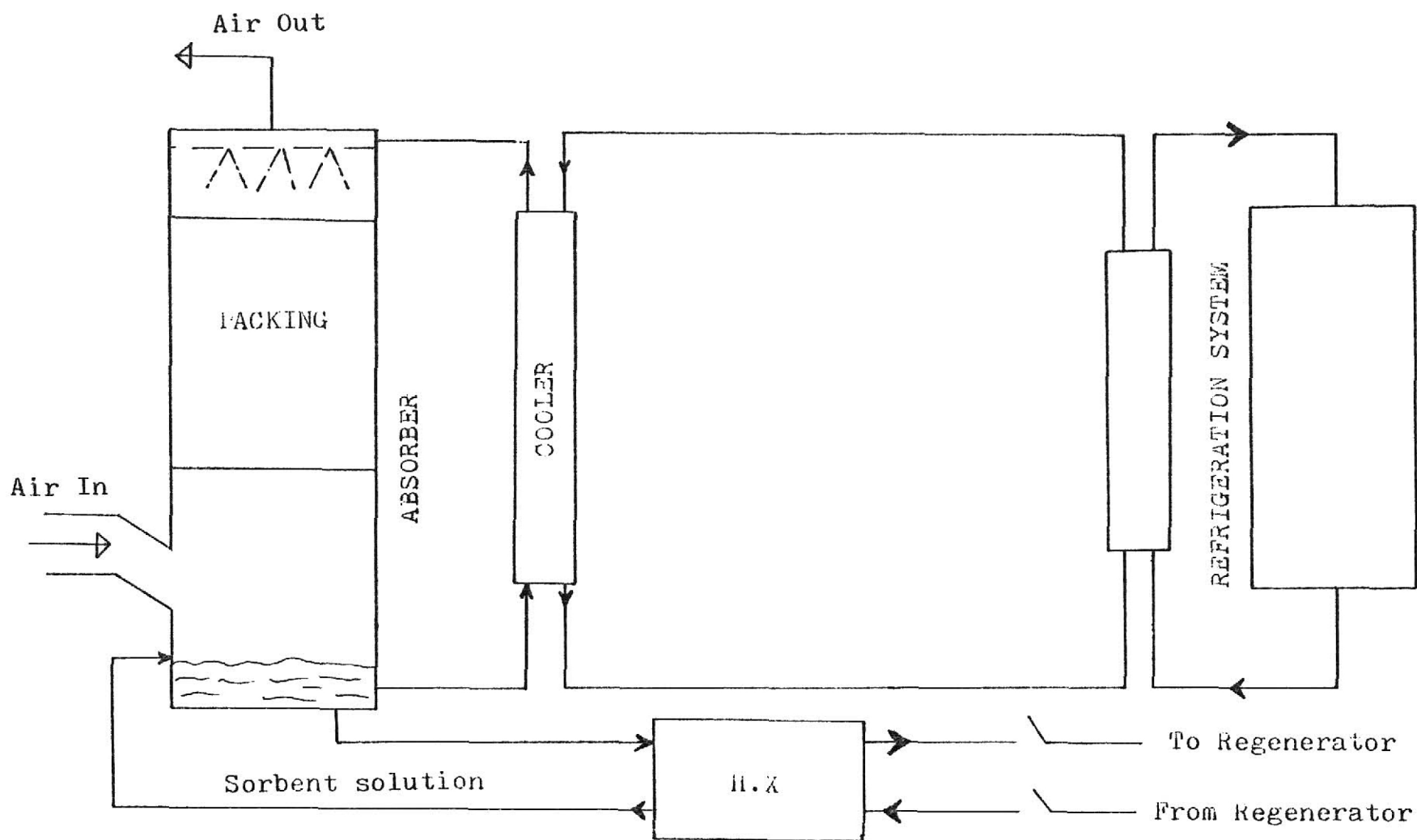


Figure 4.4: Schematic diagram of liquid sorbent system with an external source of chilled water.

TABLE 4.3: Performance of the liquid sorbent system using a 40% solution with an external source of chilled water.

Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.9	.0188	.0055	43.5	23.8
Tsi	Tso	Tco	R	Tao	Xso
61.9	95.7	55.7	15.6	74.2	39.47
Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.2	.0188	.0055	42.7	24.0
Tsi	Tso	Tco	R	Tao	Xso
63.4	95.3	57.6	14.7	75.1	39.51
Tai	Twb	Wi	Wo	Hai	Hao
95.0	75.9	.0150	.0055	39.3	24.7
Tsi	Tso	Tco	R	Tao	Xso
68.5	93.5	63.9	11.5	77.9	39.62
Tai	Twb	Wi	Wo	Hai	Hao
95.0	72.3	.0120	.0055	36.0	25.4
Tsi	Tso	Tco	R	Tao	Xso
73.3	91.5	70.0	8.4	80.4	39.74
Tai	Twb	Wi	Wo	Hai	Hao
90.0	78.8	.0188	.0055	42.3	23.8
Tsi	Tso	Tco	R	Tao	Xso
62.9	94.7	57.1	14.7	74.0	39.47
Tai	Twb	Wi	Wo	Hai	Hao
90.0	74.5	.0150	.0055	38.1	24.6
Tsi	Tso	Tco	R	Tao	Xso
69.0	92.3	64.8	10.7	77.4	39.62

Tai	Twb	Wi	Wo	Hai	Hao
90.0	70.9	.0120	.0055	34.8	25.2
Tsi	Tso	Tco	R	Tao	Xso
73.8	90.3	70.9	7.6	79.9	39.74
Tai	Twb	Wi	Wo	Hai	Hao
90.0	68.4	.0100	.0055	32.6	25.6
Tsi	Tso	Tco	R	Tao	Xso
76.8	88.8	74.6	5.6	81.4	39.82
Tai	Twb	Wi	Wo	Hai	Hao
85.0	73.1	.0150	.0055	36.9	24.5
Tsi	Tso	Tco	R	Tao	Xso
69.8	91.2	65.9	9.9	77.0	39.62
Tai	Twb	Wi	Wo	Hai	Hao
85.0	69.5	.0120	.0055	33.6	25.1
Tsi	Tso	Tco	R	Tao	Xso
74.4	89.1	71.7	6.8	79.4	39.74
Tai	Twb	Wi	Wo	Hai	Hao
80.0	71.7	.0150	.0055	35.6	24.3
Tsi	Tso	Tco	R	Tao	Xso
70.2	89.9	66.6	9.1	76.3	39.62
Tai	Twb	Wi	Wo	Hai	Hao
80.0	67.8	.0120	.0055	32.4	25.0
Tsi	Tso	Tco	R	Tao	Xso
75.0	87.9	72.6	6.0	78.9	39.74

dehumidification is required to accomplish a fixed humidity ratio at the exit. Thus a decrease in latent load of the air requires less driving force for mass transfer. For the same equipment this can be attained by having a higher solution temperature at the inlet of the absorber. Similarly for a fixed humidity ratio and a decreasing dry bulb temperature of the air at the inlet decreases the heat transfer potential between air and sorbent solution, which decreases the sensible heat transfer taking place between the air and the sorbent solution. This causes an increase in sorbent temperature at a slower rate and thus maintains the necessary potential for mass transfer throughout the absorber.

Another partial load performance analysis of this system was conducted when 30% concentrated solution of lithium chloride is used in the absorber. The performance for the same outside air conditions as were used in the previous system is given in Table 4.4. It can be seen from this table that minimum cold water temperature to accomplish the same dehumidification performance at design conditions has gone down to 39°F (3.9°C) as compared to 55.7°F (13.2°C), which is required when the system is using a 40% solution of lithium chloride. The same trends are observed for this system which were mentioned for the system using a 40% lithium chloride solution. The system using a 30% solution in the absorber requires cold water at relatively low temperature to achieve the same dehumidification for a given outside air condition, compared to

TABLE 4.4: Performance of the liquid sorbent system using a 30% solution with an external source of chilled water.

Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.9	.0188	.0055	43.5	20.3
Tsi	Tso	Tco	R	Tao	Xso
45.8	80.3	39.0	17.7	59.8	29.60
Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.2	.0180	.0055	42.7	20.5
Tsi	Tso	Tco	R	Tao	Xso
46.9	79.9	40.4	16.9	60.4	29.63
Tai	Twb	Wi	Wo	Hai	Hao
95.0	75.9	.0150	.0055	39.3	21.1
Tsi	Tso	Tco	R	Tao	Xso
51.4	78.5	46.1	13.9	63.0	29.72
Tai	Twb	Wi	Wo	Hai	Hao
95.0	72.3	.0120	.0055	36.0	21.7
Tsi	Tso	Tco	R	Tao	Xso
55.6	76.9	51.5	10.9	65.4	29.81
Tai	Twb	Wi	Wo	Hai	Hao
90.0	78.8	.0188	.0055	42.3	20.2
Tsi	Tso	Tco	R	Tao	Xso
46.4	79.3	40.0	16.9	59.4	29.60
Tai	Twb	Wi	Wo	Hai	Hao
90.0	74.5	.0150	.0055	38.1	21.0
Tsi	Tso	Tco	R	Tao	Xso
51.9	77.4	46.9	13.1	62.5	29.72

Tai	Twb	Wi	Wo	Hai	Hao
90.0	70.9	.0120	.0055	34.8	21.6
Tsi	Tso	Tco	R	Tao	Xso
56.1	75.8	52.3	10.1	64.8	29.81
Tai	Twb	Wi	Wo	Hai	Hao
90.0	68.4	.0100	.0055	32.6	21.9
Tsi	Tso	Tco	R	Tao	Xso
58.7	74.6	55.6	8.1	66.2	29.87
Tai	Twb	Wi	Wo	Hai	Hao
85.0	73.1	.0150	.0055	36.9	20.9
Tsi	Tso	Tco	R	Tao	Xso
52.6	76.4	47.9	12.2	62.1	29.72
Tai	Twb	Wi	Wo	Hai	Hao
85.0	69.5	.0120	.0055	33.6	21.4
Tsi	Tso	Tco	R	Tao	Xso
56.6	74.7	53.0	9.3	64.3	29.81
Tai	Twb	Wi	Wo	Hai	Hao
80.0	71.7	.0150	.0055	35.6	20.8
Tsi	Tso	Tco	R	Tao	Xso
53.1	75.3	48.7	11.4	61.6	29.72
Tai	Twb	Wi	Wo	Hai	Hao
80.0	67.8	.0120	.0055	32.4	21.3
Tsi	Tso	Tco	R	Tao	Xso
57.1	73.6	53.8	8.5	63.8	29.81

the cold water temperature required by this system when a 40% solution is used in the absorber. Figure 4.5 through 4.8 are plots of humidity ratio profiles, temperature profiles for air and lithium chloride solution and air enthalpy profiles as a function of absorber height for two different solution concentration used.

To achieve a constant humidity ratio of process air it was decided to analyze another system having different values of controlling parameters describing the heat and mass transfer in the absorber, to observe the effect on performance of liquid sorbent system. For this analysis the controlling parameters C_3 and C_4 were assigned the same value which have been used in the previous analysis, and to improve the mass transfer in the absorber C_1 was given a value of 4, which implies that the value of C_2 is also 4 because C_4 is unity. This system with these new values of controlling parameters was subjected to partial load performance analysis. The same configuration is used as for the previous system i.e. the cooling water is provided by some external source of refrigeration and 40% lithium chloride solution is used in the absorber.

