

**HEAT AND MASS TRANSFER CHARACTERISTICS:
DESIGN AND OPTIMISATION OF ABSORPTION
REFRIGERATION MACHINES**

A THESIS

by

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Dedication

*I dedicate this work to my godfather
Michalis Mazarakis
upon whose memory I drew courage
strength and patience.*

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CONTENTS

NOMENCLATURE	xvi
SYNOPSIS	xx

CHAPTER 1 INTRODUCTION

1.1	Historical note	2
1.2	Powering methods of the absorption cycle	5
1.3	Pumping methods of the working fluids	9
1.4	Refrigerants for the absorption machines	11
1.5	Component analysis for 2-fluid systems	15
1.5.1	Generator	16
1.5.2	Distillation column	17
1.5.2.1	Packed towers	18
1.5.2.2	Plate towers	19
1.5.3	Liquid-liquid heat exchanger	20
1.5.4	Absorber	21
1.6	Simulation models for performance predictions	23
1.7	The objectives of this study	24

CHAPTER 2 BACKGROUND ON THE HYDRODYNAMICS OF TWO-PHASE FLOW AND HEAT AND MASS TRANSFER CORRELATIONS. CHOICE OF THE TYPE OF ABSORBER

2.0	General	27
2.1	The absorption process	29
2.2	Absorbers and two-phase flow analysis	35
2.2.1	Tubular absorbers	35
2.2.1.1	Horizontal tubular absorbers	36

2.2.1.2	Vertical tubular absorbers	40
2.2.2	Film absorbers	46
2.3	Hydrodynamics of two-phase flow in vertical tubes	49
2.3.1	Film thickness calculations	52
2.3.2	Pressure drop calculations	53
2.4	Interfacial area in co-current gas-liquid flow	57
2.5	Mass transfer in two-phase gas-liquid flow	57
2.6	Operational arguments between film and vertical tubular bubble absorbers	58
2.7	Design calculations of the tubular bubble absorber	59
2.7.1	Heat transfer calculations	65
2.7.2	The overall volumetric mass transfer coefficient $K_L\alpha$	67
2.8	The influence of various parameters on the prediction of height of the absorber tubes	68

CHAPTER 3 ANALYSIS OF THE SIMULATION MODEL PRESENT WORK

3.0	General	72
3.1	Performance of the absorption refrigeration machines	72
3.2	Graphical evaluation of the heat effects of the absorption cycle	78
3.2.1	General	78
3.2.2	Individual component design of the plant	78
3.2.2.1	The evaporation process and estimation of the operating pressures	78
3.2.2.2	Vapour-liquid heat exchanger (pre-cooler)	82
3.2.2.3	Determining the conditions at the condenser and the strength of the solution required for the plant	84
3.2.2.4	Determining the conditions at the generator	85
3.2.2.5	Purification of the refrigerant mixture	89
3.2.2.6	The solution pump	93
3.2.2.7	Optimising the solution heat exchanger	93
3.2.2.8	Throttling and absorption processes	95
3.3	A case study	97
3.3.1	Problem statement	97

3.3.2 A step-by-step analysis	97
3.3.3 Comments on the optimisation procedure	101
3.4 Description of the simulation model	105
3.4.1 Pressure evaluation	106
3.4.1.1 Low pressure	106
3.4.1.2 High pressure	108
3.4.2 Determination of the evaporator coil and pre-cooler conditions	111
3.4.3 Absorber	114
3.4.4 Pump	115
3.4.5 Solution heat exchanger	117
3.4.6 Generator	119
3.4.7 Distillation column	121
3.4.8 Mass flow rates and heat rates	122
3.4.9 Flowchart for the computer program	127

CHAPTER 4 DESIGN OF THE ABSORPTION REFRIGERATION EXPERIMENTAL PLANT

4.0 General	133
4.1 Component design of the experimental plant	134
4.1.1 Condenser-deflegmator combined design	134
4.1.2 Solution heat exchanger	137
4.1.3 Absorber	137
4.1.4 Evaporator and pre-cooler	140
4.1.4.1 Evaporator tubes	140
4.1.4.2 Pre-cooler	141
4.1.5 Generator	144
4.1.6 Distillation column	147

CHAPTER 5 RESULTS AND CONCLUSIONS

5.0	General	153
5.1	Measuring equipment	154
5.1.1	Error analysis	155
5.2	Experimental procedure and conditions	158
5.3	Controls of the experimental plant	158
5.4	Comparison of results	159
5.4.1	Generator's heat and mass flow rates	159
5.4.2	Heat rejected from the absorber	165
5.4.3	Rejected heat from the condenser and deflegmator	172
5.4.4	The work of the solution-processing pump	176
5.4.5	Coefficient of performance	177
5.5	Conclusions	187
5.5.1	The simulation model	187
5.5.2	Comparison between simulated and experimental results	187
5.5.3	Design considerations of an absorption refrigeration plant	188

REFERENCES	190
-------------------	-----

BIBLIOGRAPHY	204
---------------------	-----

APPENDICES	210
-------------------	-----

SUBROUTINES	211
--------------------	-----

APPENDIX A	212
-------------------	-----

A1.	The P-T-x relation	212
A2.	Routine P-T-x (x-P are known)	214
A3.	Routine P-T-x (T-P are known)	214

APPENDIX B	215
B1. Enthalpy of the saturated liquid	215
B2. Enthalpy of the saturated vapour	216
APPENDIX C	218
C1. Vapour mixture concentration	218
APPENDIX D	221
D1. Physical properties of the working fluids	221
D1.1. Ammonia liquid	221
D1.2. Ammonia vapour	221
D1.3. Water	221
D1.4. Solution	221
D1.5. Air properties	222
APPENDIX E	223
E1. Condenser calculations	223
APPENDIX F	227
F1. Solution heat exchanger	227
APPENDIX G	230
G1. Absorber	230
G1.1. Water-side partial heat transfer coefficient a_c	230
G1.2. Solution-side partial heat transfer coefficient a_i	232
G1.3. Heat and mass transfer computations	235
APPENDIX H	236
H1. Evaporator coils	236
H1.1. Tube definition for annular flow	236
H1.2. Heat transfer calculations	238
H1.2.1 The refrigerant-side heat transfer coefficient	238

H1.2.2 The water-side heat transfer coefficient	240
H2. Pre-cooler tubes	241
H2.1. Vapour-side heat transfer coefficient	243
H2.2. Liquid-side correlations heat transfer coefficient	244
APPENDIX I	245
I1. Generator	245
I1.1. The solution-side heat transfer coefficient	245
I1.2. The gas-side heat transfer coefficient	247
APPENDIX J	249
J1. Distillation column The M ^C Cabe-Thiele Method	249
APPENDIX K	251
K1. Correlations for the interfacial area co-current gas-liquid flow	251
APPENDIX L	256
L1. Mass transfer in two-phase gas-liquid flow	256
L2. Correlations by Kasturi and Stepanek, Shilimkan and Stepanek	260
L3. Correlations by Tomida et al	267

LIST OF FIGURES

CHAPTER 1

Figure 1.1	Faraday's experimental apparatus	2
Figure 1.2	Schematic arrangement of the simple two-fluid absorption cycle	3
Figure 1.3	Line diagram of the three-fluid absorption refrigeration unit	4
Figure 1.4	Surface tension effect in the lift tube	9
Figure 1.5	Solution and vapour flow in the two-fluid absorption cycle	15
Figure 1.6	Component arrangement of the improved absorption cycle	16
Figure 1.7	The packed distillation tower	18
Figure 1.8	Schematic of the plate arrangement in the distillation column	19
Figure 1.9	Schematic of the types of the bubble plates	19
Figure 1.10	The multi-tubular solution heat exchanger	20
Figure 1.11	Types of absorbers	22

CHAPTER 2

Figure 2.1	Partial pressure and concentration gradients at a gas-liquid interface	29
Figure 2.2	Composition at an interface between phases	31
Figure 2.3	The P-T-x diagram of the non-equilibrium process	33
Figure 2.4	A typical horizontal multi-tubular absorber	37
Figure 2.5	Two-phase flow in horizontal tubes	38
Figure 2.6a	Baker's flow pattern chart for two-phase flow in horizontal tubes	39
Figure 2.6b	Taitel and Dukler's two-phase flow map for horizontal tubes	39
Figure 2.7	Two-phase flow in vertical tubes	42
Figure 2.8	The flow regime map for air-water two-phase flow at 1 Atm in vertical tubes	43
Figure 2.9	The map by Hewit and Roberts for two-phase vertical flow	44
Figure 2.10	Pressure drop relationship and flow regions by Grovier et al	45
Figure 2.11	The slug-flow model in vertical tubes	51
Figure 2.12	The gas-liquid flow mechanisms according to Martinelli	54
Figure 2.13	The differential element for the vertical tubular absorbers	63

CHAPTER 3

Figure 3.1	Component arrangement of the single stage two-fluid absorption system	73
Figure 3.2	The COP of the reversible absorption cycle	74
Figure 3.3	Comparison of the COPs between a real and a reversible absorption cycle	74
Figure 3.4	Typical COP curves for a variety of conditions	75
Figure 3.5	Optimisation curves for the ammonia-water absorption refrigeration machines	77
Figure 3.6	Vapour/liquid equilibria during evaporation of ammonia/water mixture at constant pressure	78
Figure 3.7	Saturated vapour temperatures at complete evaporation of various ammonia/water mixtures	79
Figure 3.8	Schematic of the evaporator/pre-cooler assembly	79
Figure 3.9	Schematic of the enthalpy-concentration (h-x) diagram for a condition of saturated vapour at the pre-cooler's exit	80
Figure 3.10	Schematic of the (h-x) diagram of the evaporation process	81
Figure 3.11	Determination of the pre-cooler's load on the h-x diagram	83
Figure 3.12	Diagrammatic approach to determine the condenser's conditions and the strength of the strong solution	84
Figure 3.13	Effect of the generator temperature on the heat and mass rates	85
Figure 3.14	Mass and heat rates around the generator and deflegmator	86
Figure 3.15	The Ponchon-Savarit diagram for the distillation process	89
Figure 3.16	Schematic of the vapour and liquid fluxes across the distillation column	90
Figure 3.17	Schematic of the pump's and heat exchanger's mass and heat balance	93
Figure 3.18	Sub-cooled condition of the strong solution in the heat exchanger	94
Figure 3.19	Boiling condition of the strong solution in the heat exchanger	94
Figure 3.20	Schematic of the heat and mass rates around the absorber	96
Figure 3.21	The throttling and absorption processes on a h-x diagram	96
Figure 3.22	A demonstration of the graphical approach used in the design of an absorption refrigeration plant	98
Figure 3.23	A demonstration of the graphical approach used in the design of an absorption refrigeration plant	99
Figure 3.24	A demonstration of the graphical approach used in the design of an absorption refrigeration plant	103

Figure 3.25	The complete graphical presentation of the heat effects of the optimised absorption cycle	104
Figure 3.26	Schematic of the high pressure components for the refrigerant flow	108
Figure 3.27	The evaluation of the high pressure on the h-x diagram	108
Figure 3.28	The evaporator's exit conditions on the h-x diagram	111
Figure 3.28	The flowchart for the computer model	127

CHAPTER 4

Figure 4.1	Schematic arrangement of the experimental plant	133
Figure 4.2	Modification to the condenser to include the heat of the deflegmator	134
Figure 4.3	Detail drawing for the condenser	136
Figure 4.4	Detail drawing for the absorption column	139
Figure 4.5	Detail drawing for the evaporator and pre-cooler	143
Figure 4.6	Detail drawing for the generator	145
Figure 4.7	3-D sectional view of the generator	146
Figure 4.8	The M^C Cabe-Thiele diagram for the number of theoretical plates in the distiller	147
Figure 4.9	Detail drawing of the distillation column	149
Figure 4.10	3-D sectional view of the distillation column	150
Figure 4.11	3-D assembly view of the generator and distillation column	151

CHAPTER 5

Generator's heat input versus weak solution temperature

Figure 5.1	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	160
Figure 5.2	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	160
Figure 5.3	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	161
Figure 5.4	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	161

Strong solution mass rate versus weak solution temperature

Figure 5.5	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	162
Figure 5.6	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	162
Figure 5.7	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	163

Figure 5.8	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	163
Figure 5.9	h-x diagram illustrating the disagreement between the experimental and simulated results	164
Absorber's heat rate versus weak solution temperature		
Figure 5.10	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	166
Figure 5.11	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	166
Figure 5.12	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	167
Figure 5.13	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	167
Absorber's heat balance versus weak solution temperature		
Figure 5.14	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	168
Figure 5.15	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	168
Figure 5.16	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	169
Figure 5.17	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	169
Heat exchanger's heat balance versus weak solution temperature		
Figure 5.18	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	170
Figure 5.19	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	170
Figure 5.20	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	171
Figure 5.21	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	171
Condenser's heat rate versus weak solution temperature		
Figure 5.22	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	172
Figure 5.23	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	173
Figure 5.24	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	173
Figure 5.25	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	174
Distillation column's heat rate versus weak solution temperature		
Figure 5.26	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	174
Figure 5.27	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	175
Figure 5.28	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	175
Figure 5.29	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	176
Coefficient of performance versus weak solution temperature		
Figure 5.30	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	177
Figure 5.31	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	178
Figure 5.32	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	178
Figure 5.33	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	179

APPENDICES

Figure A.1	The P-T-x relationship	213
Figure C.1	The re-constructed enthalpy-concentration diagram	220
Figure K.1	Relationship between the interfacial area and Jepsen's energy factor	252
Figure K.2	The dependence of the interfacial area on the superficial velocities	253
Figure K.3	Relationship between the liquid void fraction with the superficial velocities	253
Figure K.4	The void fraction related interfacial area versus the superficial velocities	253
Figure K.5	Kasturi and Stepanek's correlation of the interfacial area	254
Figure L.1	The Prandtl and Schmidt functions ϕ for heat and mass transfer correlations	256
Figure L.2	Comparison of Pressure drop versus volumetric mass transfer coefficient	259
Figure L.3	Comparison of volumetric mass transfer versus Jepsen's energy factor	259
Figure L.4	The volumetric mass transfer coefficient versus the superficial velocities for 6mm tubes	261
Figure L.5	The volumetric mass transfer coefficient versus the superficial velocities for 10mm tubes	261
Figure L.6	The volumetric mass transfer coefficient versus the superficial velocities for 15mm tubes	262
Figure L.7	The volumetric mass transfer coefficient versus the superficial velocities for 20mm tubes	262
Figure L.8	The mass transfer coefficient versus the superficial velocities	263
Figure L.9	Correlation of the mass transfer coefficient according to Levich theory	263
Figure L.10	The gas-side volumetric mass transfer coefficient versus the superficial velocities for 10mm tubes	265
Figure L.11	The gas-side volumetric mass transfer coefficient versus the superficial velocities for 20mm tubes	265
Figure L.12	The gas-side mass transfer coefficient versus the superficial velocities for 6mm tubes	266
Figure L.13	The volumetric mass transfer coefficient versus the superficial velocities according to Tomida et al	267
Figure L.14	The volumetric mass transfer coefficient versus the Jepsen's energy factor according to Jepsen and Tomida	268
Figure L.15	A comparative study of the correlations for the volumetric mass transfer coefficient according to Tomida, Kulic, Gregory and Banerjee	270