The performance of this system is shown in Table 4.5. As can be seen from this table that 68.1°F (20°C) cold water is required to dehumidify the outside air at 95°F (35°C) dry bulb and 0.0188 lb/lb (kg/kg) humidity ratio to a humidity ratio of 0.0055 lb/lb (kg/kg). To accomplish this same performance the previous system needs cold water at 55.7°F (13.2°C). Further for every other operating condition this

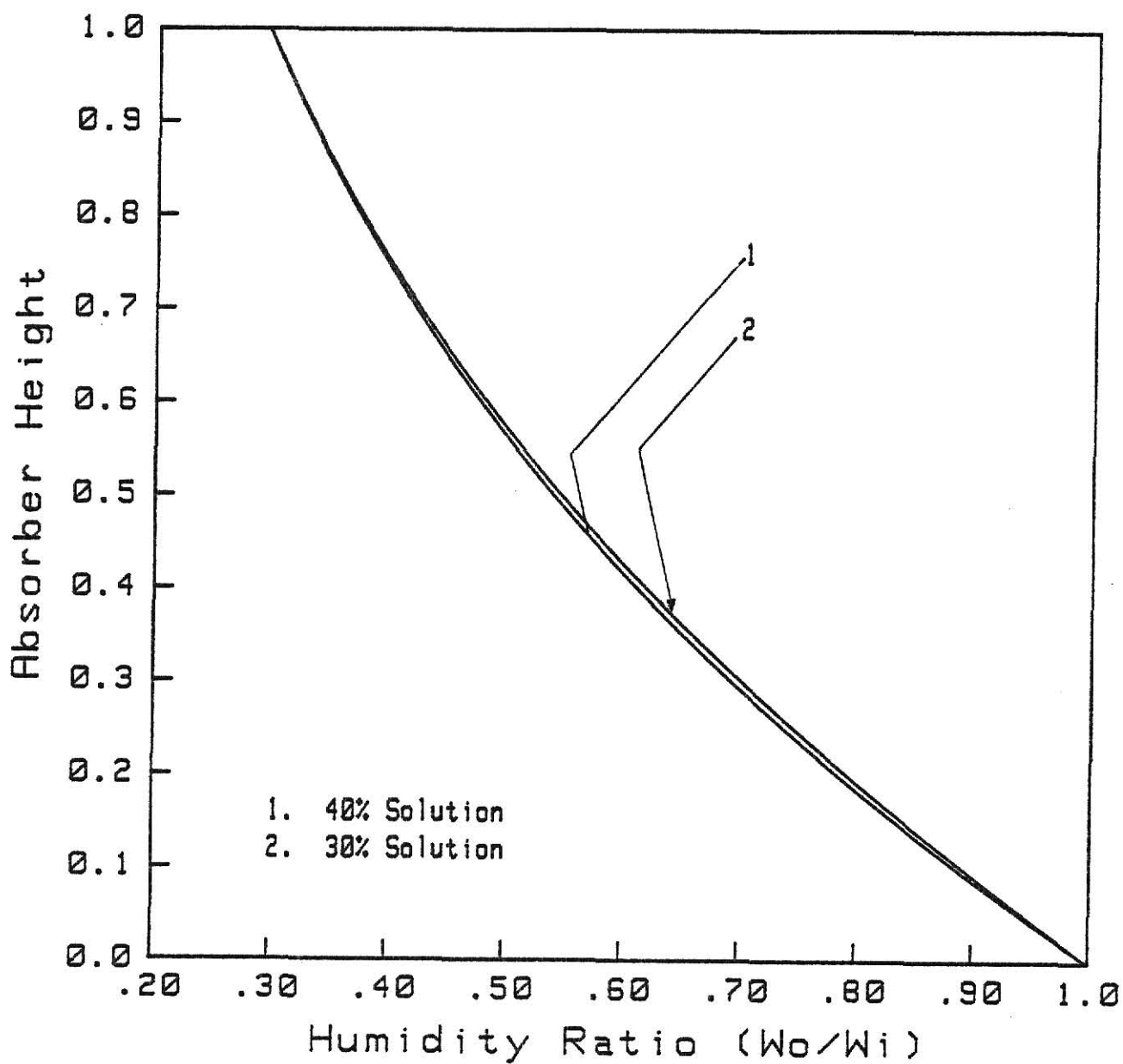


Figure 4.5: Humidity ratio profile of air in the absorber for fixed moisture content of the air at the exit.

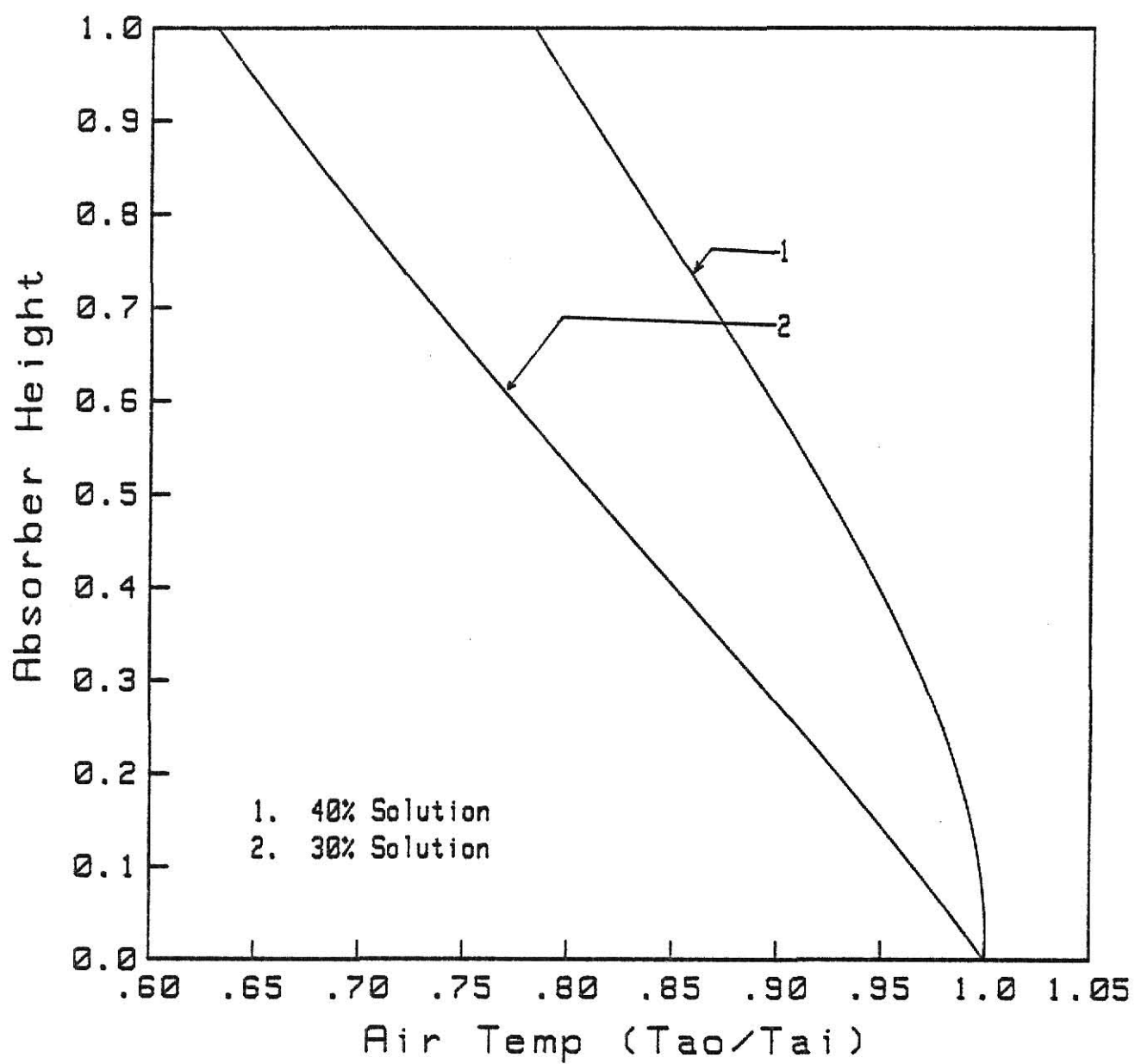


Figure 4.6: Air temperature profile in the absorber for fixed moisture content of the air at the exit.

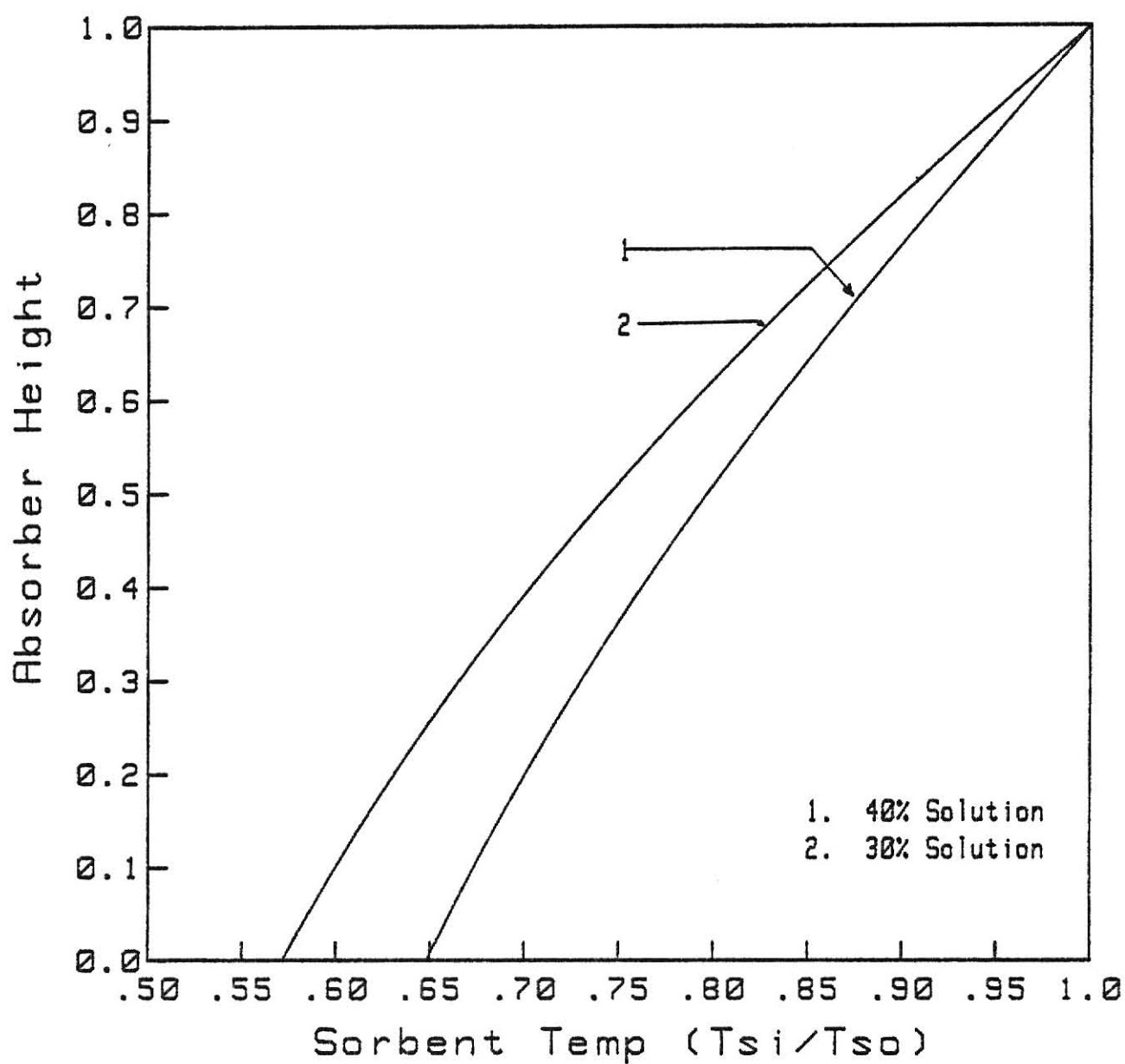


Figure 4.7: Sorbent solution temperature profile in the absorber for fixed moisture content of the air at the exit.