LIST OF TABLES

CHAPTER 2

Table 2.1	Table of parameters in Lockhart-Martinelli's four-type flow mechanism	56
Table 2.3	Comparison of the mean pressure drop in the absorber from various studies	61
Table 2.4	Comparison of the mean void fraction in an absorbing column from various studies	61

CHAPTER 3

Table 3.1	Coefficients for the polynomials of the optimisation curves	76
Table 3.2	Coefficients for the specific volume of liquid ammonia and water	115
Table 3.3	Coefficients for the specific heat C_p of liquid ammonia and water	117
Table 3.4	Simulated results per kg of refrigerant, used in the design of the experimental absorption refrigeration plant	124
Table 3.5	Heat balance and coefficient of performance simulated results per kg of refrigerant, used in the design of the experimental plant	124
Table 3.6	Simulated results, used for sizing the experimental plant	125
Table 3.7	Heat balance and coefficient of performance simulated results used for sizing the experimental plant	125
Table 3.8	Simulated results for the optimum experimental absorption refrigeration plant	126
Table 3.9	Heat balance and coefficient of performance simulated results for the optimum the experimental plant	126

CHAPTER 4

Table 4.1	Design specifications for the condenser	135
Table 4.2	Design specifications for the heat exchanger	137
Table 4.3	Design specifications for the absorber	138
Table 4.4	Design specifications for the evaporator	141
Table 4.5	Design specifications for the pre-cooler	142
Table 4.6	Design specifications for the generator	145
Table 4.7	Design specifications for the distillation column	148

CHAPTER 5

Experimental results

Table 5.1	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	180
Table 5.1	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	181
Table 5.1	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	182
Table 5.1	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	183

Simulated results

Table 5.2	$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$	184
Table 5.2	$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$	184
Table 5.2	$T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$	185
Table 5.2	$T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$	186

APPENDICES

Table T1	Coefficients of the polynomials for the Pressure-Temperature-concentration (P-T-x) relationship	212
Table T2	Coefficients of the polynomial for the Fourier equation in the enthalpy of the saturated liquid relationship	216
Table T3	Coefficients of the polynomial for the Fourier equation in the enthalpy of the saturated vapour relationship for $0.0 < y < 0.9$ concentrations	217
Table T4	Coefficients of the polynomial for the Fourier equation in the enthalpy of the saturated vapour relationship for $0.9 < y < 1.0$ concentrations	217

Table T5	Coefficients of the polynomials for the double Fourier equation in the vapour concentration relationship	219
Table G1	Comparison of the heat and mass transfer coefficients in the calculation of the height of the absorbing column	235
Table G2	The heat and mass transfer coefficients and height of the tubular absorber for complete absorption of ammonia vapour	235
Table K1	Values of the constant and exponent in the relation by Kasturi and Stepanek for various tube diameters	255

NOMENCLATURE

SYMBOL	DESCRIPTION	UNITS
A	area	m^2
A_s	surface area of rising slug	m^2
C	constant in Blasius equation for friction factor	
c	concentration of solute gas in liquid	$kmol/m^3$
c^*	equilibrium concentration of solute gas in liquid	$kmol/m^3$
C_{nb}	constant in Nusselt number expression for nucleate boiling	
COP	coefficient of performance	
C_p	specific heat at constant pressure	J/kgK
D	diameter	m
ΔT_{Ln}	logarithmic mean temperature difference	
dx	% concentration difference = $(x^* - x)$	
E_B	Banerjee's energy dissipation parameter	J/m^3s
E_J	Jepsen's energy dissipation parameter	J/m^3s
Eu	Euler number	
F	co-ordinate system in Taitel's flow chart	
f	mass flow rate of strong solution per kg of refrigerant	kg/s
f	friction factor used in Martinelli's pressure drop	
Fr	Froude number	
F_{TH}	thermal resistance for annular flow	
g	acceleration due to gravity	m/s^2
G	mass velocity	kg/m^2s
H	Henry's law constant	kPa
h	specific enthalpy	J/kg
h_{fg}	latent heat	J/kg
h_{fg}^*	latent heat correction as per Sadasivan et al [M6]	J/kg
h_s	distance between acceding bubbles	m
j	superficial velocity	m/s
Ja	Jacob's number	
K	co-ordinate system in Taitel's flow chart	
k	thermal conductivity of material	W/mK
k_L	individual mass transfer coefficient	kg/m^2s
K_L	overall mass transfer coefficient	kg/m^2s
$K_L\alpha$	overall volumetric mass transfer coefficient	kg/m^3s
$k_L\alpha$	volumetric mass transfer coefficient	kg/m^3s
L	length	m
L_c	characteristic length for nucleate boiling	
m	Henry's law coefficient	
M	mass flux	kg/m^2s
N	mass flux transfer	kg/m^2s
N	number of tubes	
Nu	Nusselt number	
P	pressure	bar

Pr	Prandtl number	
Q	heat rate	W
Q	volume flow rate	m ³ /s
r	tube radius	m
R _{FF}	fouling resistance	m ² K/W
R	reflux ratio	
Re	Reynolds number	
REF	refrigeration effect	kJ/kg
Sc	Schmidt number	
Sh	Sherwood number	
T	co-ordinate system in Taitel's flow chart	
T	absolute temperature	K
t	temperature	°C
t	tube thickness	m
u _g	vapour-slag velocity	m/s
u	velocity	m/s
U	overall heat transfer coefficient	W/m ² K
u _{GS}	superficial gas velocity	m/s
u _{L,S}	superficial liquid velocity	m/s
u _o	velocity component due to buoyancy	m/s
v	specific volume	m ³ /kg
V _f	film velocity	m/s
W	mass rate in Martinelli's expression	kg/s
W	work	W
x [*]	concentration in liquid-phase in equilibrium with its vapour	kg/kg
x	liquid-phase concentration	kg/kg
X	co-ordinate system in Taitel's flow chart	
X	2-phase flow parameter used in Martinelli's expression	
y	vapour-phase concentration	kg/kg
z	length of absorption, height of absorption column	m

GREEK SYMBOL	DESCRIPTION	UNITS
α	relative volatility	
α	partial heat transfer coefficient	W/m ² K
α	interfacial area per unit volume	m ² /m ³
α	area ratio for liquid in Blassius equation for friction factor	
β	area ratio for gas in Blassius equation for friction factor	
Δ	difference	
δ	film thickness	m
$\mathcal{E}$	effectiveness	
ε	void fraction	
η	efficiency	
λ	correction factor in Baker's flow chart	
λ_F	flooding parameter in distillation theory	
μ	dynamic viscosity	kg/ms
ν	kinematic viscosity	m ² /s
ξ	friction factor in Gnielinski's relation for H.T.C.	
Π	pole in distillation theory	
ρ	density	kg/m ³
σ	surface tension	N/m
ϕ	vapour quality	
Φ	boiling mode control parameter	
Φ	intercept concentration in vapour in the McCabe-Thiele x-y graph	
Φ	2-phase flow parameter used in Martinelli's expression	
ψ	correction factor in Baker's flow chart	

INDICES	DESCRIPTION
a	ammonia
ABS	absorber
b	bulk measurement as in T_b, x_b
c	coolant, condensate
C	condenser
D	deflegmator
E	evaporator
eq	equilibrium condition
EX	heat exchanger
f	fluid, solution, film
F	flooding index in distillation theory
g	gas
G	gas-phase, vapour, generator
GEN	generator

H	hydraulic - as in D_{II}
i	inlet, interface, internal, ideal
L	liquid-phase
L_f	liquid film
M	Martinelli's identifier as in X_M
M	identifier for the pressure drop due to change of momentum
ml	molal liquid
mv	molal vapour
O	outlet, outside
P	pump
PR	pre-cooler
REF	refrigerant
s	saturated solution, sink
sat	saturation condition
SP	single-phase
ST	strong solution
t	tube
TP	two-phase
v	saturated vapour
w	water
WE	weak solution
x_a, x_b	liquid-phase concentration at levels a ,b
x_i, y_i	liquid, vapour concentrations at level i
y_a, y_b	gas-phase concentration at levels a ,b

SYNOPSIS

An extensive literature study on the subject of absorption refrigeration has revealed that there is no proven methodology that can be used to design, optimise and size a plant. On the contrary there are numerous methods which analyse the performance of an existing plant from collected data. These methods however, do not provide any means of predicting how the analysed plant would perform if one or more of its working conditions should change.

This work provides a complete design for an ammonia-water absorption refrigeration plant. The ranges of the working conditions in the evaporator and condenser are from -5°C to -55°C and from 5°C to 45°C respectively. For any combination of temperatures within these ranges, the temperature of the generator is optimised for maximum performance of the plant. Depending on the refrigeration capacity, the components are sized and designed. The proper design of the various components takes into account both heat and mass transfer correlations, something which has not been considered in the past as a necessary step for absorption refrigeration machines. Literature indicates that there is a vast amount of research into the absorption of gases into liquids and two-phase flows through tubes of various sections. The correlations cited in these studies have been used in designing the absorption column, evaporator and generator.

The proposed optimisation method is a novel approach in designing a plant and stems from the fact that the performance of the absorption refrigerator reaches a maximum at a specific generator temperature. For this, optimisation curves have been developed, which for a particular combination of evaporator and cooling environment temperatures, both the optimum generator temperature and the maximum performance of the plant are predicted.

The equations used in the computerised simulation procedure are based on the well-established enthalpy-concentration chart for the ammonia-water mixtures. Thus the properties of the mixture at various points in the plant are accurately predicted. Published computerised procedures in the past have been proven inaccurate in predicting the properties of the mixtures at near pure-ammonia concentrations.

The validity of the simulation model is verified by tests performed on a laboratory size absorption plant. The plant was built from design parameters predicted by the simulation model for a refrigeration capacity of 1kW at -15°C evaporator coil and 25°C condensate temperatures. Thereafter the unit was operated for a range of evaporator conditions while the generator temperature was varied.

CHAPTER 1

1.0 INTRODUCTION

1.1 Historical Note

Gas absorption was known as early as 1770. Absorption refrigeration originated in 1824 by Faraday, who proved that certain gasses which were believed to exist only in vapour form could be liquefied. One of these gasses was ammonia which could be absorbed in vast quantities by silver chloride crystals. His apparatus consisted of a closed inverted “V” tube which contained in one end the ammonia silver chloride crystals, Figure 1.1. That end of the tube was heated and the opposite end cooled with water. Soon the ammonia gas was released from the compound and liquid ammonia appeared in the cool end of the tube. When he removed the heat and the water container, he observed that the liquid began to boil violently changing again into vapour and at the same time the ammonia vapour was reabsorbed by the crystals. The region of the apparatus containing the liquefied ammonia was intensely cold as the ammonia drew heat from its environment in order to evaporate [A5]. Faraday’s experimental apparatus was the first absorption machine of the intermittent type.

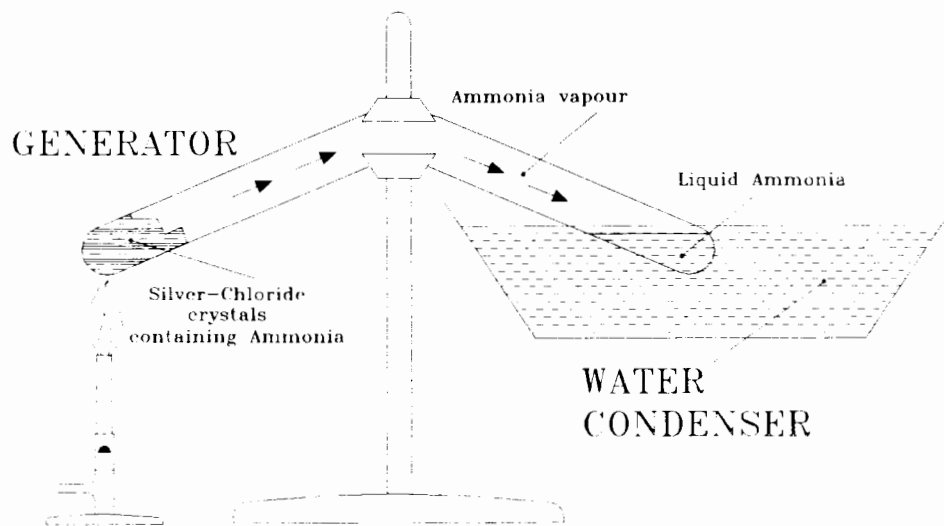


Figure 1.1 *Faraday's experimental apparatus*

The first continuously working absorption refrigeration system using sulphuric acid and water as absorbent and refrigerant respectively was invented by Carré in 1859, who in the same year, operated the first ammonia-water absorption refrigerator.

His system consisted of components capable of assisting the refrigerant through a continuous cycle, as shown in Figure 1.2.

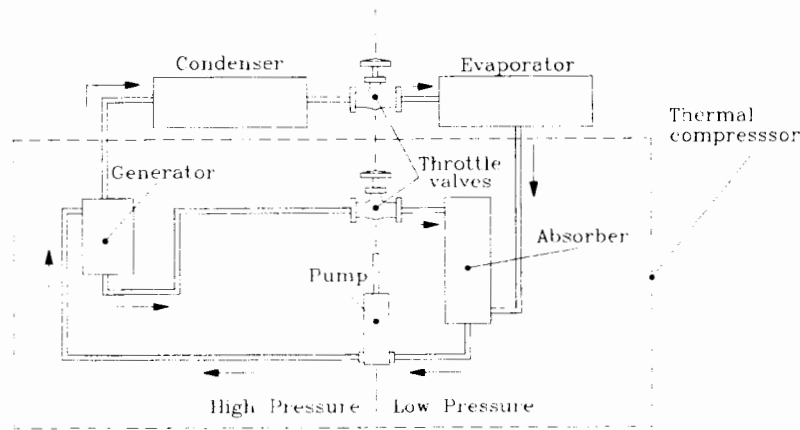


Figure 1.2 *Schematic arrangement of the simple two-fluid absorption cycle*

The refrigerant enters the evaporator where it evaporates at low pressure and temperature by drawing heat from the environment. The vapour leaving the evaporator, is absorbed in a liquid which has high affinity for the refrigerant and transferred via a liquid pump into the boiler or generator. There, heat is applied which reduces the affinity of the liquid, and drives off the refrigerant vapour at high pressure, which then passes to the condenser to be liquefied.

The liquid refrigerant enters the evaporator again and the cycle is repeated. The remaining liquid in the generator which is weak in refrigerant, passes through a throttling valve in the absorber to receive new vapour refrigerant. After it has been enriched, it flows through its own cycle. Since the generator, solution throttle valve, absorber and pump are involved in compressing the refrigerant vapour by thermal action, they are called the “thermal compressor”.