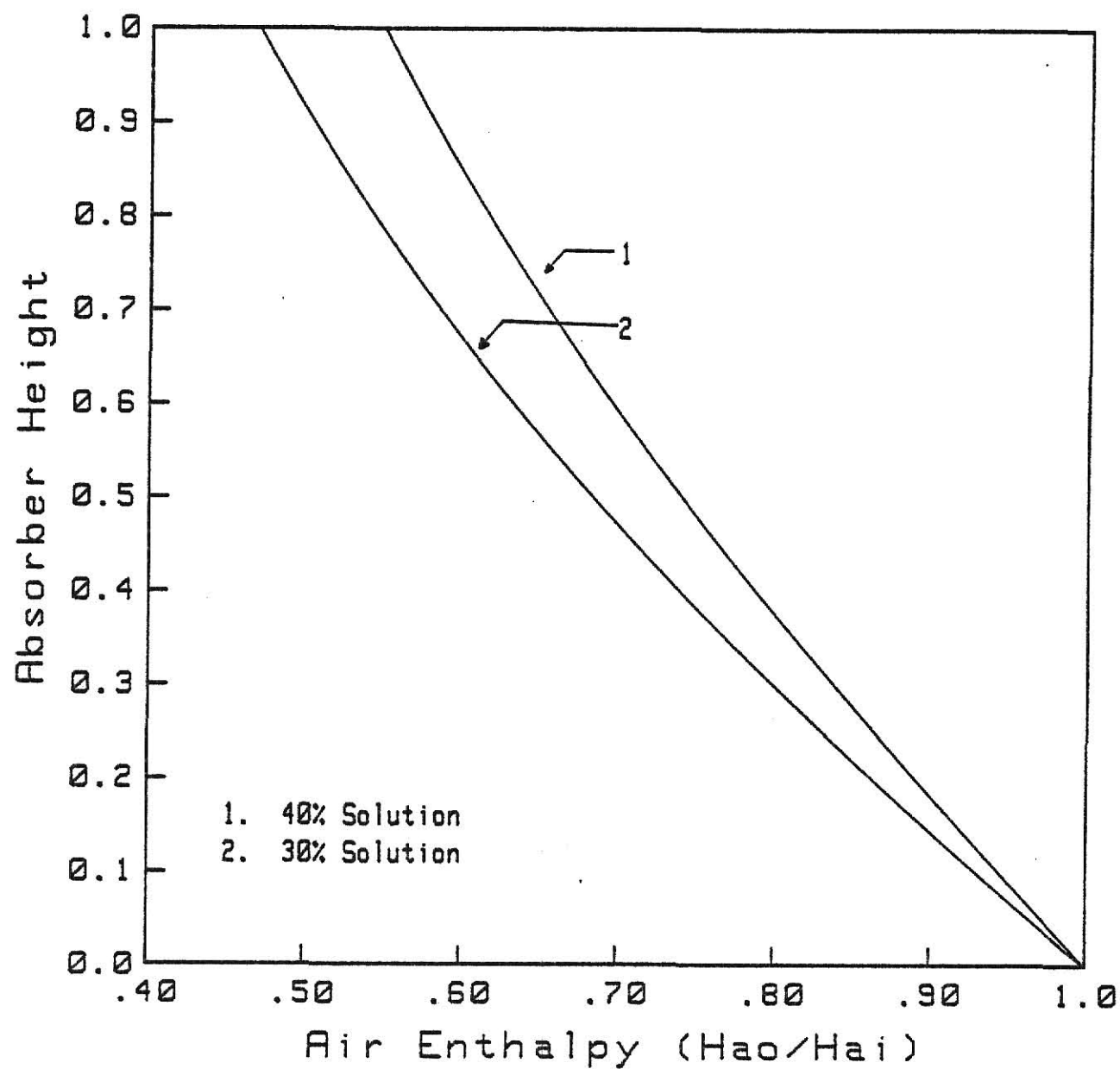


Figure 4.8: Air enthalpy profile in the absorber for fixed moisture content of the air at the exit.

TABLE 4.5: Performance of the liquid sorbent system using a 40% solution for different controlling parameters.

Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.9	.0188	.0055	43.5	25.5
Tsi	Tso	Tco	R	Tao	Xso
73.8	105.0	68.1	14.4	80.9	39.47
Tai	Twb	Wi	Wo	Hai	Hao
95.0	79.2	.0180	.0055	42.7	25.5
Tsi	Tso	Tco	R	Tao	Xso
74.5	104.0	69.1	13.6	81.2	39.51
Tai	Twb	Wi	Wo	Hai	Hao
95.0	75.9	.0150	.0055	39.3	25.6
Tsi	Tso	Tco	R	Tao	Xso
76.5	100.0	72.2	10.9	81.8	39.62
Tai	Twb	Wi	Wo	Hai	Hao
95.0	72.3	.0120	.0055	36.0	25.8
Tsi	Tso	Tco	R	Tao	Xso
78.5	96.0	75.4	8.1	82.6	39.74
Tai	Twb	Wi	Wo	Hai	Hao
90.0	79.8	.0188	.0055	42.3	25.5
Tsi	Tso	Tco	R	Tao	Xso
74.4	103.6	69.1	13.5	80.9	39.47
Tai	Twb	Wi	Wo	Hai	Hao
90.0	74.5	.0150	.0055	38.1	25.7
Tsi	Tso	Tco	R	Tao	Xso
77.1	98.6	73.2	9.9	81.8	39.62

Tai	Twb	Wi	Wo	Hai	Hao
90.0	70.9	.0120	.0055	34.8	25.9
Tsi	Tso	Tco	R	Tao	Xso
79.2	94.6	76.4	7.1	82.6	39.74
Tai	Twb	Wi	Wo	Hai	Hao
90.0	68.4	.0100	.0055	32.6	25.9
Tsi	Tso	Tco	R	Tao	Xso
80.3	91.8	78.2	5.3	82.9	39.82
Tai	Twb	Wi	Wo	Hai	Hao
85.0	73.1	.0150	.0055	36.9	25.6
Tsi	Tso	Tco	R	Tao	Xso
77.5	97.1	73.9	9.1	81.7	39.62
Tai	Twb	Wi	Wo	Hai	Hao
85.0	69.5	.0120	.0055	33.6	25.8
Tsi	Tso	Tco	R	Tao	Xso
79.6	93.1	77.1	6.2	82.5	39.74
Tai	Twb	Wi	Wo	Hai	Hao
80.0	71.7	.0150	.0055	35.6	25.7
Tsi	Tso	Tco	R	Tao	Xso
78.1	95.7	74.9	8.1	81.7	39.62
Tai	Twb	Wi	Wo	Hai	Hao
80.0	67.8	.0120	.0055	32.4	25.8
Tsi	Tso	Tco	R	Tao	Xso
80.0	91.6	77.9	5.4	82.3	39.74

system needs cold water at higher temperature than the previous system to achieve the same dehumidification performance. Also for the same outside air conditions, a comparison of Tables (4.3) and (4.5) show that for any particular outside air condition the temperature of air at absorber exit is higher for this system as compared to the previous system. This happens because the solution temperature at absorber inlet is higher in this system which would decrease the mass transfer potential to achieve the dehumidification more efficiently with less sensible cooling of air, while in the previous system due to lower inlet sorbent temperature the solution is able to take care of the same dehumidification but more sensible load of air. The plots of humidity ratio distribution, air enthalpy distribution, temperature distribution of air and temperature distribution of lithium chloride solution for this system and for the system previously analyzed with C_1 value of 2.5 when 40% and 30% solution is used in the absorber are shown in Figure 4.9 through Figure 4.16. The partial load performance of this system with a modified configuration was also analyzed. The schematic diagram of this system with modified configuration is shown in Figure 4.17. In this configuration a cooling tower is also used with an external refrigeration source. The cooling tower is used to cool the water from the cooler whenever favourable conditions are existing. The range of R in this analysis which is the load of cooler is divided into two components R_a and R_c where R_a is the range taken care

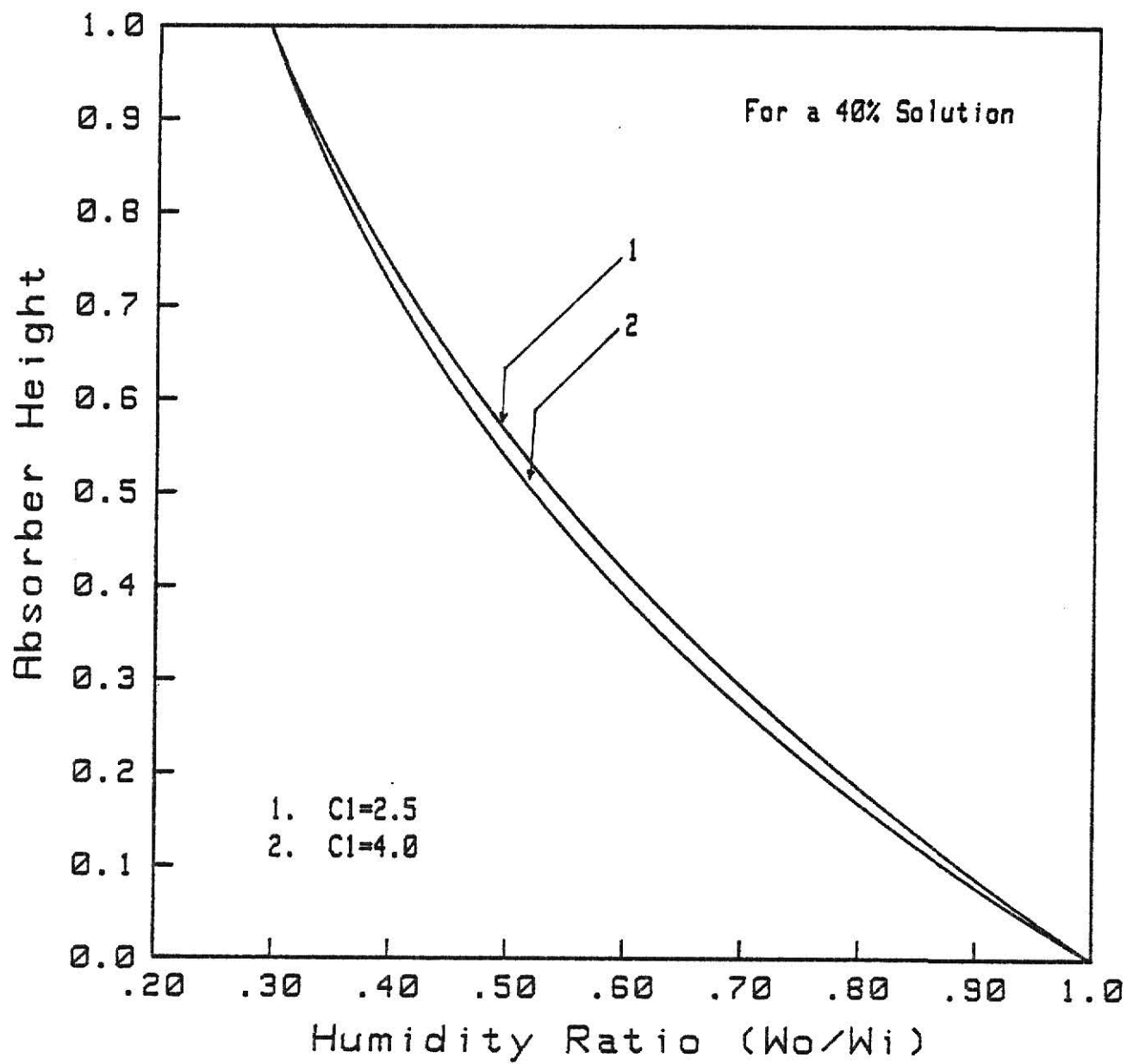


Figure 4.9: Humidity Ratio distribution of air in the absorber using different controlling parameters.