Early in the 20th century the idea and the application of the absorption system was shelved, mainly due to its high power consumption, giving ground to the development of the fully enclosed ammonia compression refrigeration. In other words, the thermal compressor was substituted by the mechanical compressor, which was placed there to withdraw the refrigerant vapour from the evaporator and then to compress it. In all other respects, the refrigeration cycle is identical in both systems.

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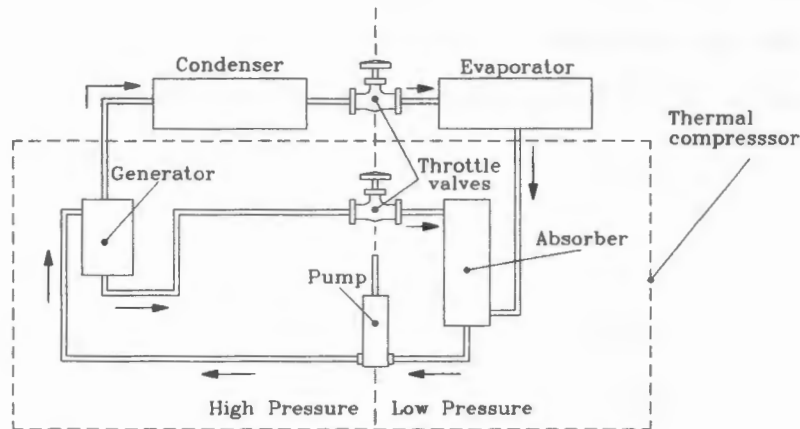


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Absorption refrigeration was revived again around the 1930's with the invention of the so-called "candle fridge", Figure 1.3, by the two Swedish researchers, B. Von Platen and C. Munsters. It is a domestic-size, three-fluid system (refrigerant, absorbent and a pressure equalising inert gas), the cycle of which is based on the theory of partial pressures. To boil the refrigerant off the solution, the system used initially the heat from a paraffin or a gas burner [B7]. It was further developed to utilise gas, as well as either a 12V, 120V or 240V supply, but still only for domestic purposes.

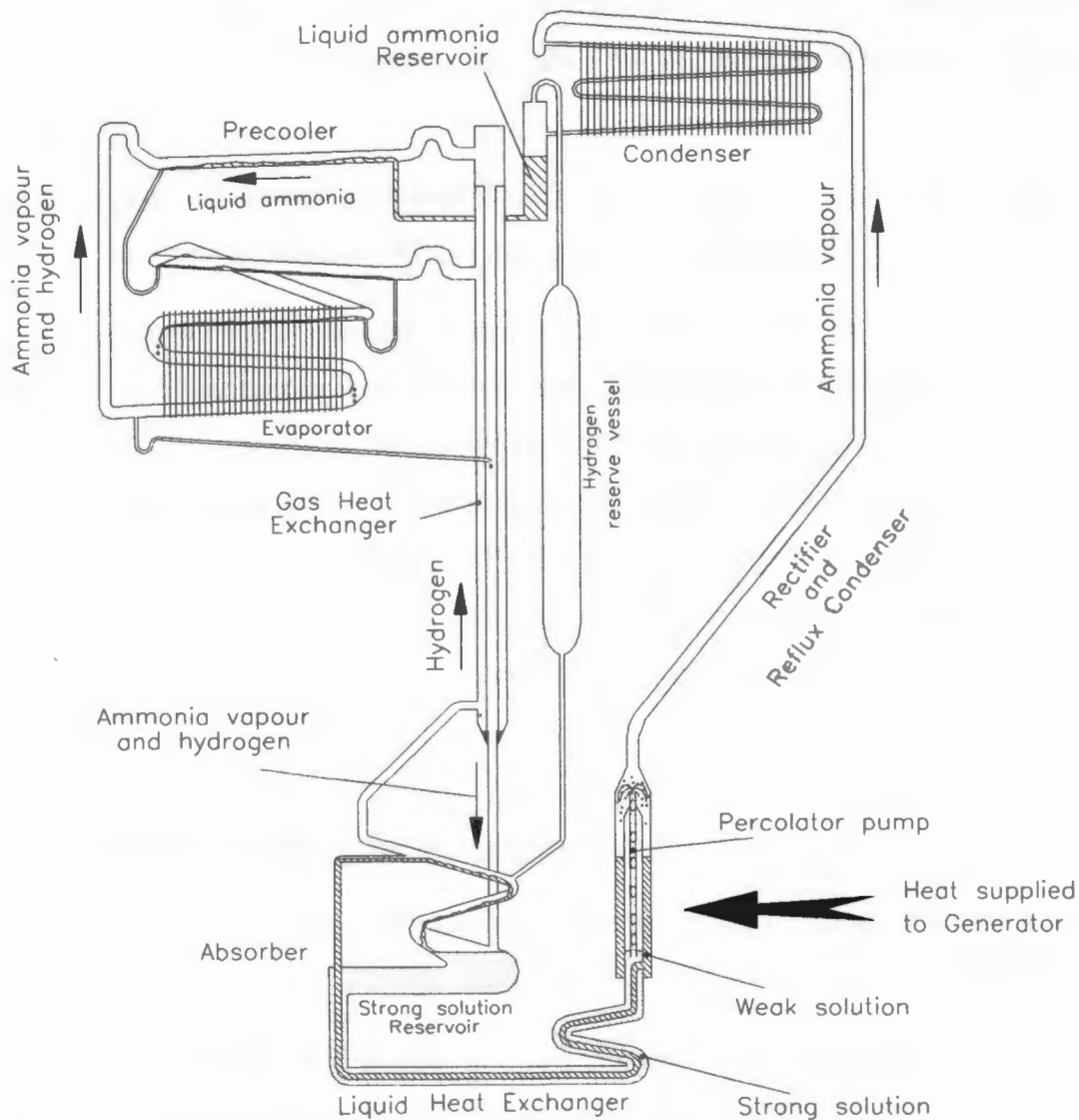


Figure 1.3 *Line diagram of the three-fluid absorption refrigerator unit*

The main reason for its limited size, is due to the solution pumping arrangement which relies upon a percolator-type pump. This pump brings the generator's solution into a higher level and thereafter the fluid circulation depends upon potential levels. Since medium and large scale industrial refrigerators require large and definite amounts of fluid flow, the percolator-type pump cannot meet the requirements. A further drawback to the potential level flow is its inability to survive movement and inclination, as this would disturb the fluid levels.

Despite this ingenious development by the two Swedish researchers, the absorption unit has acquired a bad reputation due to the irresponsible craftsmanship of various manufacturing firms and due to lack of understanding of the absorption behaviour. The development was mainly through a trial and error construction with a minimal backing of theoretical knowledge [S11].

Studies in the last two decades improved the absorption refrigerator and showed that the unit is capable of meeting the demands of the modern refrigeration industry.

Indeed, refrigeration units of cooling capacity in the order of 2 to 5MW exist in some parts of the world, which by far exceed the maximum capacity obtained by a single vapour compression refrigerator [J4]. For example, there are two absorption refrigeration installations each capable of 22.1MW cooling capacity at -5°C evaporator temperatures. These were built in 1981 for the former USSR, by BORSIG GmbH, Berlin [H5]. They are both powered by superheated steam and are used to cool by-products from the process of synthetic unvulcanized rubber (caoutchouc). Unfortunately, there are no performance reports published.

1.2 Powering methods of the absorption cycle

The absorption cycle could use a low grade heat source, in the temperature range of 70°C to 120°C .

Sources of low grade heat are numerous, as for example, solar energy, low-pressure steam, flue and exhaust gases, geothermal, etc. These sources of energy are available, at least some of them at minimal cost and their utilisation has obvious advantages.

A number of countries have launched research programmes to convert solar energy into heat, to power absorption type refrigerators, air-conditioners and heat pumps directly.

Renato Lazzarin [L1] has published results of a two year experiment in North Italy, describing how solar energy was used, either directly or stored, in order to create a working temperature environment for an ammonia-water absorption heat pump. The plant was provided with a seasonal storage, so that useful solar heat could be collected by solar collectors all year round and used when the solar energy was poor or non-available. The earth storage was connected with an energy roof installation and alternated in providing the cold source between the winter and summer months. The performance of the plant was anything but satisfactory. The main source of inefficiency was the poor heat exchange between the ground and the earth storage components and even greater, the low performance of the absorption heat pump within the experimental working temperatures.

Exell et al. [E3] published their work on mathematical modelling of an intermittent ammonia-water absorption refrigerator powered through flat solar collectors. Their results show that for the climate of Thailand, the average daily yield of ice should be 3.2 kg per m² of collector.

Agarwal et al. [A3] reported on results from a laboratory model of a solar powered intermittent refrigeration system using refrigerant R22 and DMF (Dimethyl Formamide), capable of cooling 60kg of water daily from 30°C to 15°C. In the month of July the average refrigeration temperature was 5°C for about 1.5 hours, while in January the average temperature was maintained at -3°C for a period of 3 hours.

Their proposed water-cooler system, uses a water cooled condenser during daytime vapour generation and a water cooled absorber during night time. From the performance specifications it could be deduced that their intermittent refrigeration unit requires enormous amounts of cooling water to produce the required effect. On the other hand, the authors claim that the importance of an intermittent system lies in its simplicity of its operation on, account of the combined use of the solar collector as generator and absorber. However, in their design the two components are independent and separate as to overcome the problem of cooling. This results in a sequence of opening and closing of control valves at sunset and in the early hours of the morning. If this sequence is automated their unit could require the usage of high forms of energy.

Professor I. Borde of Israel, advisor of GALIL Advanced Technologies [G1] gives evidence about the development of an absorption refrigerator utilising waste heat and solar energy, to provide refrigeration in the range of 0°C to -3°C. According to the report, a “novel”

combination of working fluids was used, with the intention of moving away from the conventional ammonia-water system. Without the constituents of the refrigerant-absorbent pair being revealed, it is stated that the refrigerant belongs to the "FREON" group and the absorbent is an organic liquid solvent. Subsequent papers by the same author report on hydrogenated chlorofluorocarbons and Dimethyl Formamide (DMF) or Tetraethylene Glycol Dimethyl Ether (TEG.DME) combinations.

The project was for a 250 ton capacity refrigeration plant, to be used in agricultural centres. The company produced an analysis which compares the proposed absorption refrigerator to the vapour compression refrigerator, showing the former to be about 60% more economical in the machinery set-up and service, and about 50% more economical in power consumption.

Alvares and Trepp [A6] from the University of Brazil report on a mathematical model of a solar driven aqua-ammonia absorption refrigeration system. The modelling was done in such a way that the coefficient of performance (COP) could be calculated with the components of the system being able to be "turned on" or "turned off" or varied in characteristics. In this way thirteen absorption cycles were studied in the search for the optimum coefficient of performance. The cycle which gave the highest performance was that which had the highest refrigerant concentration, and included a solution heat exchanger and a vapour-liquid heat exchanger (pre-cooler). The COP achieved by this cycle was 0.602.

Kaushik and Bhardwaj [K3] from the Indian Institute of Technology, investigated an ammonia water absorption cycle for solar refrigeration, heat pump, and air conditioning operations. With the aid of a numerical analysis, it was found that at high generator temperatures the COP was reduced. It was also shown that an increase of condenser's temperature improves the performance at higher generator temperatures. In a subsequent paper by Kaushik, Kumar and Chandra [K4] a study is presented of a cascade system of two absorption refrigerators powered by separate solar collectors. Their system could achieve lower evaporator temperatures than the single stage absorption refrigerator, but the overall COP is lower than that of the single stage.

El-Shaarawi and Ramadan [E2] from Alazhar University in Egypt, manufactured and tested an intermittent solar absorption refrigerator. Their paper describes the construction of the refrigeration unit as well as the solar collectors. The experimental data compare favourably with the proposed theoretical model.

Al-Marafie, Suri and Maheshwari [A4] from the University of Kuwait, analysed in a techno-economic study three cycles: an absorption system powered by flat plate solar collector, an absorption system powered by a solar pond and a vapour compression refrigerator powered either directly through the network or through solar photovoltaic collectors. Their results show that while the flat plate solar collector was not economically viable, the solar pond showed a promising and attractive alternative. On the other hand solar photovoltaic cells could become competitive in the future only if their unit cost drops drastically.

The above mentioned publications indicate that it is a world wide tendency to utilise the most abundant source of energy for refrigeration and air-conditioning purposes and it is for this reason that the research on absorption refrigeration, inclines towards utilising solar energy.

However, the efforts are focused on refining the solar panels and the solar generator. Studies on the steam-powered absorption cycle and on the refinement of the thermodynamic analysis of the absorption refrigeration cycle itself, have been neglected. The steam-powered absorption refrigerator is being regarded as a “standard absorption system” and is only re-examined if other machinery and cycles are incorporated. This approach of investigation was adopted by Agarwal et al. [A2] from the Indian Institute of Technology, who analysed with the aid of computers a combined refrigeration cycle. The cycle consisted of a vapour compression refrigerator, the compressor of which was powered by a steam turbine, and an absorption refrigerator powered by the exhaust steam of the turbine. Amongst other performance observations, they praise that the proposed system as a whole, was superior to the absorption system alone. This conclusion however, is erroneous, as the performance of the vapour compression refrigerator and that of the absorption refrigerator cannot be compared, let alone when the absorption cycle is compared with a whole combined cycle.

Beasley and Hester [B4] from University of Clemson, USA proposed a study for a pressure driven absorption cycle. The cycle utilises a semi-permeable membrane to separate the refrigerant from the absorbent. The flow of refrigerant across the membrane is achieved by a pressure differential in excess of the osmotic pressure. Since the refrigerant is separated in a liquid form and also in the absence of heat, there is no need for a condenser. The proposed system is only a theoretical model, as the performance of the membrane is unpredictable. But on the other hand, the predicted coefficients of performance reach values matching the vapour compression cycle.

1.3 Pumping methods of the working fluids

The method for the circulation of the working fluids depends on whether the unit operates on the two or three-fluid principle.

The three-fluid system utilises the partial pressure of an inert low molecular weight gas (for example hydrogen), to equalise the total pressures between the evaporator, absorber, generator and condenser. The circulation thereafter is achieved by gravity effects, as shown in Figure 1.3. In this three-fluid system, there is an optimum circulation rate of solution through the boiler and absorber and the “lift-tube” of the percolator pump must be designed to provide this optimum rate. In the smaller size three-fluid-unit the lift-tube operates under the influence of surface tension at the bottom end of the tube, which dips into and breaks free from the solution surface, the level of which is essential for the lift-tube performance [S10]. In the tube itself, vapour bubbles and liquid slugs are formed in a regular manner.