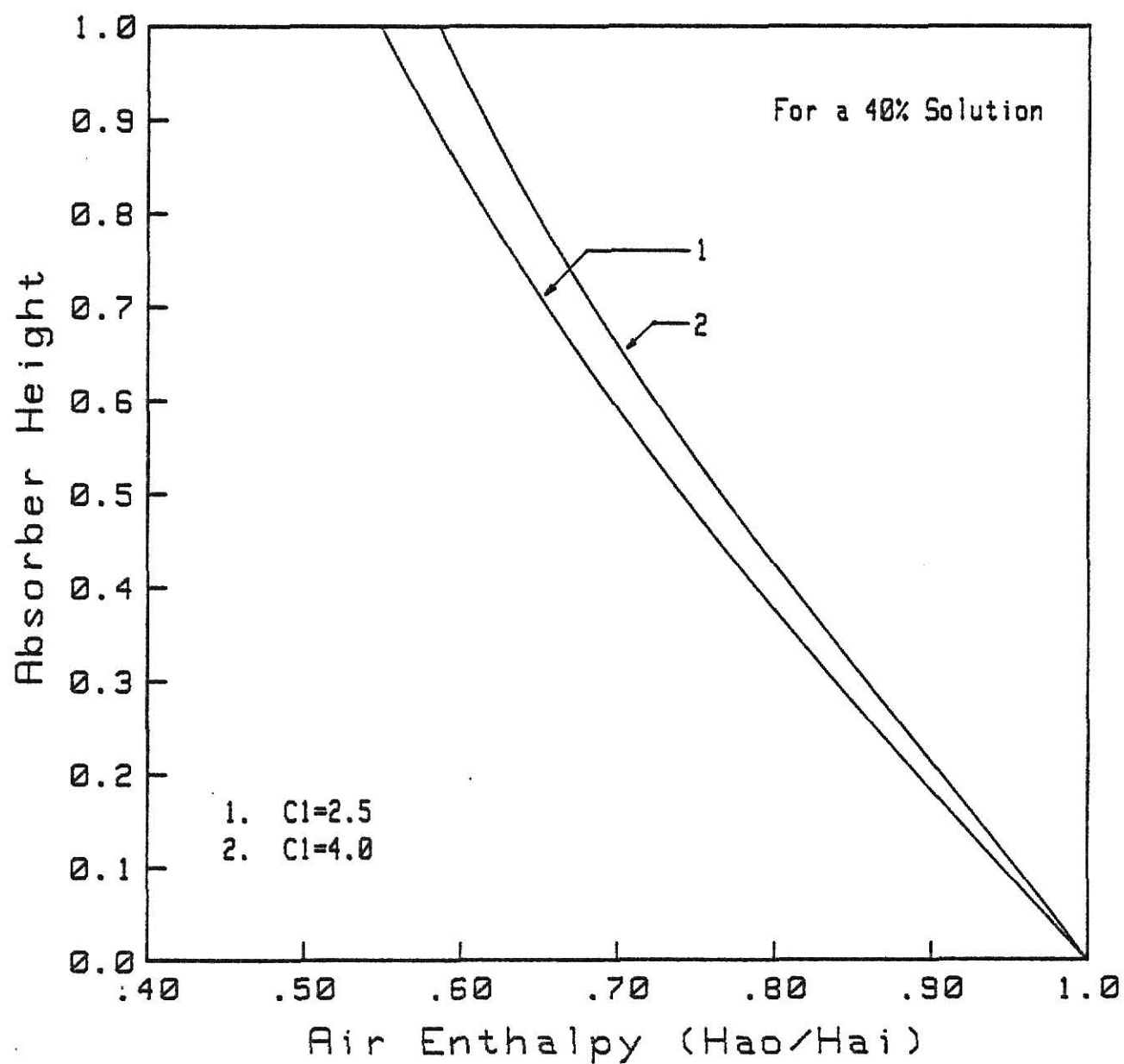


Figure 4.10: Air enthalpy distribution in the absorber using different controlling parameters.

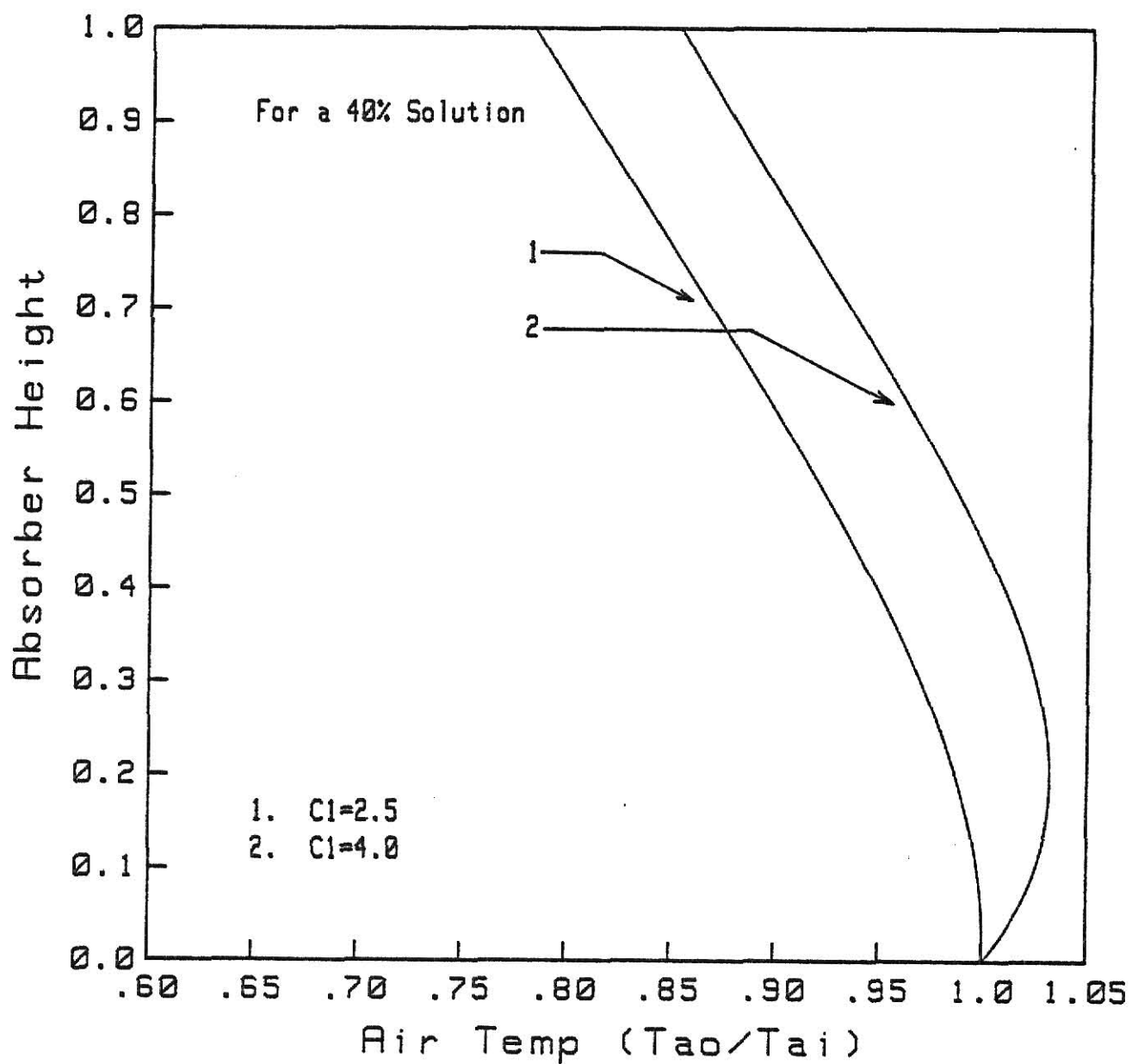


Figure 4.11: Air temperature distribution in the absorber using different controlling parameters.

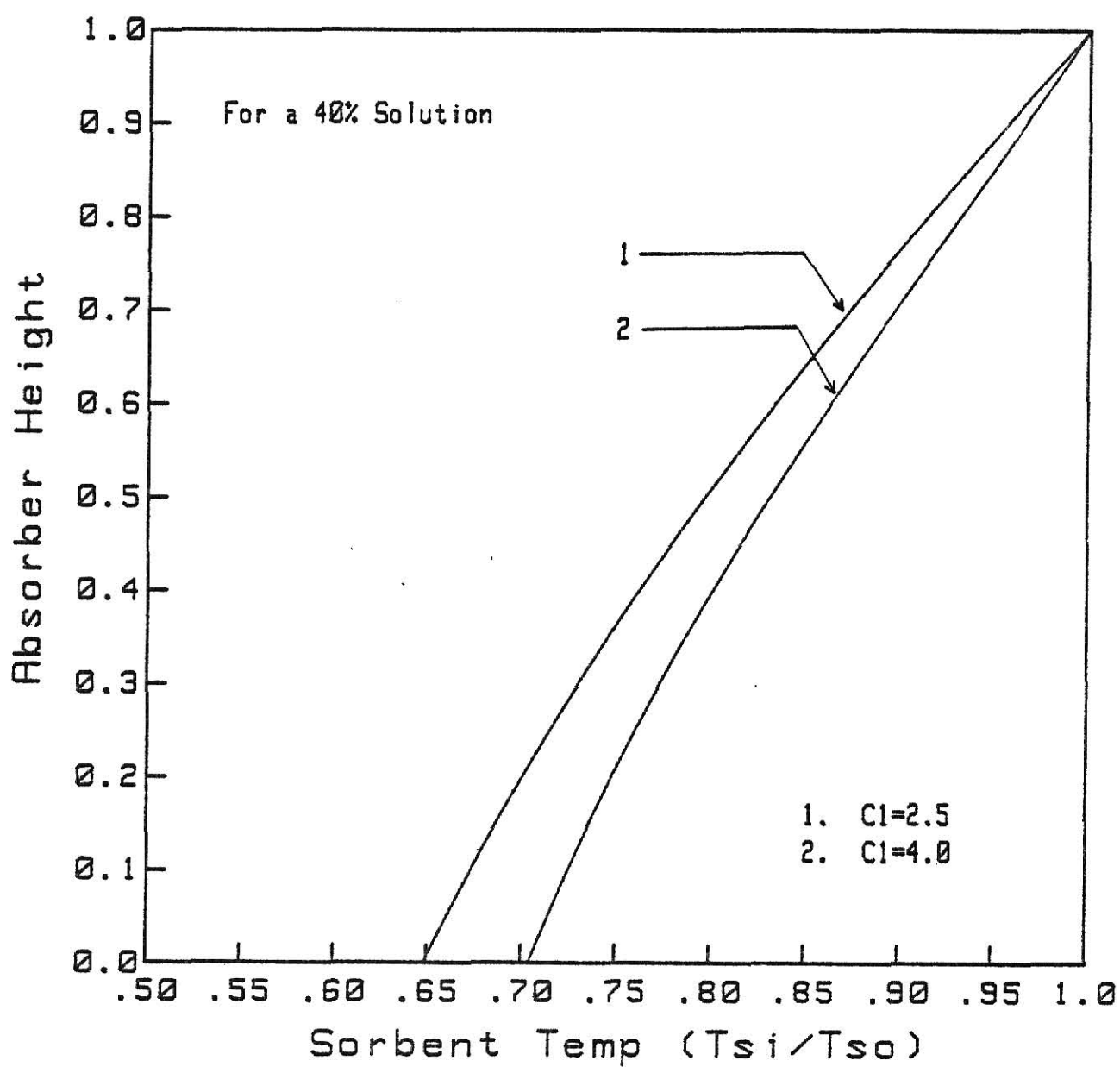


Figure 4.12: Sorbent solution temperature distribution in the absorber using different controlling parameters.