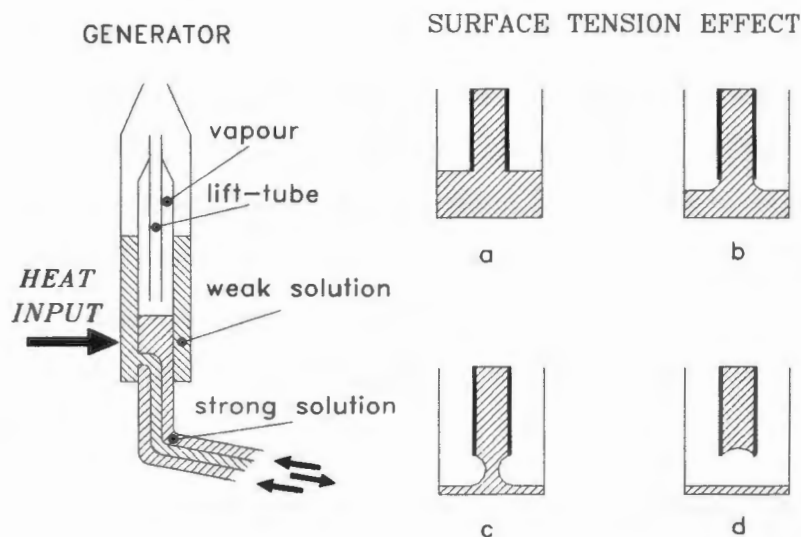


Figure 1.4 *Surface tension effect in the lift-tube arrangement*

Figure 1.4 shows in sequence the formation of the liquid slug in the lift-tube [S10]. The surrounding vapour pushes at the liquid surface causing the solution from the annular space to rise in the tube. At the same time the level drops away from the tip of the tube, but due to surface tension the liquid is still attached. As the lift-tube is the only escape route for the vapour, separation occurs and the liquid slug is pushed up by the rising vapour. New liquid

from the absorber flows into the annulus and as soon the level touches the bottom tip of the lift-tube the process repeats itself.

In larger three-fluid refrigerators, boiling occurs inside the lift-tube and the liquid is pushed upward by the rising vapour.

In the two-fluid system (for example in moderate to large size units), there is a definite pressure difference between the absorber and the generator. This pressure difference has to be bridged by a pump. Pumps of different operating principles have been used for this purpose, such as piston, centrifugal, diaphragm, all of which require external "high grade" energy. It is usually an electric motor which provides the motive power.

The type of pump used depends upon the refrigeration requirements and the temperature range of evaporator temperatures. For example, at low evaporator temperature, equivalent to sub-atmospheric pressures, a positive displacement pump is recommended. At higher evaporator temperatures and large circulation rates, a centrifugal pump could be used.

The invention of the three-fluid system, has motivated the search to find ways to pump the solution of a two-fluid system, by a pump which requires a low grade heat. In this way, the need for higher grade energy input can be reduced. This is possible by using pumping methods which need power of the same quality as the generator. One of these methods is the "thermal pump", the principle of which is based on the invention of Szucs [S14]. An immediate advantage in incorporating a thermal pump is that the plant would be powered exclusively by low grade heat and because there is no mechanical movement, this would lead to minimum maintenance.

Such a pump was built and tested in an absorption unit by the author [V2]. Besides its high cost of construction it is reported that its intermittent behaviour caused instability to the absorption machine during performance testing. Although this "thermal pump" requires heat energy as a prime mover, tests showed that it needs electricity to supply its controls and valves.

1.4 Refrigerants for absorption machines

There are many refrigerants available, but when considered for absorption machines their number is diminished.

The criteria which assist in the choice of refrigerant-absorbent combinations depend on the thermophysical properties of the working fluids. These are:

- a. The heat of vaporisation of the refrigerant. This is considered as one of the most highly valued properties of the refrigerant. It is responsible for the refrigeration effect and the mass flow rate of the refrigerant to produce the required capacity.
- b. The heat of solution. This is the heat which is released when the refrigerant is absorbed in the absorbent. The same quantity of heat must be applied to the solution (under the same conditions) for the refrigerant to be released from the absorbent. These processes are called absorption and desorption respectively. The heat of solution depends upon the affinity that the two substances have for each other.
- c. The vapour pressure of the refrigerant and the absorbent. This property determines the high and low pressures of the system. It is preferable that the difference between the vapour pressures of the refrigerant and the absorbent be as large as possible. If the contrary occurs it is necessary to introduce a rectification column.
- d. The volubility of the refrigerant in the absorbent. It is important for the absorbent to absorb large amounts of refrigerant in order to keep the solution mass flow rates small. However, the greater the volubility of the refrigerant in the absorbent the larger the heat of solution and therefore, large heat exchangers will be required to remove the heat of solution.
- e. The heat capacity of the solution. This property has an impact on the heat transfer between the working fluids. Its value is the combined value of the two constituents according to their concentration. A large value means a small mass flow rate of the working fluids.
- f. The viscosity of the solution. It determines the size of the tubes for a fully turbulent flow.

- g. The thermal conductivity of the solution. This property is involved with the evaluation of the heat transfer coefficient, needed for the heat exchanger's calculations.
- h. The toxicity of the working fluids. The toxic behaviour of the refrigerant or the absorbent concerns safety and comfort. There is more concern now on whether the chemical compounds are in the long run environmental benign, rather than on the direct human safety [K7].
- i. The chemical stability of the working fluids. Decomposition of either constituent would require frequent evacuation of the system. It is therefore essential, that the refrigerant, the absorbent and their mixtures, are stable within the working range of the absorption machine. For example, ammonia in the ammonia-water pair is unstable at temperatures of 170°C and above, producing inert gases which cannot be absorbed or removed from the system [B9].
- j. The corrosive properties of the working fluids. This determines the construction material of the unit.

Researchers have reported a variety of chemical substances with the ability to absorb refrigerant compounds.

Renz and Steikle [R4] used the following working fluids for their proposed continuous absorption system:

- a. ammonia-thiocyanate (NaSCN) aqua solution
- b. R 22-Dimethyl ether - Tetraethylene glycol (DTG)
- c. ternary system methanol-lithium bromide-zinc bromide

Their report compared mass, heat flow rates and component efficiencies, for all three working fluids. It was concluded that, although the ammonia-salt solution appeared to have better thermodynamic properties over the other two combinations and required no rectification, the formation of salt crystals was a drawback. This conclusion was confirmed by Blytas and Daniels [B6] who also reported on the thermodynamic properties of ammonia-NaSCN mixtures.

Tyagi et al. [T6] has reported on the coefficient of performance of an absorption unit with a selection of 26 different refrigerant-absorbent combinations. Emphasis was given to ammonia as refrigerant combined with both water and organic solvents and with various salts to improve the overall performance. Sulphur dioxide and refrigerants R21 and R22 were also included in the list, combined with organic compounds, but with mass flow rates of mixtures almost double to those reported with the ammonia combinations.

Eiseman [E1] compared six halogenated absorption refrigerants in terms of their volubility, chemical stability, construction material decomposition and thermodynamic properties. He concluded that refrigerant R22 is to be preferred above all others, although it features an undesirable relative high pressure. The absorbent was Dimethyl ether-Tetraethylene glycol (an organic solvent first used by Zellhoefer in combination with refrigerant R21) [E1, Z1, Z2].

According to Eisman [E1] FREON 22 (former name of HCFC 22) offers a much safer behaviour towards the environment. Some refrigerator and air-conditioning manufacturers consider it as an alternative to CFC 12.

Ammonia has been used for many decades as refrigerant for both absorption and compression systems. Therefore it could be treated as the centre of comparison for other refrigerants. There are of course certain drawbacks in using the ammonia-water combination in absorption systems:

1. ammonia is a toxic gas
2. it explodes when combined with the right amount of air
3. the condensing pressure is high
4. it becomes unstable at temperatures in the region of 170°C, producing inert gases which have to be purged from the system. This becomes more apparent in the presence of water
5. ammonia becomes very corrosive in the presence of water, to non-ferrous metals, in particular to copper and its alloys. Therefore the construction of the units must be entirely of steel.
6. ammonia-water absorption plants require rectification. With good rectification, vapour concentrations approaching unity is feasible. This can happen at the expense of the distillate, part of which returns to the rectifier as reflux.

In past years the ammonia-water combination was used almost exclusively in absorption systems. It is only in the last two decades that other combinations have been considered and although higher coefficients of performance were achieved, there are still merits in the ammonia-water combination which have not been overshadowed. Some of them are:

1. Water can absorb vast quantities of ammonia, therefore solution circulation rates and the components involved are small.
2. A detailed data-bank exists for the thermodynamic properties of ammonia, water and their vapour and liquid mixtures.
3. Both ammonia and water can be obtained at low cost
4. If salts are added to the $\text{NH}_3\text{-H}_2\text{O}$ mixture, such as LiNO_3 , LiSCN and NaSCN , there is no need for vapour rectification. However, one must ensure against the danger of crystallisation.

An absorption machine designed for the ammonia-water combination still offers a large ground for study into more specific behaviour of its performance and in particular into the performance of the components with respect to the mass and heat flow characteristics of the working fluid.

1.5 Component analysis for 2-fluid systems

The absorption refrigeration system is composed of two cycles: namely, the refrigerant cycle and the solution cycle.

In the refrigerant cycle, the refrigerant goes through the condenser, the expansion valve and the evaporator. In the solution cycle the refrigerant-absorbent solution goes through the so called "thermal compressor" which consists of the absorber, the solution pump, the expansion valve and the generator. All these comprise the simple absorption refrigeration cycle and they are the minimum number of components the absorption refrigerator can have, Figure 1.5.

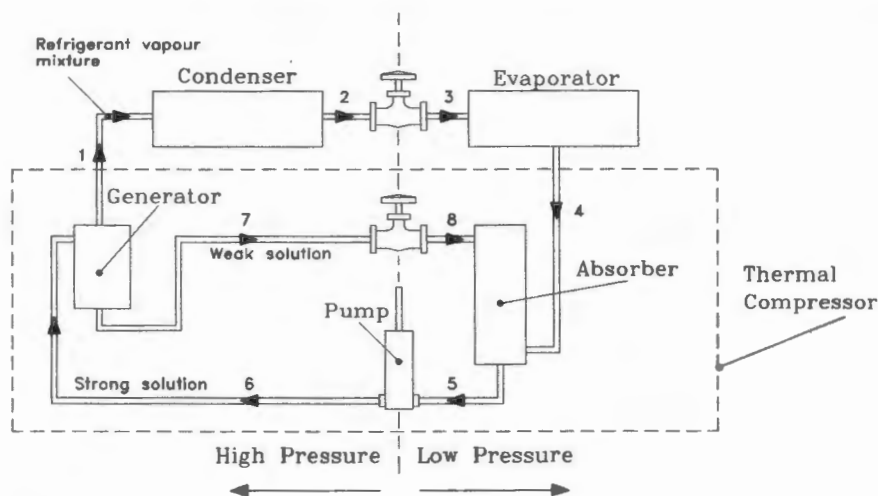


Figure 1.5 *Solution and vapour flow in the two-fluid absorption cycle*

It has become an established procedure and practice to incorporate heat exchangers, in both the refrigerant and the solution lines, in order to improve the cycle performance, Figure 1.6.

In the refrigerant line the vapour-liquid heat exchanger is placed between the condenser and the expansion valve. This reduces the temperature of the liquid refrigerant prior to its expansion, thus increasing the refrigeration effect.

The absorber receives hot-weak solution from the generator which has to be cooled first, before absorption can take place [M1]. On the other hand the generator receives cold-strong solution from the absorber which has to be heated, before liberation of the refrigerant can take place. Therefore a major portion of generator energy requirement can be saved by a liquid-liquid heat exchanger, placed between the absorber and generator.

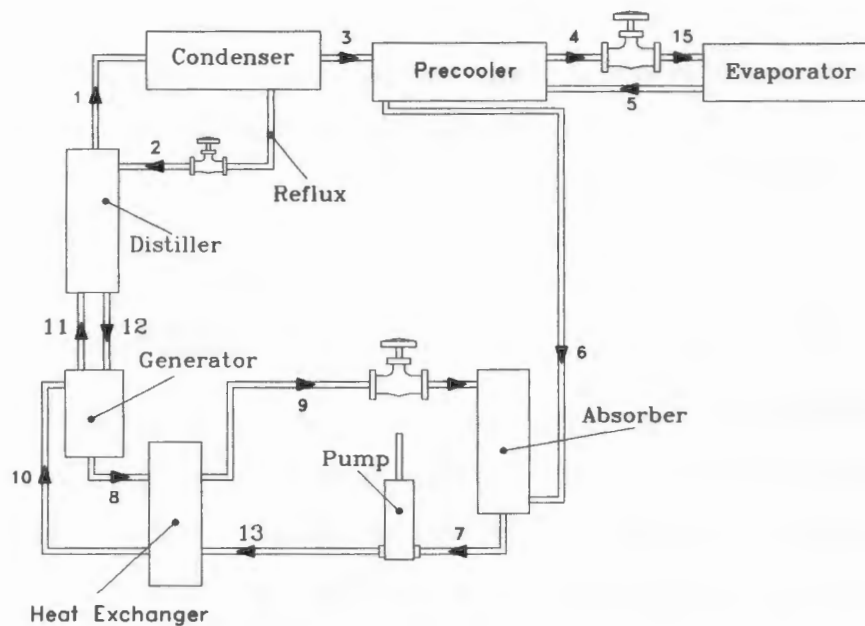


Figure 1.6 *Component arrangement of the improved absorption cycle*

In the case where the absorbent is volatile, as for example in the water-ammonia mixture, a distillation column must be introduced between the generator and the condenser. This will improve the purity of the refrigerant vapour, of course at the expense of the condensed distillate which is returned to the distiller as reflux. The distillation column, complicates the thermodynamic analysis of the cycle and introduces a large increase in the cost of the absorption machine.

1.5.1 Generator

The generator is the device through which heat is supplied to the solution in order to release the refrigerant from its absorbing medium. Its action is more complicated than just heating a liquid from one temperature to another. The generator's process can be divided into a series of step processes [W3, M1]:

1. breaks up the bond of association between the refrigerant and the absorbent.
2. changes the temperature of the resulting liquid refrigerant to its saturation temperature.
3. vaporises the liquid refrigerant.

- 4 changes the temperature of the incoming strong solution to the generator's temperature.
5. if the absorbent is volatile, it evaporates a certain amount of absorbent and raises its temperature to the temperature of the generator.

1.5.2 Distillation column

Distillation is the process of separation, based on the difference in composition between a liquid and the vapour formed from it [R6]. In the case of the absorption refrigerator, the distiller's function is to enrich the refrigerant vapour by removing the vaporised absorbent. It is placed between the generator and condenser and it can be either an integral part of the generator, or a separate component.

The generator-distiller integral component is used mainly for small capacity two-fluid absorption machines and consists of a number of half-plates cascaded on top of the generator. The rising vapour comes into contact with the falling strong solution and thus it is "stripped" of the majority of the absorbent vapour. Ammonia concentrations in the region of 0.990 are feasible.

The three-fluid absorption machines have only a tube connecting the generator to the condenser, and the vapour is purified by condensation of the less volatile absorbent. Some of the refrigerant is also condensed and flows down the tube as reflux and therefore this tube is often called "the reflux condenser".