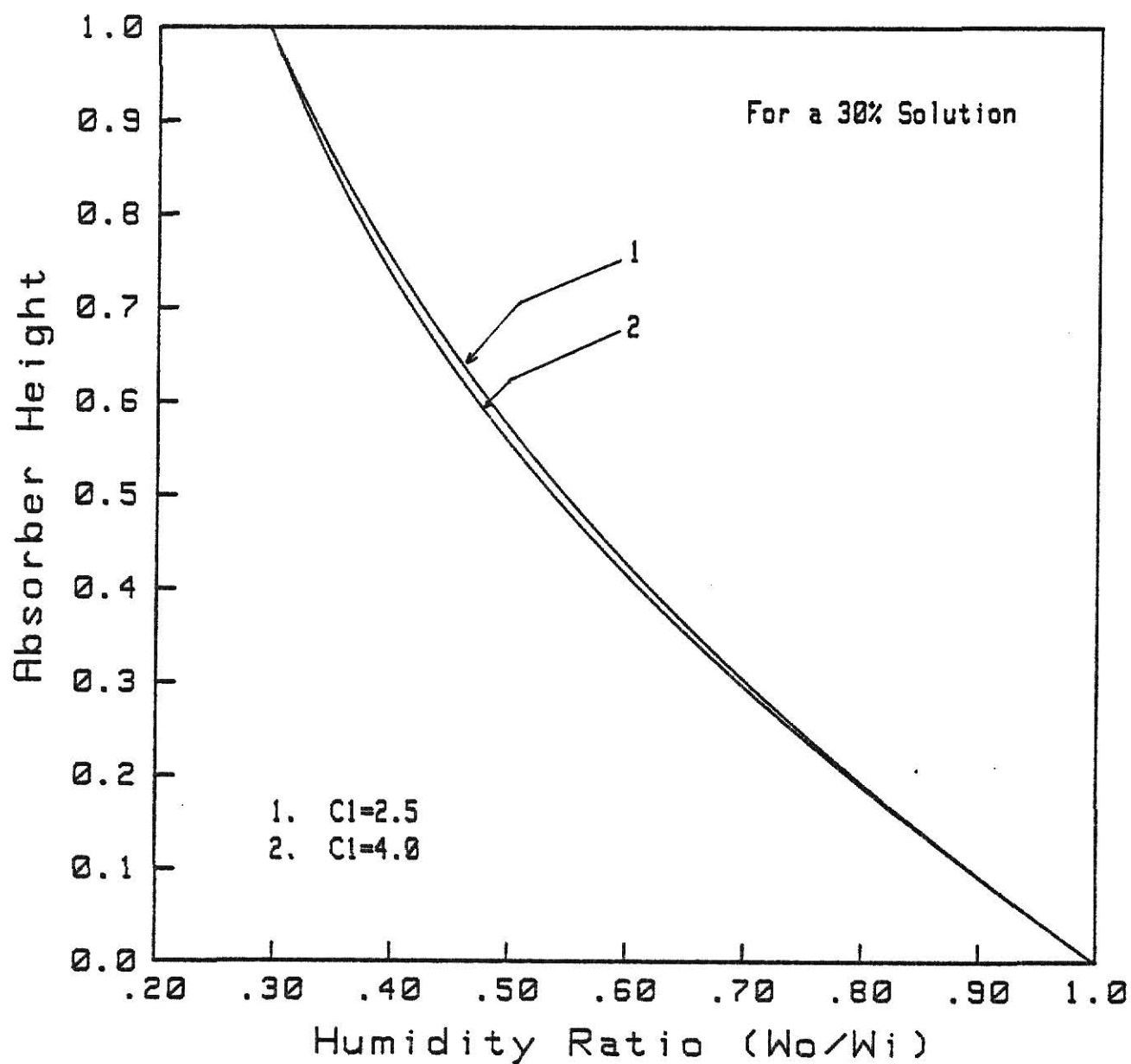


Figure 4.13: Humidity ratio distribution of air in the absorber using different controlling parameters.

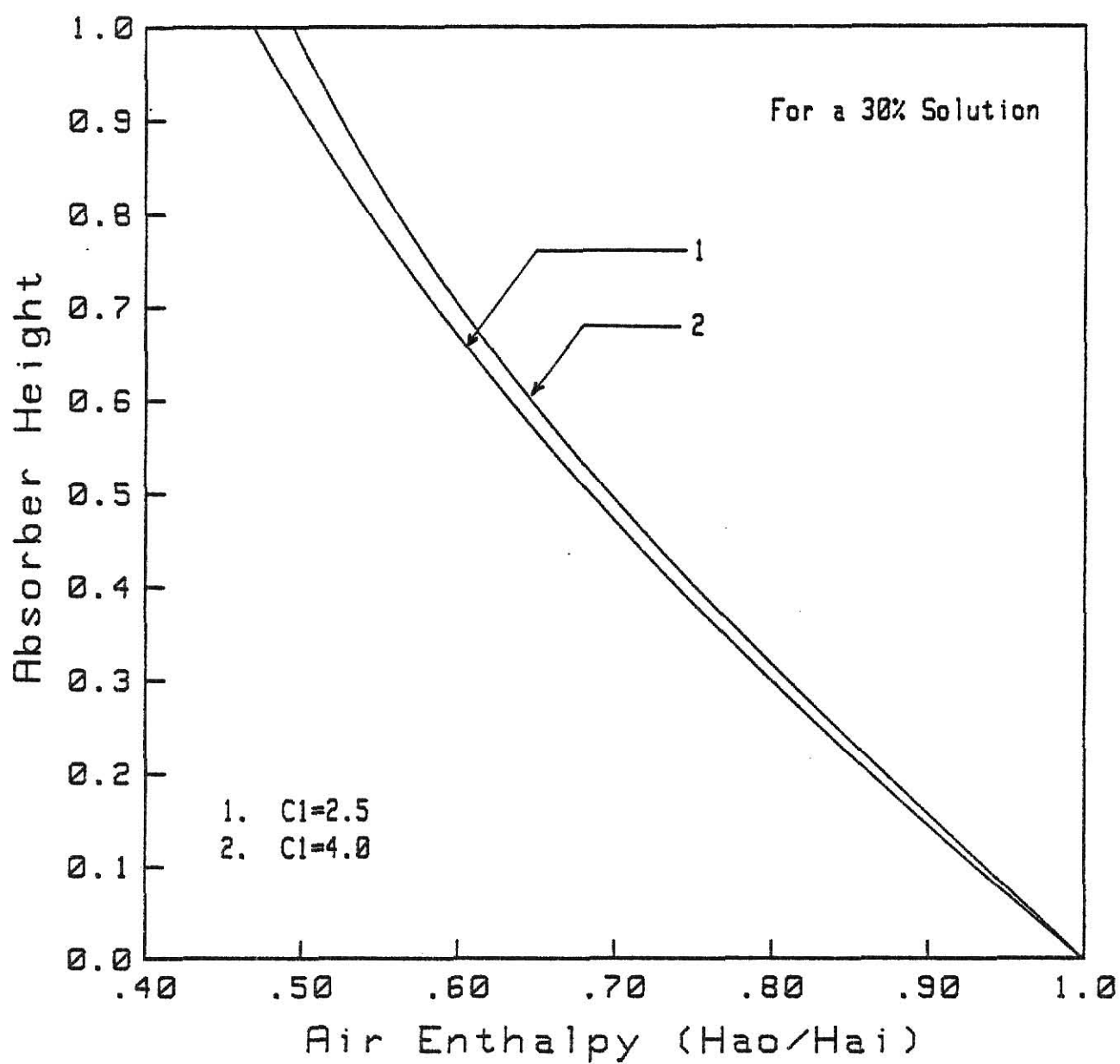


Figure 4.14: Air enthalpy distribution in the absorber using different controlling parameters.

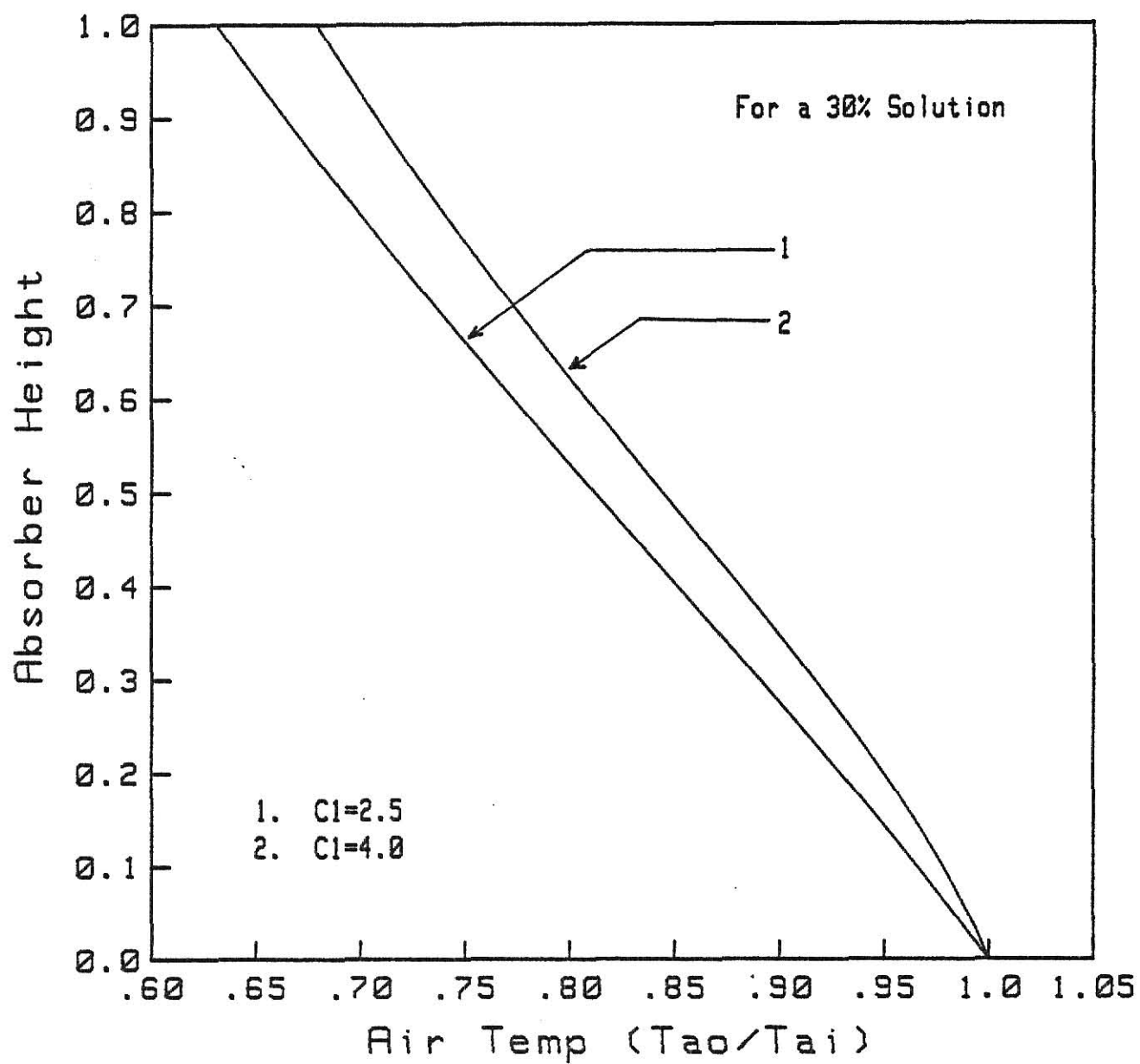


Figure 4.15: Air temperature distribution in the absorber using different controlling parameters.