Reflux is of great importance in distillation practice, regardless whether the distiller is small or large, a simple tube or an integral part of the generator. If reflux is absent, the distiller becomes just a transfer line without the ability of distillation.

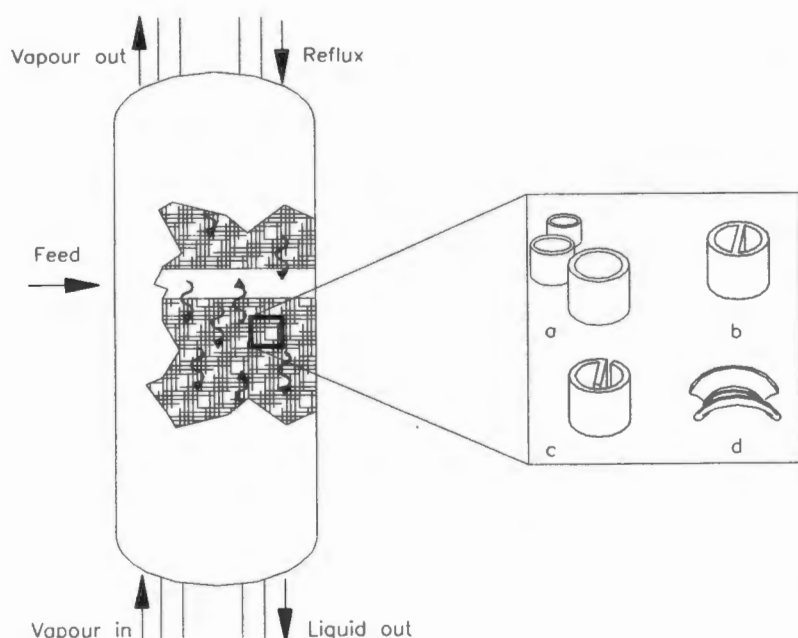
Distillation columns are classified into two general categories, in the way the separation process occurs:

1. the packed tower
2. the plate tower.

In the packed tower the separation occurs in a counter-flow action, between the rising vapour and the falling liquid.

In the plate tower, separation takes place from plate to plate in finite steps.

1.5.2.1 Packed towers



These are cylindrical columns, filled with uniform size particulate solids, Figure 1.7. On the surface of these solids, contact occurs between the liquid and the vapour. The most common shape is the Raschig ring, Figure 1.7(a), a short hollow cylinder, having roughly the same length as diameter [Y1].

Figure 1. 7

These towers are the most popular, because of the low cost of construction, low pressure drop and high efficiency. But if mass flow rates require tower diameters in excess of 800mm, “channelling” occurs between the liquid and the vapour and there is no uniform reflux. Due to the above, the plate-type tower is preferred for large scale operations.

1.5.2.2 Plate towers

These rectifying columns consist of an upright cylindrical shell which has a number of equally spaced horizontal plates, Figure 1.8. The rising vapour passes through the plates, while the liquid flows across the plate by gravity. The depth of the liquid is controlled by a weir positioned at the end of the plate. The liquid flows over the weir and through down-pipes to the plate below. The efficiency of contact between the gas and the liquid on a bubble plate, depends upon complex hydraulic and diffuse phenomena, which in turn are functions of the fluids' properties and construction parameters of the plates [N3]. There are two types of plates which are shown in the diagram of Figure 1.9:

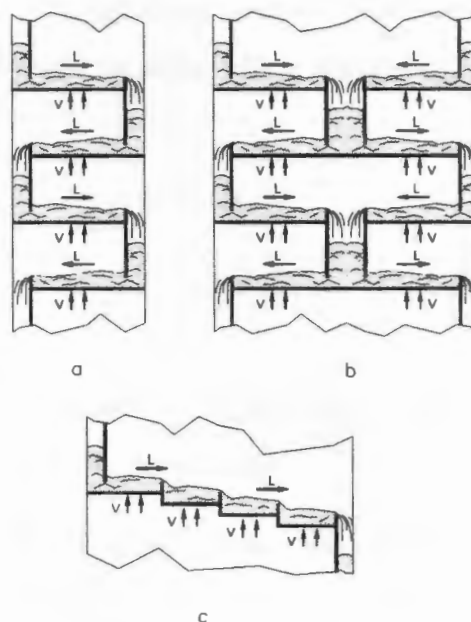


Figure 1. 8

THE BUBBLE PLATE

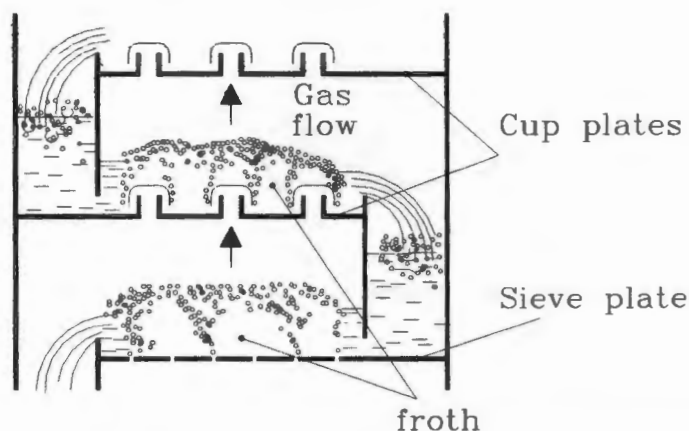


Figure 1. 9

1. the bubble-cup plate which consists of a number of slotted circular cups, positioned over vapour risers.

The bubble-cup distillation towers are efficient over a large range of flow rates, but they are very costly due to the many components involved.

2. the sieve plate is no more than a flat plate punched with holes through which the vapour rises. The liquid flows across the plate and over the holes. The sieve plate has a low cost of construction, but it cannot operate through the same wide rate of flow rates

as the bubble-cup. If the vapour flow rate is reduced by, between 50% to 70% of the maximum flow rate, the flow through the sieve plate becomes unstable. A further decrease, can result in a leakage of reflux liquid through the holes with a simultaneous drop in efficiency.

1.5.3 Solution heat exchanger

The solution heat exchanger is part of the “thermal compressor”. Although it is not a fundamental component for the absorption cycle, it is crucial to its performance.

It is of a simple design, that of the double tube, or a shell and tubes arrangement, with the weak hot solution flowing through the inner tube or tubes, while the cold strong solution flows through the annulus in a counter-current direction, Figure 1.10.

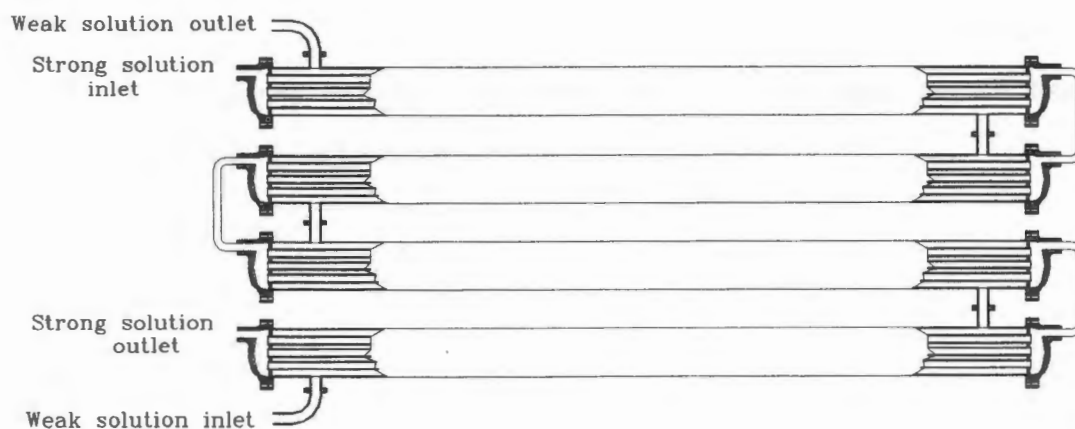


Figure 1. 10 *The solution multi-tubular heat exchanger*

Its function is to transfer heat from the hot weak solution to the cold strong solution and thus to reduce the amount of heat required at the generator and the amount of heat rejected from the absorber.

The design calculations in the past were simplified by assuming, either that there is a constant temperature difference between the hot-end streams [R2], or that the strong solution maintains its concentration right through the heat exchanger [K3, S13]. Practice has shown that it is possible, for the strong solution to boil in the heat exchanger [V2].

1.5.4 Absorber

The absorber is involved in simultaneously occurring processes of mass and heat transfer. It is important, that the vapour pressure of the weak solution in the absorber is less than that of the evaporator. This will occur when the temperature of the weak solution is low. The absorber has the function of ensuring a thorough mixing of the fluids, as well as removing the heat of solution released upon absorption. These functions lead to high efficiency demands on the absorber whose performance is influenced by two main requirements.

Mass transfer requirement:

The absorption process must provide a contact area between the gas and the liquid solution as large as possible, with the ability to:

- increase the interface between the gaseous and liquid phases.
- create a relative motion between the phases, thus increasing the mass transfer.

There are various existing designs of absorbers which provide large contact areas. The classification according to the geometry of the absorber includes:

- a. the packed column
- b. the plate column
- c. the bubble column
- d. the spray column
- e. the surface absorber or film absorbers

With respect to the creation of the interface between the liquid and the vapour, these absorbers can be grouped into three categories, Figure 1.11.

- a. the liquid flows downwards over surfaces in the form of a thin film
- b. the liquid is sprayed into the gas
- c. the gas is injected into the liquid.

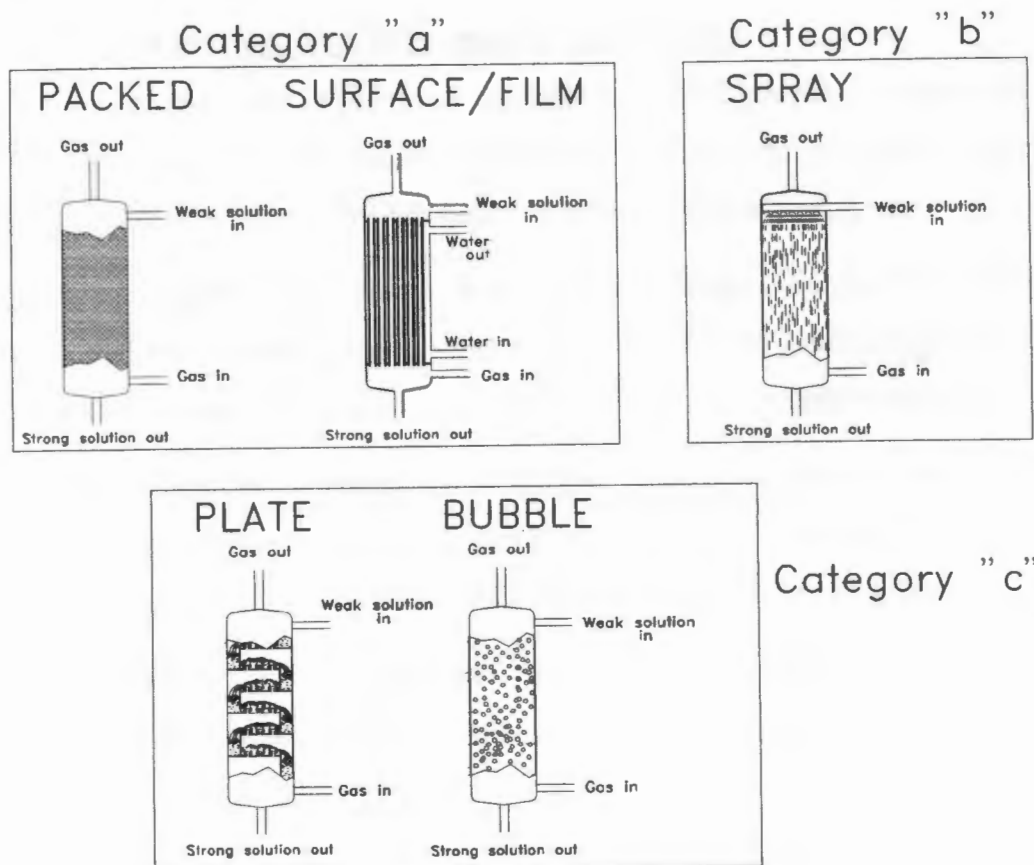


Figure 1. 11

Heat transfer requirement:

The absorber must also provide a large area for heat transfer, because, in-order to satisfy the mass transfer the temperature of the solution must be sufficiently low. Cooling the absorber is essential, but some absorbers, such as the packed and spray columns, might offer difficulties in designing external cooling facilities.

To date, the predominant design of absorbers for refrigeration purposes is the water-cooled type. With the development of the Von Platen Munsters unit for household refrigeration, the air-cooled absorber was introduced, but this is limited only to small refrigeration and air-condition units.

1.6 Simulation models for performance predictions

The design of an absorption refrigeration machine depends upon the environmental cooling conditions, the availability of the supplied heat to the generator, the refrigeration demands on the evaporator and the choice of the refrigerant-absorbent combination.

The independent variables for the design of an absorption refrigeration machine are the cooling environment and the evaporator space temperatures, and the refrigeration capacity. Once the independent variables are set, the generator working conditions are determined [V3].

There is one basic approach in solving the absorption refrigeration system. This is, to evaluate the cycle of the refrigerant and the cycle of the solution. If the absorbent is volatile, there is also the cycle of the entrained absorbent which follows the path of the refrigerant.

Performance calculations for the absorption system were done in the past, using state point property data from graphical and tabulated sources. This is a tedious procedure, and optimisation of a design becomes very difficult.

Jain and Gable [J2] were the first to provide a model which computes polynomial equations for the equilibrium property data of aqua-ammonia mixtures for conditions normally encountered in air-conditioning systems. Two sets of equations were developed covering a high pressure range from 1724.14kPa to 2413.79kPa and a low pressure range from 344.83kPa to 551.72kPa. These pressures correspond approximately to condenser and evaporator temperature ranges from 43°C to 55°C and from -5°C to 7°C respectively. The accuracy becomes questionable, if computations are performed beyond the specified ranges.

Schulz computational model [S2] covers all temperatures ranging from -70°C to 180°C and all pressures from 1kPa to 2500kPa. It is remarked by Keizer in his Ph.D. thesis [K5] that the calculations using Schulz relations, show a large deviation from the well known enthalpy-concentration diagram by Bosnjakovic [K10, T2] especially at concentrations near pure ammonia. This was a surprising conclusion, especially as Schulz used experimental data of many authors to formulate his equations, among which was Bosnjakovic's data too. Despite the availability of the experimental data, Schulz remarked that the data itself was inconsistent. The same conclusions were drawn from authors earlier to Schulz, such as Scatchard et al. [S1] and Macriss [M2] and by Von Stockar in the publication of Stoecker [S12].

Keizer also remarked in his thesis [K5] that, modelling absorption machines is rather an easy task because the heat and mass balances are very simple. This is true as far as the procedure for

the analysis is concerned, and as long as the assumptions made simplify the cycle's operational constraints. He also used Schulz's relations in his computational analysis, despite the inaccuracies found, and makes reference to his research published in a report for the Commission of the European Communities [K6] for an analytical procedure. Reference to this report reveals that the Gibb's free energy was used as it was suggested by Schulz, but no relations were made available for the subroutines.

The author experienced similar deviations for the mixture from subsequent publications by Schulz [S3, S4]. In addition to this, the pure ammonia enthalpy values predicted by Schulz's revised equations were also quite different from the values reported in tables [H1].

The simulation program suggested by Visser [V4] of Delft University, calculates the thermodynamic properties of the aqua-ammonia solutions, but also contains inaccuracies at the near pure concentrations of the components. Inconsistency in the data, especially in the region of pure ammonia can be instrumental of notable variations in the calculation of the refrigeration effect and the performance of the cycle.

1.7 The objectives of this study

There should be a world wide motivation to turn to absorption machines, where low grade heat could be put to use. Yet to date, the author is unaware of a proven model which could be used to design and optimise an absorption unit. Search of the literature on the subject shows that analyses of absorption refrigeration cycles are based on given data from existing plants. The methodology demonstrated serves in evaluating an existing plant, or as an academic exercise. In addition, no attempt is made to evaluate the performance, if parameters are varied. Unlike the compression refrigeration cycle, which is designed and optimised on the basis of refrigeration capacity and evaporator temperature, the approach to the absorption refrigeration is not as straight forward.

For example, one could design a system on the basis of required refrigeration capacity and evaporating temperature; or one could design a system on the basis of available "inexpensive" waste low grade heat as input to the generator.

The scope of this study is to produce a methodology by which an absorption refrigeration plant can be designed.

This will be attempted in two ways:

- The graphical analysis. The cycle will be plotted on an enthalpy-concentration chart, thus establishing the method by which the heat effects will be evaluated.
- The computer simulation procedure. Based on the graphical approach, a computerised model will be developed to predict the heat and mass transfer requirements of the components of an absorption machine, which will be utilised in order to size the plant.

It was felt appropriate to direct this study towards the design and optimisation of the ammonia-water absorption cycle. The experimental verification of the simulation model will be carried out by a laboratory unit to be constructed for the purpose.

Ammonia-water has been selected notwithstanding the comments of paragraph 1.6, because of the availability of thermodynamic properties of both individual components as well as their liquid and vapour mixtures. Ammonia itself offers the highest heat of vaporisation of all known refrigerants and because of its neutral behaviour to the environment (ozone friendly), it makes it a very attractive choice.

CHAPTER 2

BACKGROUND ON THE HYDRODYNAMICS OF TWO-PHASE FLOW
AND HEAT AND MASS TRANSFER CORRELATIONS

CHOICE OF THE TYPE OF ABSORBER

2.0 GENERAL

The term "two-phase flow" covers the interacting flow of two phases, where the interface between the phases is influenced by their motion. Two-phase flow phenomena take place in a wide range of industrial plants and the requirement for an economic design, safety and optimisation of operating conditions, leads to the need for quantitative information [B14]. However, these phenomena are difficult to observe in full scale industrial plants and therefore simulated laboratory set-ups have been used to provide a visual observation. This is done by classifying the two-phase flows in flow regimes, each of which has common features. These features enable one to describe the form of distribution of the liquid and gas-phases. Such a description is necessary before any mathematical model can be constructed to predict a property, for example a pressure drop. But because these studies depend on visual observations, unstable regions of transition make precise definition of flow patterns extremely difficult.

The variables which are considered in two-phase phenomena are the viscosities, densities, mass flow rates of individual phases, interfacial tension between the phases, duct size, shape and inclination and surface roughness. If there is heat transfer involved, the additional variables are the thermal conductivity, specific heat and temperature profile of each phase. If there is mass transfer involved, the diffusion coefficient, the concentration of each component in each phase and the equilibrium relationships between phases in each component are also considered [S5]. In addition, any chemical reaction between the components of each phase adds an enormous difficulty in the derivation of a mathematical model which will adequately address all possible flow regimes [G4].

A process which consists of great complexity is the absorption of gases in the liquid-phase. It is a process of simultaneous mass and heat transfer, in which diffusional mass transfer plays the main role.

The theory of absorption initially concentrated on physical absorption of one substance in a liquid. Early studies have considered two-phase flows with respect to hydrodynamic stability and heat transfer properties. Only in the 1960's a chemical reaction was considered as an important parameter in the absorption process. In this respect, of importance were the investigations by Astarita and Danckwerts in which relations for absorption were accompanied

by a chemical reaction [bib. 49]. During the 1960's, mass transfer also became popular attracting more attention from investigators [C4].

In processes in which there is simultaneous heat and mass transfer, not only mass transfer can cause heat transfer but the opposite can take place. However, interactions of heat transfer on mass transfer are of lesser importance and are usually omitted. In practice simultaneous heat and mass transfer become important when the contacting phases differ significantly in temperature, or the process of solution of a gas in a liquid produces strong thermal effects, such as in the process of ammonia absorption in water.

At present there are no reliable methods which could be used to calculate theoretically the mass transfer coefficient. The existing theoretical models only give an estimate of these coefficients, leaving precision to experimental techniques.

In this chapter some of these models will be discussed, and in particular those which provide equations for the design of the components of the absorption refrigeration machine. These equations are related to heat and mass transfer coefficients and pressure drops in two-phase flows. APPENDICES G, H and I make use of these equations in the evaluation of the mass and heat fluxes involved in the design of the absorber, evaporator and generator respectively.

Reference is made to APPENDIX K where the calculations of the interfacial area from various investigators are compared. These are used in the calculations of the mass transfer coefficient in APPENDIX L.

2.1 THE ABSORPTION PROCESS

Absorption is a phenomenon that involves the transfer of one or more materials from the gas-phase to a liquid solvent [L2]. The mechanism by which a gas is absorbed into a liquid, is distinguished by the effects of mass diffusion action in both the gaseous and liquid-phases. Since the gas is diffusing from its own phase to the liquid, there must be a concentration gradient in the direction of diffusion. Treyball [T5] states that in the gas-phase, the diffusion resistance, which the concentration gradient overcomes, lies within a thin film. It was suggested by Whitman in 1923 that the rate of absorption is controlled by the rate of diffusion on either side of the gas-liquid interface. The rate at which the transfer takes place depends on the solubility characteristics of the gas in the liquid and the physical conditions which bring the substances together. Thus, the geometry of the interfacial contact area, the distribution of both phases within the absorber and the temperature and pressure, are extremely important parameters. When dynamic equilibrium has been reached, the rate at which the gas molecules cross from the gas-phase to the liquid-phase is exactly the same as the rate at which liquid molecules cross from liquid to gas.

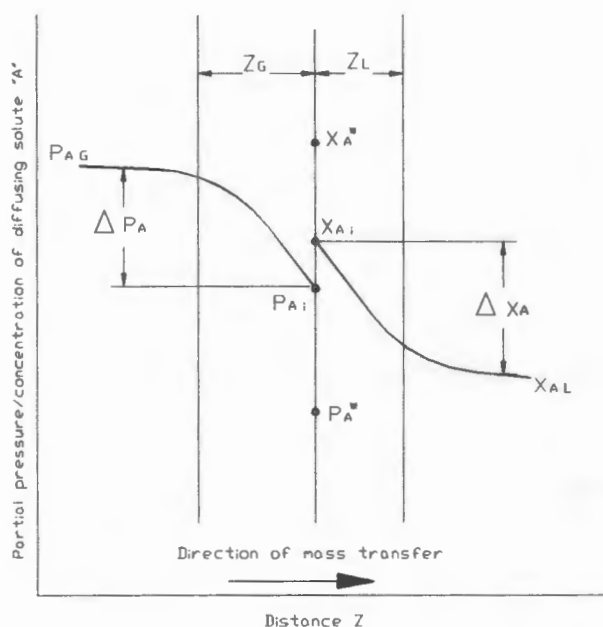


Figure 2. 1 *Partial pressure and concentration gradients at a gas-liquid interface*

Lewis and Whitman (1924) [L2] stated in their two-resistance theory that the transfer is confined to two thin stagnant films on either side of the gas-liquid interface. All the resistance to the mass transfer is concentrated within the two thin films and there is none at the interface itself¹, Figure 2.1. The average concentration of solute A in the gas-phase, in terms of its partial pressure, is P_{AG} and it drops to P_{Ai} at the interface within the thickness of the stagnant film z_G . Similarly the concentration of the solute A in the liquid-phase drops from x_{Ai} at the interface to x_{AL} ,

¹ The apparent discontinuity of the concentration curve is due to the two different concentration units used that is the concentration of the solute in the gas-phase and the concentration of the solute in the liquid-phase. The curve would have been continuous if a concentration measure could be employed to provide equality of numerical values at equilibrium [T5]

the bulk concentration within the distance z_L . Since there is diffusion taking place, the bulk concentrations in the gas and in the liquid, are not equilibrium values. At equilibrium the concentration curves would be flattened to straight horizontal lines.

At the interface the gas and liquid concentrations are in equilibrium and obey Henry's law which states that the amount of gas that dissolves in a given quantity of liquid is directly proportional to its partial pressure over the solution.

$$p_A = H_A \cdot x_A$$

where

p_A = the equilibrium partial pressure of the solute A at the interface

H_A = Henry's law constant

x_A = the equilibrium mole concentration of the solute A at the interface, in the liquid-phase.

The above relation can be expressed in terms of the concentration values in the gas and in the liquid-phases as:

$$(2.1) \quad y_{Ai} = m \cdot x_{Ai}$$

where

y_{Ai} = mole concentration of the solute in the gas at the interface

x_{Ai} = mole concentration of the solute in the liquid at the interface

$m = H_A/P$ where P and p_A obey the definition $p_A = y_A \cdot P$

Therefore the driving force for diffusion within the phases is related to pressure difference or concentration difference.

The measure of resistance to this driving force is a constant k , called a mass transfer coefficient such that:

the amount of mass

transferred through = k (interfacial area). (concentration difference)

an interfacial area

The k coefficients can be identified for the gas-phase by the subscript "G" and for the liquid-phase by the subscript "L".

Thus if N_A is the flux of mass transferred, then:

$$k_L = \frac{N_A}{x_{Ai} - x_A} \quad k_G = \frac{N_A}{p_A - p_{Ai}}$$

The ratio of these " k " coefficients is a slope in a p_A - x or y - x diagram indicating the driving force line, upon which the resistances to diffusion act, eq. (2.2):

$$(2.2) \quad \frac{p_{AG} - p_{Ai}}{x_{Ai} - x_{AL}} = -\frac{k_L}{k_G} = -\frac{\frac{1}{k_G}}{\frac{1}{k_L}} = \frac{\text{gas-phase}}{\text{liquid-phase}}$$

The ratio of the resistances to diffusion, the slope of value " m " from eq. (2.1) and the interfacial compositions identified by the resistances ratio are shown in Figure 2.2

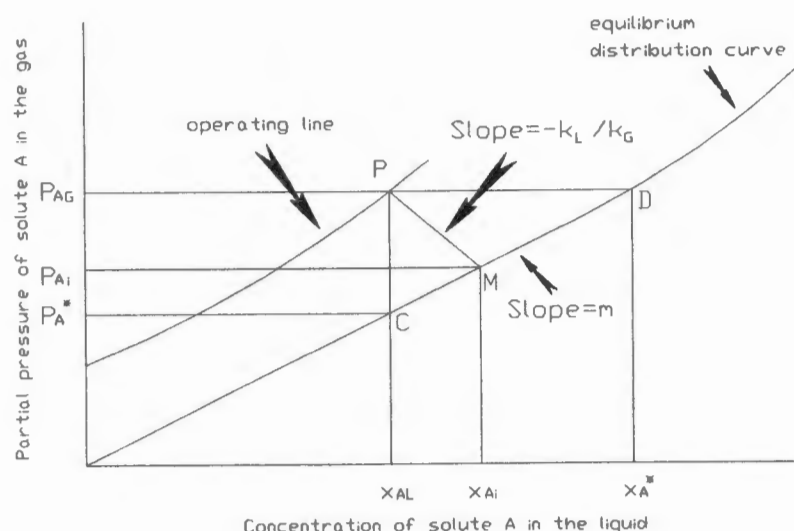


Figure 2.2 Composition at an interface between phases

Since it is not possible to measure the partial pressures and concentrations at the interface, it is convenient to introduce overall mass transfer coefficients based on the overall driving forces between p_A and x_A , eq. (2.3).

$$(2.3) \quad K_L = \frac{N_A}{x_A^* - x_A} \quad K_G = \frac{N_A}{p_A - p_A^*}$$

where

p_A^* = the partial pressure in equilibrium with the bulk concentration x_{AL}

x_A^* = the concentration in equilibrium with the bulk partial pressure p_{AG}

[T5] showed that there is a relation between the “K” and the “k” coefficients, such that:

$$(2.4) \quad \frac{1}{K_L} = \frac{1}{m_G k_G} + \frac{1}{k_L} \quad \frac{1}{K_G} = \frac{1}{k_G} + \frac{m_L}{k_L}$$

Each term of the right hand side of eq. (2.4), represents the relative resistance to diffusion within the individual phases.

The values of “m”, “ k_G ” and “ k_L ,” as well as the ratios by which they appear in eq. (2.4) play a significant role in determining the equipment used for the absorption. For example if “m” is small, a small partial pressure in the gas would support a large concentration in the liquid. This means that the solute is very soluble in the solvent. For this type of absorbent-refrigerant combination, showering droplets of liquid down a continuum of gas would be more effective than bubbling the gas through a continuum of liquid. This is because the gas inside a bubble is stagnant, with mass transfer occurring through molecular diffusion. This gives rise to the gas-phase resistance. The overall resistance to mass transfer will drop if the gas-phase resistance is reduced by agitating the gas with a rain of liquid drops. Also the resistance to diffusion of the liquid drops is small due to the small value of “m”. However, other considerations must be taken into account when designing an absorber, such as the presence of thermal effects generated by absorption.

If the time of contact, or the interfacial area is not sufficient to provide equilibrium between the phases leaving the absorber, then the concentrations will be $y_A > y_{Ai}$ and $x_A < x_{Ai}$

where x_{Ai} and y_{Ai} are equilibrium line concentrations, as shown in Figure 2.2.