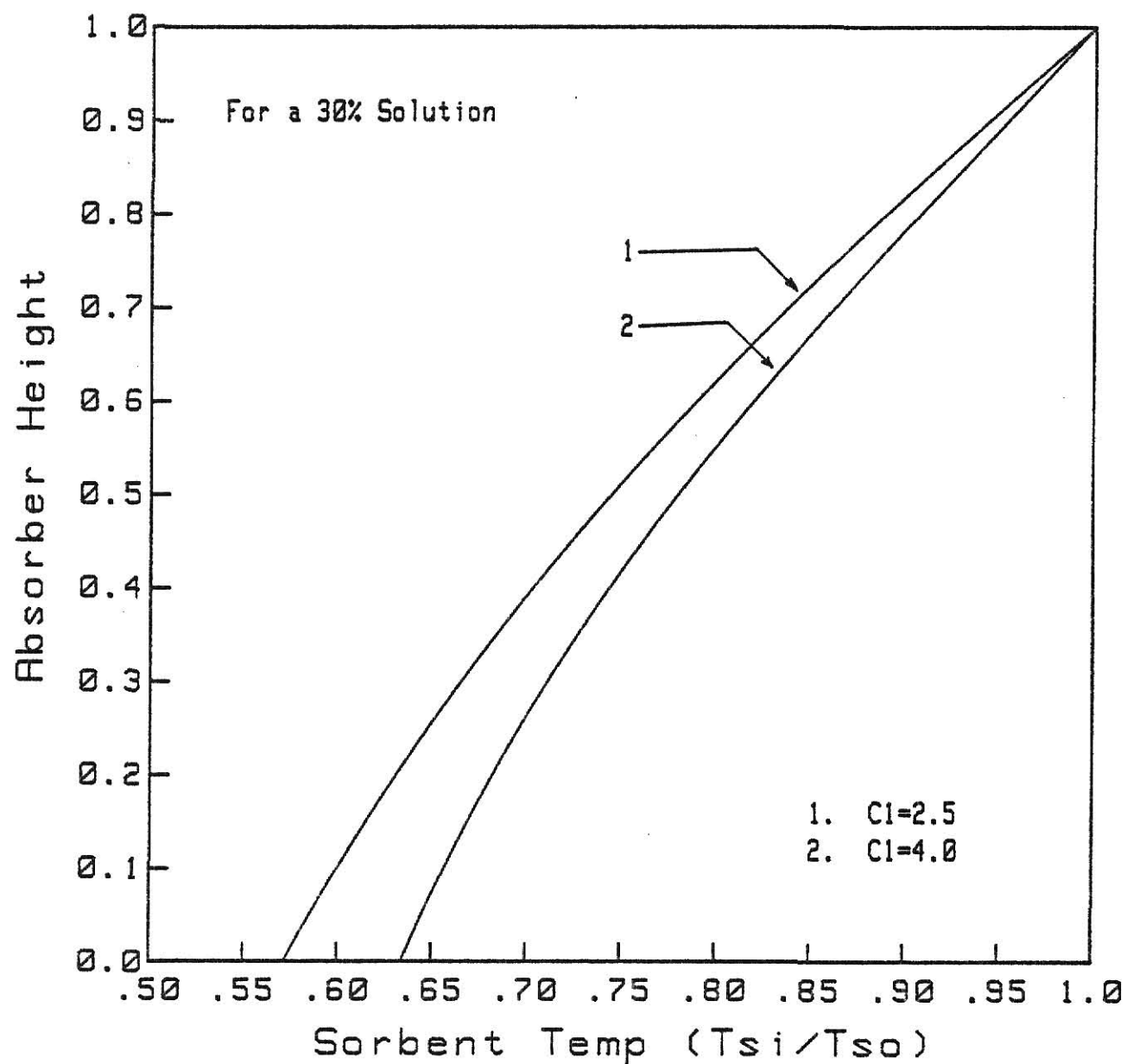


Figure 4.16: Sorbent solution temperature distribution in the absorber using different controlling parameters.

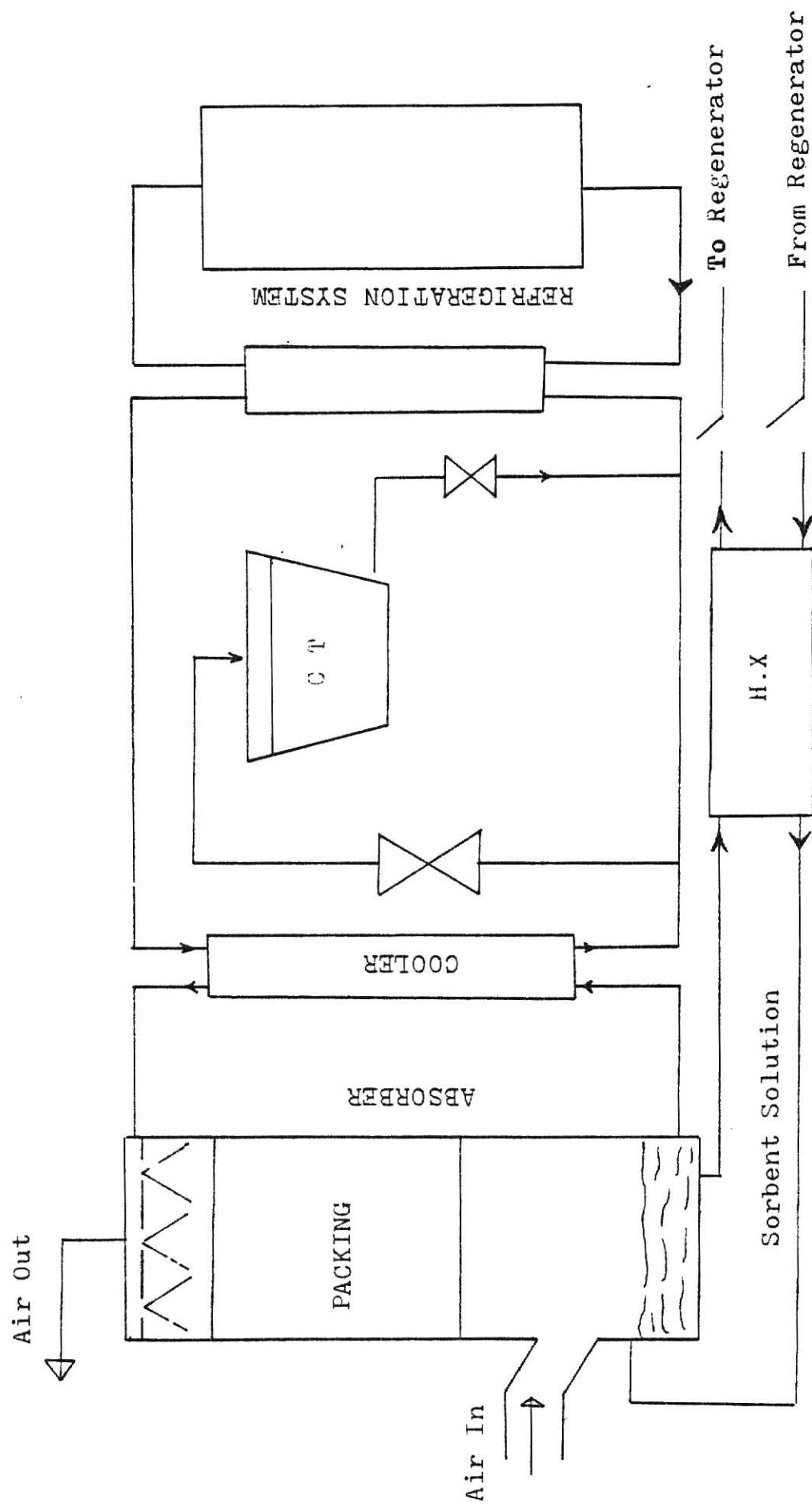


Figure 4.17: Schematic diagram of a liquid sorbent system with modified configuration.

by the refrigeration source and R_c is the range taken by cooling tower. The partial load performance of this system using 40% and 30% concentrated lithium chloride solution in absorber is presented in Tables 4.6 and 4.7.

4.3 Regenerator Analysis

It was mentioned earlier that the mathematical model developed in chapter III can also be used to analyze the performance of a regenerator. For the regenerator performance analysis it was assumed that 25% lithium chloride solution is regenerated enough so as to maintain the sorbent concentration at a fixed value. Thus, the amount of moisture to be removed in the regenerator depends on the amount of moisture added in the absorber. The mathematical model presented in Chapter III was used to predict the regenerator performance. Return air from facility at 75°F (23.9°C) dry bulb and 0.012 lb/lb (kg/kg) humidity ratio was used for regeneration purposes. A computer program similar to the one used to predict absorber performance was written. A few computer runs were performed to determine the values of the controlling parameters, which would result in efficient regenerator performance at design conditions. The following values of controlling parameters were used in regenerator performance evaluation:

TABLE 4.6: Performance of the liquid sorbent system with modified configuration using a 40% solution.

Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	79.9	.0188	.0055	43.5	25.4	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
73.8	105.0	82.5	68.1	80.9	39.47	14.4
Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	79.2	.0180	.0055	42.7	25.5	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
74.5	104.0	82.7	69.1	81.2	39.51	13.6
Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	75.9	.0150	.0055	39.3	25.6	1.3
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
76.5	100.0	83.0	72.2	81.8	39.62	9.5
Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	72.3	.0120	.0055	36.0	25.8	2.3
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
78.5	96.0	83.4	75.4	82.6	39.74	5.7
Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	78.8	.0188	.0055	42.3	25.5	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
74.4	103.6	82.6	69.1	80.9	39.47	13.5
Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	74.5	.0150	.0055	38.1	25.7	1.7
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
77.1	98.6	83.1	73.2	81.8	39.62	8.2

Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	70.9	.0120	.0055	34.8	25.9	2.6
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
79.2	94.6	83.5	76.4	82.6	39.74	4.5
Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	68.4	.0100	.0055	32.6	25.9	2.8
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
80.3	91.8	83.5	78.2	82.9	39.82	2.5
Tai	Twb	Wi	Wo	Hai	Hao	Rc
85.0	73.1	.0150	.0055	36.9	25.6	1.9
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
77.5	97.1	82.9	73.9	81.6	39.62	7.1
Tai	Twb	Wi	Wo	Hai	Hao	Rc
85.0	69.5	.0120	.0055	33.6	25.8	2.6
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
79.6	93.1	83.4	77.1	82.5	39.74	3.6
Tai	Twb	Wi	Wo	Hai	Hao	Rc
80.0	71.7	.0150	.0055	35.6	25.7	2.2
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
78.1	95.7	83.0	74.9	81.7	39.62	5.9
Tai	Twb	Wi	Wo	Hai	Hao	Rc
80.0	67.8	.0120	.0055	32.4	25.8	2.7
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
80.0	91.6	83.2	77.9	82.3	39.74	2.7

TABLE 4.7: Performance of the liquid sorbent system with modified configuration using a 30% solution.

Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	79.9	.0188	.0055	43.5	21.4	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
56.9	89.9	67.3	50.4	64.4	29.61	16.9
Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	79.2	.0180	.0055	42.7	21.5	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
57.4	89.0	67.4	51.2	64.6	29.63	16.2
Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	75.9	.0150	.0055	39.3	21.7	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
59.3	85.6	67.6	54.1	65.4	29.72	13.5
Tai	Twb	Wi	Wo	Hai	Hao	Rc
95.0	72.3	.0120	.0055	36.0	21.8	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
61.0	82.1	67.7	56.9	66.0	29.81	10.8
Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	70.8	.0188	.0055	42.3	21.4	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
57.4	88.6	67.2	51.3	64.3	29.60	16.0
Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	74.5	.0150	.0055	38.1	21.7	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
59.8	84.3	67.5	55.0	65.3	29.72	12.5

Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	70.9	.0120	.0055	34.8	21.9	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
61.5	80.8	67.6	57.7	66.0	29.81	9.9
Tai	Twb	Wi	Wo	Hai	Hao	Rc
90.0	68.4	.0100	.0055	32.6	21.9	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
62.6	78.4	67.6	59.5	66.3	29.87	8.1
Tai	Twb	Wi	Wo	Hai	Hao	Rc
85.0	73.1	.0150	.0055	36.9	21.6	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
60.0	82.9	67.3	55.6	65.1	29.72	11.7
Tai	Twb	Wi	Wo	Hai	Hao	Rc
85.0	69.5	.0120	.0055	33.6	21.8	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
61.8	79.4	67.3	58.3	65.8	29.81	9.0
Tai	Twb	Wi	Wo	Hai	Hao	Rc
80.0	71.7	.0150	.0055	35.6	21.6	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
60.6	81.6	67.2	56.4	65.1	29.72	10.8
Tai	Twb	Wi	Wo	Hai	Hao	Rc
80.0	67.8	.0120	.0055	32.4	21.8	0.0
Tsi	Tso	Tci	Tco	Tao	Xso	Ra
62.3	78.1	67.3	59.2	65.8	29.81	8.1

$$C_1 = 2.0$$

$$C_2 = 1.4$$

$$C_3 = 0.5$$

$$C_4 = C_1/C_2$$

The amount of moisture to be removed in the regenerator is a function of amount of moisture added in the absorber, and because the absorber operates at a fixed concentration of the solution, the regeneration requirements float with the absorber performance. The computer model for regenerator is very similar to the model used for absorber, which has been explained earlier. The same approach was used to analyze regenerator performance.