Therefore, depending upon the substance of interest, the absorber's efficiency can be defined either as:

$$(2.5) \quad \eta_{ABS} = \frac{x_A - x_{inlet}}{x_{Ai} - x_{inlet}} \quad \text{to produce a liquid in solute}$$

or

$$\eta_{ABS} = \frac{y_{inlet} - y_{Ai}}{y_{inlet} - y_A} \quad \text{to produce a mixture of gases leaner in solute}$$

and is rewritten in terms of the nomenclature employed, as:

$$\eta_{ABS} = \frac{x^{ST} - x^{WE}}{x^{ST}_{ideal} - x^{WE}}$$

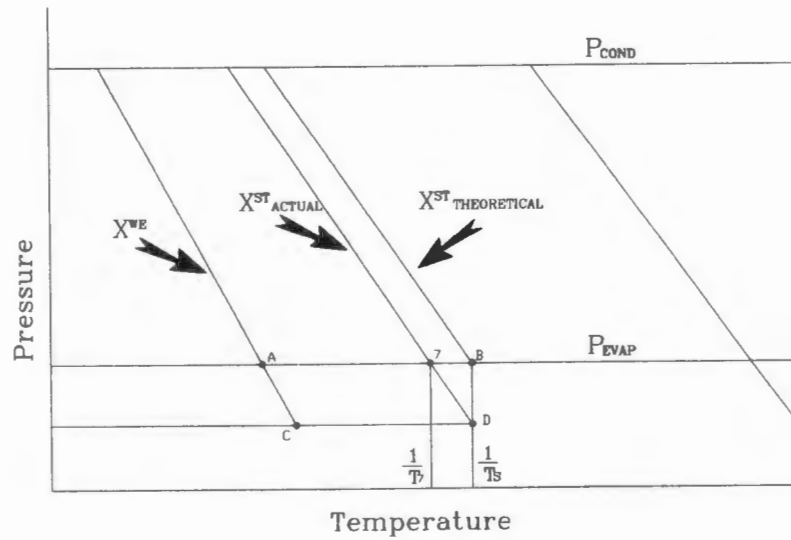


Figure 2.3 The P-T-x diagram of the non-equilibrium absorption process. Point 7 refers to the conditions of the solution at the exit of the absorber.

The actual non-equilibrium absorption process as well as the ideal equilibrium process are drawn in a P-T-x diagram and shown in Figure 2.3. For the cooling available at a sink temperature T_s , the ideal process proceeds from point A towards point B which would give the theoretical concentration of the strong solution. The non-equilibrium process proceeds from a point C towards point D.

2.2 ABSORBERS AND TWO-PHASE FLOW ANALYSIS

This section deals mainly with the two-phase flow through tubes. Due to the absorber's delicate design requirements, it was felt appropriate to include together with this analysis, the discussion on absorbers in terms of the gas-liquid interface. This analysis however, holds true for all the other components which operate with a two-phase flow pattern.

The predominant design of absorbers for refrigeration purposes is the water-cooled type.

They can be classified into two categories with respect to the fluid mixing and to the way of cooling [A1].

1. Tubular absorbers
 - a. Horizontal
 - b. Vertical
2. Surface or film absorbers

2.2.1 Tubular absorbers

The functions of the tubular absorber are far from simple. The mixing of the gas and liquid-phases is performed in a variety of ways, creating a difficulty in establishing a governing equation to define the kind of flow which prevails in the tubes. This difficulty is enhanced by the mass and heat transfers between the phases.

The perfect absorber must combine the two phases (liquid and gas) into a single liquid-phase at its outlet. Upon saturation conditions, there will be no further absorption and the entering gas will simply flow through a liquid medium. The analysis of this latter condition has facilitated the understanding of the different liquid/gas flows in tubes. When there is no mass and heat transfer involved in the two-phase flow, air bubbling through water can be regarded as a representative experimental medium for absorbers which have reached saturation conditions. In this case, the variables involved are the velocities, densities and mass flow rates of the individual phases, surface tension between the two phases, acceleration of gravity and buoyancy which are influenced by the inclination of the tube. The tube characteristics are therefore also important variables, such as diameter and inclination.

When mass and heat transfers are involved during the flow, then in addition to the above variables, the transfer properties must also be considered, [S5]. These are:

- the diffusion coefficient in each phase of each constituent
- the concentration of each constituent
- the equilibrium relationships between the phases
- the thermal conductivity and specific heat of each phase
- the temperature conditions.

This large number of variables indicates the degree of complexity of the two-phase flows. Despite this complexity, there are methods which predict the flow characteristics and classify them into regions based on visual observations or pressure drop behaviour. These predictions have been grouped for two distinguished types of tubular absorbers, namely the horizontal and the vertical types.

2.2.1.1 Horizontal tubular absorbers

These absorbers consist of an array of horizontal double tubes connected in series and arranged in a vertical plane. The weak solution and the vapour are introduced at the lower end of the inner tube and a gradually changing two-phase flow delivers the strong solution at the top end. The cooling water flows in a counter-current direction in the outer tube.

The multi-tube absorber of Figure 2.4, is an improvement on the double tube design with respect to the heat transfer from the solution to the coolant. An improvement on the mass transfer is achieved by the introducing the vapour from several points along the absorber. These absorbers are constructed in high stacks of tubular components, which have as major disadvantage the pressure drop between the bottom and the top ends, due to both hydrostatic and friction losses. A consequence of this pressure drop, which can be as big as 0.5bar, is a decrease in concentration of the strong solution.

HORIZONTAL MULTI-TUBULAR ABSORBER

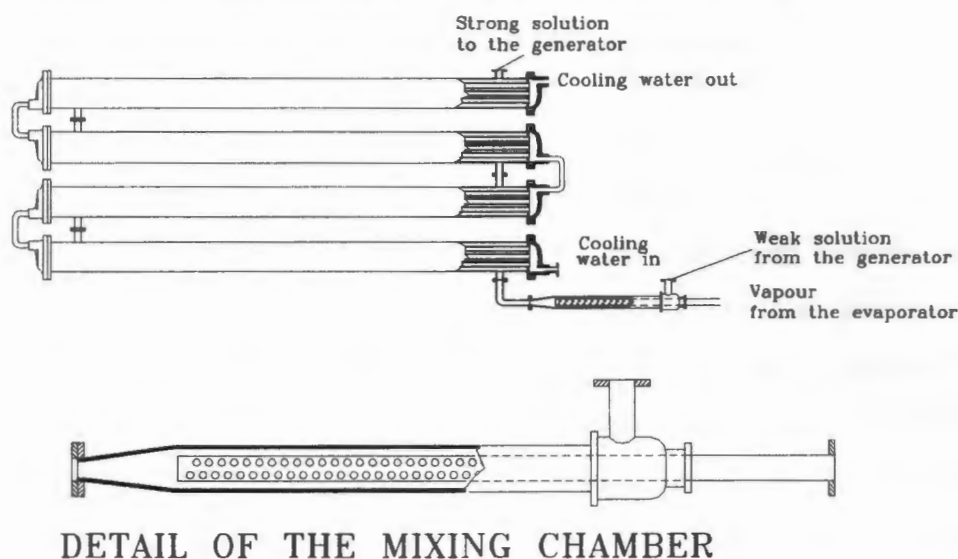


Figure 2. 4 *A typical horizontal multi-tubular absorber*

The horizontal tubular absorbers were the first to appear in the gas absorption industry, but little is known about the gas-liquid flow behaviour through the tubes. However, if there is no mass transfer between the two phases, then flow patterns can be recognised. These patterns could fall under classifications based either on detailed theoretical analysis and data, or on visual observations. The most common way of classifying the flow patterns is the visual observation, and according to Alves [A7, A8] the majority of flow patterns for two-phase flow can be described adequately. The phases shown in Figure 2.5 are as follows:

- Bubble flow

Well-defined bubbles move at the upper surface of the pipe at the same speed as the liquid. If the speed of the liquid increases, the bubbles disperse and the flow is characterised as "froth flow".

- Plug flow

As gas flow rates increase the bubbles coalesce forming large 'bullet-shaped' bubbles which fill the major part of the cross-sectional area of the pipe.

- Stratified flow

At low liquid and gas rates there is a constant smooth interface between the liquid and the gas. This is a common situation in large pipes.

- Wavy flow

If in the stratified flow the gas rate increases, waves are formed on the surface of the liquid.

FLOW PATTERNS IN HORIZONTAL TUBES

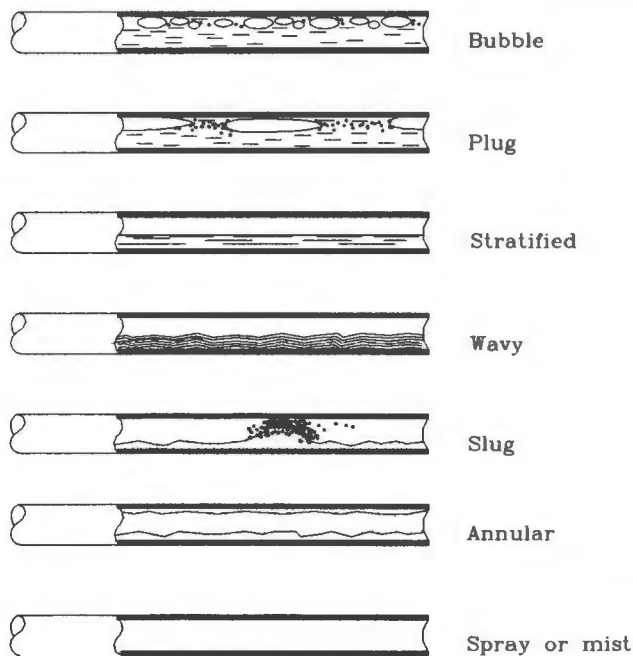


Figure 2. 5 *Two-phase flow in horizontal tubes*
[S5]

- Slug flow

In gas flow rates beyond those which produce wavy flow, the waves increase in amplitude and are picked up by the rapidly moving gas to form frothy slugs. Slugs can also be formed starting from the plug condition when the gas flow rate is increased at a constant liquid flow rate. The speed of the slug is much higher than that of the liquid.

- Annular flow

This pattern is also named as “film flow” because of the thin layer that the liquid has formed inside the tube. The gas flows at a high speed in the core and carries some of the liquid as a spray.

- Spray flow

When the gas flow increases to even higher relative speeds the annular

formation diminishes until all the liquid is entrained in the gas.

Based on these descriptions and on known flow variables, empirical charts have been prepared by which the state of the flow can be predicted. In these charts, contours outline the regions in which the various flow patterns are expected to occur, but in some charts this is not accurate. Some of the contours have been prepared from a small number of flow parameters. Often correction factors for fluids of different densities, viscosities and surface tensions are required.

The chart proposed by Hoogendoorn and of Hoogendoorn and Buitelaar [H7, H8] has considered variables emphasising a volume based arrangement of flow areas. But important ranges of flow behaviour are squeezed together in a small area of the chart rendering the shape of the patterns inconvenient. A more useful chart is that of Beker's, modified by Scott (1966), which is in agreement with the latest data of Hoogendoorn and Govier and Omer.

Beker's chart uses variables emphasising a mass basis arrangement of the patterns and plots the gas-mass velocity against the liquid to gas-mass velocity feed ratio. The correction factors "λ" and "ψ" which appear in the definitions of the axes in Figure 2.6, are used for variations in densities, and for surface tension and liquid viscosity respectively.

These factors are defined as [S5]:

$$\lambda = \frac{\rho_G}{0.075} \sqrt{\frac{\rho_L}{62.3}} \quad \psi = \frac{73}{\sigma} \left[\mu_L \left(62 \frac{3}{\rho_L} \right)^2 \right]^{\frac{1}{3}}$$

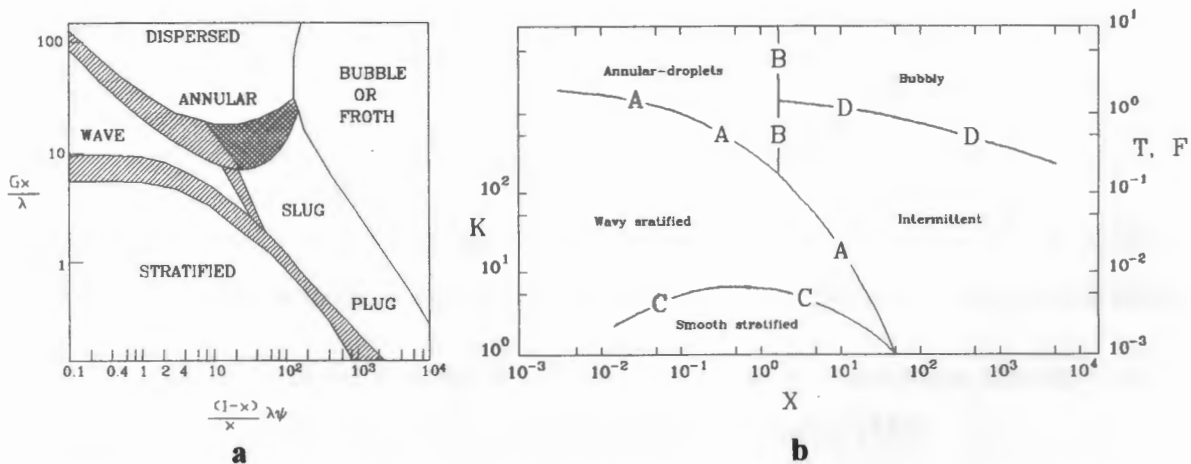


Figure 2. 6a Baker's flow pattern chart, modified by Scot, for two-phase concurrent flow in horizontal tubes. The shades areas are transitional zones

Figure 2. 6b Taitel and Dukler's two-phase map for horizontal flow

Similar patterns are recognised in the chart of Figure 2.6b, proposed by Taitel and Dukler [T1] in which pressure drops and momentum fluxes characterise the different regions of two-phase flows.

The co-ordinates are defined as follows:

- the abscissa is the ratio of the liquid to vapour single-phase pressure drops $(dP/dz)^{SP}$
- the ordinates are the ratios of pressure drops and momentum fluxes:

$$X = \left[\frac{\left(\frac{dP}{dL} \right)_L^{SP}}{\left(\frac{dP}{dL} \right)_V^{SP}} \right]^{\frac{1}{2}} \quad T = \left[\frac{\left(\frac{dP}{dL} \right)_L^{SP}}{(\rho_L - \rho_V) \cdot g} \right]^{\frac{1}{2}} \quad F = j_V \cdot \left[\frac{\rho_V}{(\rho_L - \rho_V) \cdot g D} \right]^{\frac{1}{2}} \quad K = \left[\frac{\rho_V \cdot j_V^2 \cdot j_L}{(\rho_L - \rho_V) \cdot g v_L} \right]^{\frac{1}{2}}$$

where

$$j_V \text{ and } j_L \text{ are the vapour and liquid superficial velocities defined as: } j = \frac{m}{\rho \cdot A}$$

The curves are plotted with the following co-ordinates:

- curves A and B: X Vs F
- curve C: X Vs K
- curve D: X Vs T

These maps are based on experimental data for adiabatic flows. In evaporating or condensing flows where the heat transfer is significant, the predicted flow regimes can become unreliable.

However, knowledge of the mass quality of the condensing or evaporating substance can give an indication of the design parameters of the components involved [M6].

2.2.1.2 Vertical tubular absorbers

The study of the flow in vertical tubes starts with the invention of the airlift pump [bib. 33]. Its application in the 1920's to the pumping of oil from oil wells, supported a number of mathematical and experimental studies.