The iterative scheme which is used in the computer model for the regenerator, starts by first, guessing the outlet sorbent solution temperature from the regenerator, so as to determine the temperature distribution of solution in the regenerator. The calculation cycle continues by changing this temperature in either direction until required concentration at the regenerator outlet is accomplished. The hot water temperature t_{hi} required to heat the solution is then calculated. The performance of this regenerator for various regeneration requirements when 30%, 35% and 40% lithium chloride solution is required in the absorber is given in Table 4.8. A plot of moisture removal from

TABLE 4.8: Regeneration performance of the system for different solution requirements.

Tagi	Wagi	Xgi	Xgo	Tagi	Thi
75.0	.0120	39.5	41.7	249.1	***
75.0	.0120	39.5	41.6	243.0	***
75.0	.0120	39.6	41.5	230.4	***
75.0	.0120	39.8	40.8	183.5	194.5
75.0	.0120	39.8	40.6	165.9	174.5
Tagi	Wagi	Xgi	Xgo	Tagi	Thi
75.0	.0120	29.6	31.3	211.7	***
75.0	.0120	29.6	31.2	203.2	***
75.0	.0120	29.8	30.9	176.4	189.0
75.0	.0120	29.8	30.6	153.7	162.8
75.0	.0120	29.8	30.4	139.4	146.4
Tagi	Wagi	Xgi	Xgo	Tagi	Thi
75.0	.0120	34.5	36.5	227.8	***
75.0	.0120	34.6	36.4	221.0	***
75.0	.0120	34.6	36.0	197.9	***
75.0	.0120	34.8	35.7	166.4	176.3
75.0	.0120	34.8	35.5	152.7	160.6

***The temperature of water exceeds 212°F.

the solution versus regeneration temperature for a fixed 30%, 35%, and 40% concentrated solution requirement in the absorber is shown in Figure 4.18.

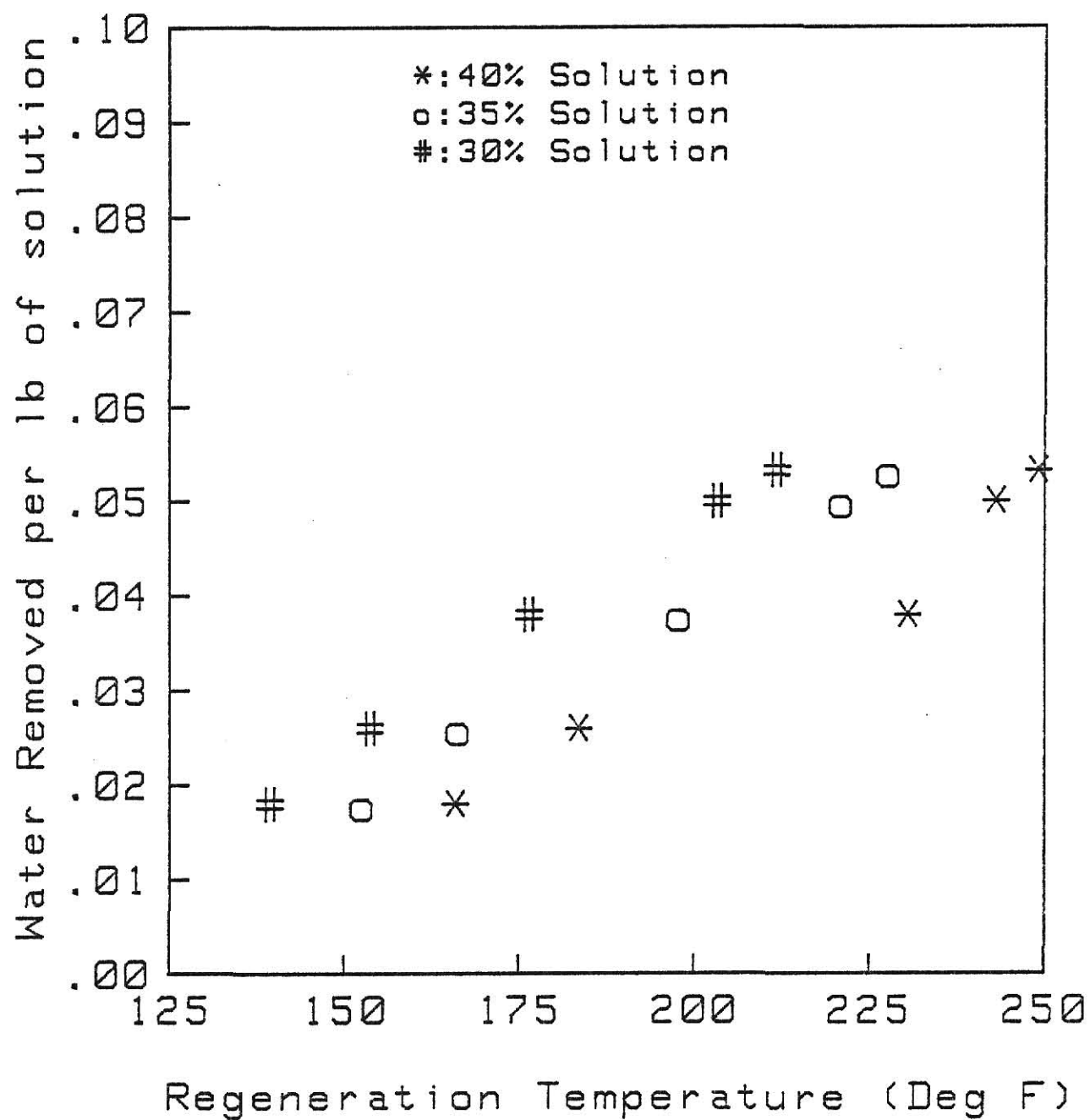


Figure 4.18: Plot of moisture removal vs. regeneration temperature for different solution concentration.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

In the previous chapters a liquid desiccant system was modeled. The system was analyzed when subjected to various air conditions less severe than the design conditions. The conclusions which can be drawn from the performance analysis of liquid sorbent systems with different cooling sources employed are:

1. Cooling and dehumidification performance of liquid sorbent system, when cooling of the solution is furnished by the cooling tower will supply air at a humidity ratio less than the design humidity ratio when operating at partial loads. To have the supply air at a certain fixed condition it is necessary to humidify as well as cool/heat this air to accomplish appropriate supply conditions. This can be achieved by evaporatively cooling the air and then sensibly cooling or heating it to obtain supply conditions. This would mean that the liquid sorbent system will be oversized for any air condition other than the design condition.
2. Liquid sorbent systems having higher mass transfer coefficient or more packing (more surface area) can achieve same dehumidification with cooling water at relatively higher temperatures to cool the sorbent.

3. Liquid sorbent system is not a good cooler and in order to accomplish low dry bulb temperature of process air by this system alone, the air has to be dehumidified more than required, and then through evaporative cooling it can be cooled to supply air conditions. This would require larger transfer surface which would result in larger equipment size and higher pumping power requirements.
4. It can be inferred from the performance tables of different liquid sorbent systems that natural sources of cooling water or systems generating cold water at higher temperatures can serve the purpose of cooling sorbent solution when these sorbent systems are operating at partial loads, because as the latent load of outside air decreases or the dry bulb of this air goes down the cooling water is required at higher temperatures to accomplish certain exit air conditions. This certainly would decrease the energy requirements of a liquid sorbent system.
5. Regeneration temperature of the sorbent solution which depends on the concentration and amount of moisture absorbed, is not very high for ordinary application and therefore low grade thermal energy can be used for regeneration, especially at partial loads when dehumidification demand to achieve a specific process air condition decreases.

5.2 Recommendations

To develop a method to select an optimum cooling/dehumidification system for day in and day out performance, it is necessary that the performance of the system when operating at partial loads should be known. In order to predict the performance it is necessary that the system used should be modeled to analyse any operating conditions. The systems which are capable of accomplishing cooling/dehumidification can be mechanical refrigeration system, absorption refrigeration system or desiccant system, using any of the above systems, alone or in combination can accomplish cooling and dehumidification. Thus the choice of method or methods is based on

1. Process air inlet temperature and humidity
2. Process air temperature and humidity desired at outlet
3. Regeneration temperature in case of desiccant systems, or condenser heat sink temperature in case of refrigeration systems.

To develop a uniform procedure for determining the optimum system for both design load and seasonal operation, it is recommended that

- a. Mathematical model should be developed to predict the performance of other individual systems.
- b. These systems should be optimized at certain design conditions.
- c. The performance of these systems when operating at off design conditions should be analyzed.

- d. Annual energy requirements of these systems should be determined by using weather data for a certain location and application. Finally having annual energy consumptions, it's availability and cost, equipment costs, can be used as inputs to the procedure of optimization.
5. The results obtained from mathematical model can be checked experimentally to detect shortcomings or limitations if any, in the models relative to actual test data.

BIBLIOGRAPHY

1. Rodda, E. D., "Humidity Controlled Drying," Grain Conditioning Conference Proceedings, University of Illinois, 1975.
2. Danziger, M. T., et al., "Drying of field corn with Silica Gel," Trans. ASAE, vol. 15, 1972, pp. 1071-1074.
3. Pla Barby, F. E., "Solid Desiccant Air Conditioning with Silica Gel Using Solar Energy," Ph. D. Dissertation, University of Texas, Austin, 1978.
4. Bichowsky, F. R., "Chemical Dehumidification Agents," Heating, Piping and Air Conditioning, October 1940, pp. 627-632.
5. Gifford, E. W., "Dehumidification by Liquid Desiccants," Heating, Piping and Air Conditioning, pp. 156-169, April, 1957.
6. Griffith, W. C., "Use of Liquid Sorption Dehumidification to Improve Energy Utilization of Air Systems," Symposium on Energy Conservation in Air Systems Through The Use of Sorption Dehumidifier Techniques," ASHRAE Transactions 1980, vol. 86, Part 1.
7. Löf, G. O. G., "Cooling with Solar Energy," 1955 Congress on Solar Energy, Tucson, Arizona, pp. 171-189, 1955.

8. Kapur, J. C., "A Report on the Utilization of Solar Energy for Refrigeration and Air Conditioning Applications," Solar Energy, vol. 4, No. 1, pp. 39-44, 1956.
9. Mullick, S. C. and Gupta, M. C., "Solar Desorption of Absorbent Solution," Solar Energy, vol. 16, No. 19, pp. 19-24, 1974.
10. Kakabaev, A. and Khandurdyyer, A., "Absorption Solar Refrigeration Unit with Open Regeneration of Solution," Geliotekhnika, vol. 5, No. 4, pp. 28-32, 1969.
11. Kakabaev, A. et al., "Regeneration Capacity of an Absorption Solar Refrigeration Plant with Flat Glazed Solution Regenerator," Geliotekhnika, vol. 8, No. 2, pp. 60-67, 1972.
12. Kakabaev, A. and Golaev, M., "Glazed Flat Surface as a Solution Regenerator for Use in an Absorption Cooling System," Geliotekhnika, vol. 7, No. 4, pp. 44-49, 1971.
13. Collier, R. K., "The Analysis and Simulation of an Open Cycle Absorption Refrigeration System," Solar Energy, vol. 23, pp. 357-366, 1979.
14. Robinson, H. I., "Liquid Sorbent Solar Air Conditioner," Alternative Energy Sources Encyclopedia, T. N. Veziroglu, Editor, Hemisphere Publishers, Washington, D. C., 1978.
15. Turner, N. C., United States Patent No. 4,171,620.

16. Peng, S. C., "The Analysis and Design of Liquid Absorbent/Desiccant Cooling/Dehumidification Systems for Low Grade Thermal Energy Applications," Ph. D. Diss., University of Texas, Austin, December, 1980.
17. Lodwig, E., Wilke, D. and Bressman, Jr., "A Solar Powered Desiccant Air Conditioning System." Solar Collectors Heating and Cooling Systems, vol. 1, Proceedings of the 1977 Annual Meeting, American Section of ISES, 1977.
18. Whitman, W. G., "A Preliminary Experimental Confirmation of the Two-Film Theory for Gas Absorption," Chem. Met. Eng., vol. 29, No. 4, 1923.
19. Treybal, R. E., "Adiabatic Gas Absorption and Stripping in Packed Towers," Industrial and Engineering Chemistry, vol. 61, No. 7, pp. 36-41, 1969.
20. Sherwood, T. K., and Pigford, R. L., Absorption and Extraction, McGraw-Hill, 1952.
21. Threlkeld, J. L., Thermal Environmental Engineering, Second Edition, Prentice-Hall, 1970.
22. Chilton, T. H., and Colburn, A. P., "Distillation and Absorption in Packed Columns," Industrial and Engineering Chemistry, vol. 27, pp. 255-260, 1935.
23. Merkel, F., Forschungsarb., vol. 275, pp. 1-48, 1925.
24. ASHRAE Handbook & Product Directory, 1979, Equipment, Figure 17, pp. 21.11.
25. Keenan, J. H., Keyes, F. G., Hill, P. G., and Moore, J. G., steam tables: Thermodynamic Properties of

Water, Including Vapor, Liquid, and Solid Phases,

John Wiley and Sons, New York, NY, 1969.

26. Uemura, Tadashi, "Studies of the Lithium Chloride -
Water Absorption Refrigerating Machine" Technology
Report of the Kansai University, No. 9, pp. 81-88,
1967.

```

10  PRINTER IS 16
30  PRINT "The input value of Dry bulb Temperature will have to be in Deg F."
40  PRINT "Humidity ratio should be dimensionless."
50  DIM Nsto(400),Wsto(400)
60  INPUT "Type d.b.t. & W, values seperated by a comma, and press CONT",Tai,Wi
61  ! GOTO 1690
70  T=(Tai-32)*5/9+273.18
80  T=T-.2
90  Tca=T-273.18
100  Tub=Tca*9/5+32
110  F(1)=-741.9242
120  F(2)=-29.721
130  F(3)=-11.55286
140  F(4)=-.6685635
150  F(5)=.1094098
160  F(6)=.439993
170  F(7)=.2520658
180  F(8)=.05218684
190  Sigma=0
200  FOR I=1 TO 8
210  Sig=F(I)*(.65-.01*Tca)^(I-1)
220  Sigma=Sigma+Sig
230  NEXT I
240  Pw=EXP(LOG(217.99)+.01/T*(374.136-Tca)*Sigma)*14.696
250  Ww=.62198*Pw/(14.696-Pw)
260  Wp=((1093-.556*Tub)*Ww-.24*(Tai-Tub))/(1093+.444*Tai-Tub)
270  IF Wp>Wi THEN 80
280  C1=2.5
290  C2=2.5
300  C3=.70000
310  Tso=95.6
320  Tso=Tso+.1
330  GOTO 350
340  Tso=Tso-.1
350  Hai=.24*Tai+Wi*(1061+.444*Tai)
360  Hao=Hai
370  Wo=Wi
380  Tsi=Tso
381  Tao=Tai
390  Xso=39.474982
400  Xsi=Xso
410  N=0
420  J=1
430  Tsia=(Tsi-32)*5/9+273.18
440  Tsic=Tsia-273.18
450  Alpha=0
460  FOR I=1 TO 8
470  Alph=F(I)*(.65-.01*Tsic)^(I-1)
480  Alpha=Alpha+Alph
490  NEXT I
500  Pws=EXP(LOG(217.99)+.01/Tsia*(374.136-Tsic)+Alpha)*14.696
510  A=7.5713+.037964*Xsi-.00090133*Xsi^2
520  B=1582.1+10.364*Xsi-.14462*Xsi^2
530  P=10^(A-B/(Tsic+230))
540  Phi=P/(Pws*51.714752)
550  Wos=Wo/Wi
560  Nstos=N/173
570  Hsto(J)=Nstos
580  Wsto(J)=Wos
590  N=N+1
600  J=J+1
610  We=.62198*Pws*Phi/(14.696-Phi*Pws)
620  Hae=.24*Tsi+We*(1061+.444*Tsi)
630  Dha=-C1*(Hao-Hae)*.005
640  Hao=Hao+Dha
650  Dwa=-C1*(Wo-We)*.005

```

```

660 Wo=Wo+Dwa
665 Tao=(Hao-1061*Wo)/(1.24+.444*Wo)
670 Dx=-C2*Dwa/C1
680 Xsi=(1+Dx)*Xsi
690 Dns=C2*Dna/C1
700 A4=1.0020-.012505*Xsi+.00007575*Xsi^2
710 B4=-5.554E-4-1.5178E-5*Xsi+6.0248E-7*Xsi^2
720 C4=5.2266E-6+3.6623E-8*Xsi-3.9345E-9*Xsi^2
730 Cp=A4+B4*Tsic+C4*Tsic^2
740 Dts=(Dns-.01*Xsi^2.5*Dx)/Cp
750 Tsi=Tsi+Dts
760 IF N>173 THEN 790
770 PRINT Hao;Wo;We;N;Tsi;Tco;Tao
780 GOTO 430
790 D=2.5*(1-Cp+C3)
800 E=(1-EXP(D))/(Cp+C3-EXP(D))
810 Tco=Tso-(Tso-Tsi)*1/E
820 Tci=Tco+C3*1.06*Cp*(Tso-Tsi)
830 R=Tci-Tco
840 IF Wo-.0055>.00002 THEN 340
850 IF .0055-Wo>.00002 THEN 320
870 PRINT "          Tai          Tub          Wi          Wo          Hai
      Hao"
880 PRINT
890 PRINT USING 940;Tai,Tub,Wi,Wo,Hai,Hao
900 PRINT
910 PRINT "          Tsi          Tso          Tco          R          Tao
      Xso"
920 PRINT
930 PRINT USING 950;Tsi,Tso,Tco,R,Tao,Xso
940 IMAGE 6X, DDD.D,9X,DD.D,8X,.DDDD,9X,.DDDD,10X,DD.D,8X,DD.D
950 IMAGE 6X,DDD.D,8X,DDD.D,9X,DD.D,10X,DD.D,9X,DDD.D,8X,DD.D
960 PRINT
970 PRINT
980 GOTO 1690
990 PRINT "          Tagi          Wagi          Xgi          Xgo          Tsgi
      Thi"
1000 PRINT
1010 ! *****
1020 ! REGENERATOR ANALYSIS STARTS HERE*
1030 ! *****
1040 C11=2
1050 C22=1.4000
1060 C33=.5
1070 Tagi=75
1080 Wagi=.012
1090 Tsgo=96.67
1100 Tsgo=Tsgo+.1
1110 GOTO 1130
1120 Tsgo=Tsgo-.1
1130 Xgo=31.189083
1140 Xgi=Xgo
1150 Hagi=.24*Tagi+Wagi*(1061+.444*Tagi)
1160 Hago=Hagi
1170 Tsgi=Tsgo
1180 Wago=Wagi
1190 N=0
1200 N=N+1
1210 Tsgic=(Tsgi-32)*5/9
1220 A1=7.5713+.037964*Xgi-.00090133*Xgi^2
1230 B1=1582.1+10.364*Xgi-.14462*Xgi^2
1240 Pa=10^(A1-B1/(Tsgic+230))
1250 Wage=.62198*Pa/(760-Pa)
1260 IF Wage>.90 THEN 1120
1270 Hage=.24*Tsgi+Wage*(1061+.444*Tsgi)
1280 Dhag=-C11*(Hago-Hage)*.005

```

```

1290 Hago=Hago+Dhag
1300 Dwag=-C11*(Wago-Wage)*.005
1310 Wago=Wago+Dwag
1320 Dxgi=-C22*Dwag/C11
1330 Xgi=(Dxgi+1)*Xgi
1340 Dhsg=C22*Dhag/C11
1350 A5=1.0020-.012505*Xgi+.00007575*Xgi^2
1360 B5=-5.554E-4-1.5178E-5*Xgi+6.8248E-7*Xgi^2
1370 C5=5.2266E-6+3.6623E-8*Xgi-3.8345E-9*Xgi^2
1380 Cp=A5+B5*Tsgic+C5*Tsgic^2
1390 Dtsgi=(Dhsg-.01*Xgi^2.5*Dxgi)/Cp
1400 Tsgic=Tsgic+Dtsgi
1410 IF N>459 THEN 1450
1420 PRINTER IS 16
1430 PRINT Tsgic;Wago;Wage;Hago;Xgi;N;Tsgo
1440 GOTO 1200
1450 D=2.5*(1-Cp*C33)
1460 E=(1-EXP(D))/(Cp*C33-EXP(D))
1470 Thi=Tsgo+(Tsgic-Tsgo)*1/E
1480 Rh=Cp*(Tsgic-Tsgo)
1490 Tago=(Hago-1061*Wago)/(1.24+.444*Wago)
1500 PRINT Tago
1510 PRINTER IS 0
1520 IF Xgi-29.63>.05 THEN 1100
1530 IF 29.63-Xgi>.05 THEN 1120
1540 Tagoc=(Tago-32)*5/9
1550 Tagoa=Tagoc+273.15
1560 Beta=0
1570 FOR I=1 TO 8
1580 Bet=F(I)*(.65-.01*Tagoc)^(I-1)
1590 Beta=Beta+Bet
1600 NEXT I
1610 Pusa=EXP(LOG(217.99)+.01/Tagoa*(374.136-Tagoc)*Beta)*14.696
1620 Pua=14.696*Wago/(Wago+.62198)
1630 Phi=Pua/Pusa
1640 A$="***"
1650 IF Thi>212 THEN PRINT USING 1680;Tagi,Wagi,Xgi,Xgo,Tsgic,A$
1660 IF Thi<212 THEN PRINT USING 1670;Tagi,Wagi,Xgi,Xgo,Tsgic,Thi
1670 IMAGE 7X,DDD.D,8X,.DDDD,8X,DD.D,9X,DD.D,9X,DDD.D,6X,DDD.D
1680 IMAGE 7X,DDD.D,8X,.DDDD,8X,DD.D,9X,DD.D,9X,DDD.D,8X,AAA
1690 END

```

PARTIAL LOAD PERFORMANCE ANALYSIS OF
LIQUID SORBENT COOLING/DEHUMIDIFICATION
SYSTEMS

by

ARSHAD YAQUB

B.E., N.E.D. University of Engineering, Karachi, Pakistan.

1978

AN ABSTRACT OF A MASTER'S THESIS

submitted in partial fulfillment of the
requirements of the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1982

ABSTRACT

Cooling/dehumidification systems are designed for most severe outside conditions but are seldom subjected to design conditions. A liquid sorbent cooling/dehumidification system consisting of an absorber, a regenerator a cooling tower and a solution cooler was selected in this study. A mathematical model describing the performance of individual components used in the system was developed. The partial load performance of the system using lithium chloride solution as the sorbent was analyzed. Variations in the parameters used to define and control the performance of liquid sorbent systems were studied. Sorbent systems furnishing supply air at a specific moisture content irrespective of outside air conditions were modeled and analyzed. Regenerator performance for different operating conditions and solution concentrations was also analyzed, to observe the heating requirements for regeneration of sorbent solution.

The model developed in this study is general in nature and therefore any liquid sorbent system using different sorbent solutions in which cooling and heating of the solution is performed externally can be analyzed, when operating at off design conditions.