The vertical tubular contactors provide the interface area between the liquid and the gas in a different manner to that of the horizontal tubular contactors. The flow patterns in vertical tubes are not as clearly defined as in horizontal tubes. The influence of gravity causes a back flow of liquid at relatively low gas speeds, dispersing the flow pattern. Difficulty in establishing a visual definition of flow patterns also arises from the varying composition of the gas-liquid mixture, even when the flow has reached steady state conditions. This phenomenon is referred to under the headings of “slip” and “hold-up” [G2] where the gas tends to slip past the liquid. Slip is defined as the difference in the linear velocities of the gas and liquid-phases, measured at the same average temperature and pressure. Hold-up is defined as the ratio of the gas-liquid volume ratio in the supply mixture to the gas-liquid volume ratio in the flow section.

The two-phase flow in vertical absorption columns is divided into two categories. In large diameter columns the prevailing patterns are those of froth or bubbles, such as in the bubble-cup, sieve plates and bubble columns, as shown in Figure 1.11 category “c”.

The flow pattern in small diameter tubes (smaller than 50mm), depends on the gas to liquid volume ratio. These patterns based on visual observation, are:

- Bubble-flow

The gas is dispersed in a continuum of an upward flowing liquid as bubbles of various sizes.

- Slug-flow

The gas flows as large bubbles, almost filling the tube. These bubbles are bullet-shaped and separated by liquid. The velocity of the slug depends on the distance between the slugs and it assumes a constant asymptotic value when this distance is large enough to support potential flow. When the distance is smaller than a critical value, the wake of the leading bubble influences the trailing bubble to rise faster and eventually to coalesce. These are often called Dumitrescu or Taylor bubbles [G4, D1]. This kind of flow is resembled when a liquid is emptied from a vertical tube.

As the slugs rise, the liquid flows down through the liquid annulus surrounding the slug.

- Froth-flow

This is a turbulent mixture of degenerated bubbles and liquid without a pattern. It is formed when the tendency of the liquid to fall around the slugs has nearly stopped and the slug has become unstable.

FLOW PATTERNS IN VERTICAL TUBES

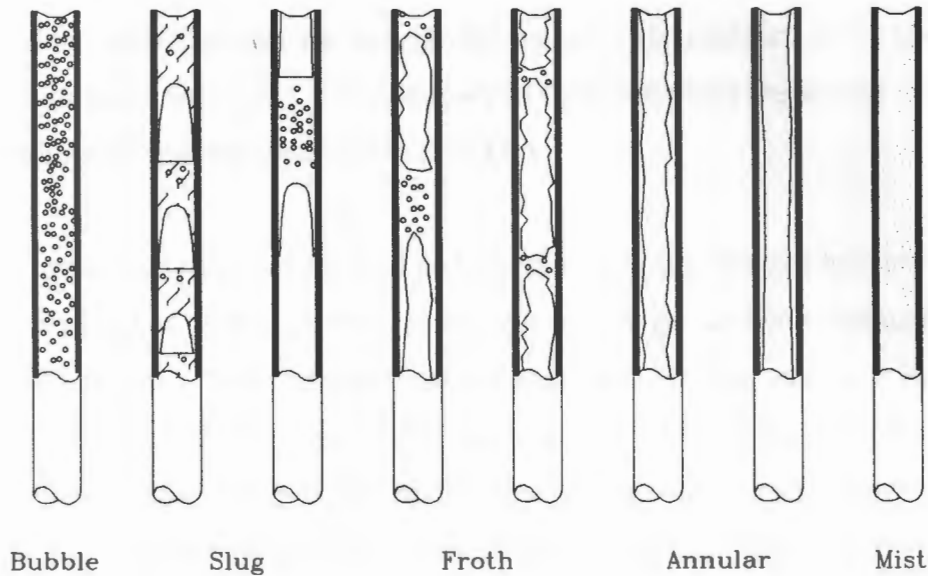


Figure 2. 7 *Two-phase flow in vertical tubes*

- Annular-flow

In this flow the liquid is not the continuum any more. The gas travels at high velocity at the core of the tube, while the liquid forms an annulus around the tube walls. At the lower end of gas velocities the annulus is thick and wavy. But as the velocity increases, the thickness of the liquid annulus diminishes and the amount of liquid entrained in the gas flow as droplets increases.

- Mist-flow

This flow is characterised by the total dispersion of the liquid in the gas. This happens at very high gas flow rates, where the liquid annulus has disappeared and all the liquid is carried with the gas as spray or mist.

In most industrial and refrigeration applications where vertical absorbers are used the flow pattern is of the bubble, slug or froth type. Therefore it would be helpful to have values of the

interfacial area and the mass transfer coefficient between the gas and the liquid for these flow regimes. The available information is insufficient and incomplete with respect to the influence of solutes on the system's behaviour. Specifically, the size of the bubbles and hence the hold-up of gas and the interfacial area are strongly influenced by the tendency of the smaller bubbles to coalesce into larger ones [D2]. This tendency is enhanced when the solvent or the liquid is pure and the bubbles coalesce much faster [D3, D1].

Due to the complicated nature of the flow and the various flow patterns which are encountered in vertical tubes, it has been impossible to express in a single mathematical model all of the two-phase flow regimes. Some studies therefore have been concentrated on a particular flow pattern, such as that of Griffith and Wallis who report on a two-phase slug flow in vertical tubes [G4]. Their mathematical model of the flow is based on a visual observation of that regime. It is also based on the study of Dumitrescu [D2] and of Davis and Taylor [D1] who considered slug flow bubbles rising in a liquid when that liquid is emptied from the bottom of a vertical tube. They supported their theory by equations for the bubble velocity as well as the spherical shape of the top of the bubble. The authors of [G4] expressed the length of the slug bubble with an empirical equation and claimed that mathematical models can be applied to all flow regimes, without the necessity of any visual observations, once the flow variables are known.

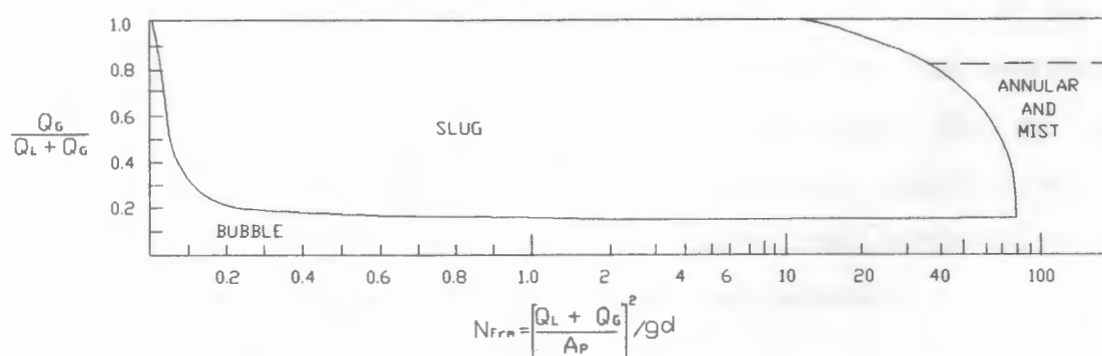


Figure 2. 8 The flow regime map for air-water two-phase flow at 1 atm in vertical tubes

According to the standard classification of the flow regimes which is based on visual observations, Figure 2.7, Griffith and Wallis prepared a flow map for the prediction of flow

regimes using the co-ordinates suggested by Koslov [K9]. In this flow chart, all possible flow patterns are “lumped” into three areas classed as bubble, slug and annular-mist flows, Figure 2.8. It is quite surprising, as remarked by Brown and Govier [B12] that the “froth” flow is omitted from the map, although according to Koslov, the froth flow is a marked region of stable flow between the slug and annular flows.

The proposed map by Hewitt and Roberts [B14] shown in Figure 2.9, gives a more detailed definition of the flows and distinguishes all five patterns. The map correlates on the same basis, data for air/water at atmospheric pressure and data for steam/water at high pressures. The relationship is between superficial momentum fluxes of the gas on the y-axis and of the liquid on the x-axis.

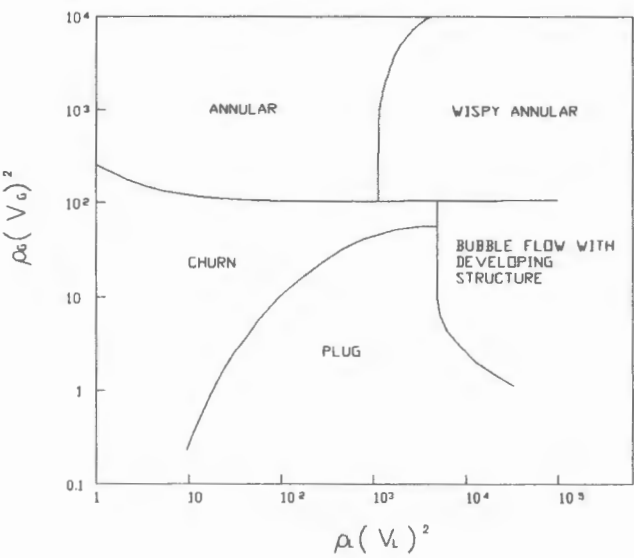


Figure 2. 9 *The map by Hewitt and Roberts [B14] (1969) for two-phase vertical flow*

Govier et al. [G2] classify the flow regimes based on pressure-drop behaviour rather than on the visual observations. This method of describing the flow patterns was originated by Radford and Dunn (1949-1952) and used later by Calvert and Williams (1955). It offers a greater definition and clarity of the flow regimes, than the visual observations of the flow. Despite this, the various pressure-drop regions were still explained using terminology according

to the widely preferred and understood classical way of visual observation.

Grovier et al. experimented on air-water mixtures and showed the effect of the air-water ratio on the flow pattern, the hold-up and the pressure drop.

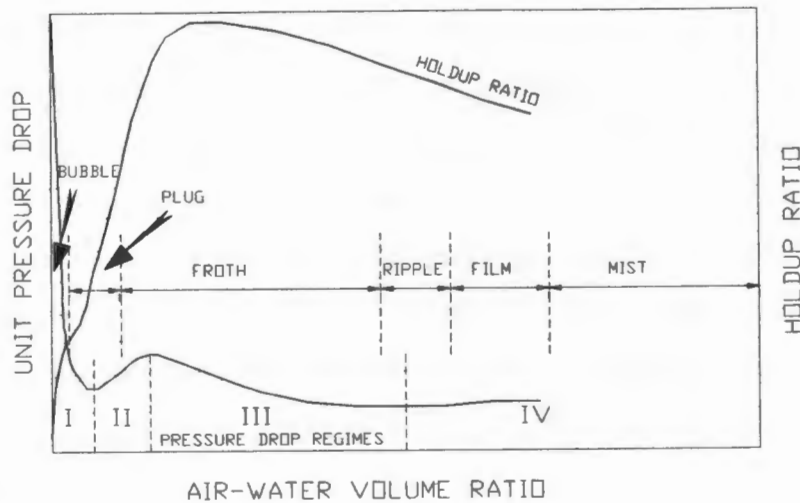


Figure 2. 10 Typical pressure drop relationship and flow regions as proposed by Grovier et al. [G2]

The equations and family of graphs which were obtained for different water rates, clearly indicate regions between maxima and minima which separate the various flow patterns. The patterns which could be predicted by the charts, Figure 2.10, were from bubble to annular. The mist flow regime was not included in their study.

The variables which were considered in their investigation are the following:

Liquid and gas properties

Density " ρ_L ", " ρ_G "

Viscosity " μ_L ", " μ_G "

Surface tension of liquid with respect to gas " σ "

Geometrical properties

Tube diameter " d "

Tube roughness " δ/d "

Tube height " z "

Flow rates

Mass flow rate of the liquid " m_L "

Mass flow rate of the gas " m_G "

The work of Griffith and Wallis [G4] on a two-phase slug-flow, formed the basis for further investigations, such as Moissis [M7]. Their work on the transition from slug to homogeneous two-phase flows stresses the fact that the existing flow charts rely on the conditions and observations of experimental work. These conditions do not consider flow development,

addition of heat, or varying surface behaviour and the transition lines between two-flow regimes are in fact the mean lines through the transition bands. His mathematical interpretation is based on a system of equations, one of which relates to the length of the bubble. The bubble length depends upon many parameters, such as the inlet conditions, the purity of the fluid, the gas and liquid absolute velocities (relative to the ground). Since a given slug-flow system consists of bubbles of various lengths, it is unreasonable to give a particular equilibrium value to the bubble length. Therefore transition between two regimes should appear as a band and not as lines, as confirmed by Baker's chart in Figure 2.6.

The bubble, slug and froth-flows are the types of flows that exist in tubes, which constitute the modern absorbers for refrigeration purposes. These tubular absorbers offer higher performance and meet the two requirements mentioned in paragraph 1.5.4. Research on these absorbers by Infante Ferreira et al. [I2] revealed that for ammonia-water mixing, 65% of the absorption bubble column is slug-flow and about 20% is froth-flow. The remaining is described as transitional flow patterns. The froth-flow is concentrated at the entrance region, changing into a developed slug-flow. The results from preliminary experiments considering mass transfer in adiabatic conditions [I2] were in close proximity with those proposed by Nicklin [N1]. The main study involved simultaneous heat and mass transfer. As provided by Infante Ferreira et al. [I2] the overall heat and mass transfer coefficients and their dependence on flow parameters, enables the design of the bubble absorbers for a range of flow requirements and operating conditions.

2.2.2 Film absorbers

The falling film absorbers have been used in absorption refrigeration machines since 1935. The weak solution, supplied from the top of the absorber, falls as a thin film of liquid, while the vapour, supplied from the bottom of the absorber ascends. The resultant strong solution is collected from the bottom of the absorber, Figure 1.11 category "a". The liquid film is formed on either the inside or the outside of tubular surfaces in the absorber, which are in either a vertical or a horizontal arrangement.

Tubular film absorbers are to be preferred over other geometries, such as packing, as the tubes provide an easy arrangement for cooling purposes. The film absorber provides an extremely

thin liquid layer and mixing, results from film undulations.

Because of the very thin liquid layer, heat has to travel a short distance. The transfer of heat between the liquid film and the tube wall, as well as the heat and mass transfer rates at the gas-liquid interface are very high. This is advantageous to the development of air-cooled falling film absorbers. At present, large absorbers are water-cooled to avoid increased heights. A limitation of about 6m exists due to hydrodynamic instability of the film, with the cooling being administered by water flowing in a counter-flow direction to the falling film.

To improve the film stability in vertical tubular film absorbers, the design calls for the liquid film on the inner surface of the tube rather than the outer. However, it is difficult to distribute the liquid over a large number of tubes when homogeneous films and large liquid-flow rates are desired.

The vertical film absorbers do not suffer from hydrostatic pressure drops. The only pressure drop that exists is due to friction between the film and the tube and it is of a negligible value [K5]. The main disadvantage of the film absorbers is film instability, triggered by slight deviations from the vertical positioning of the contacting surfaces and by surface irregularities, such as scale or deposits on the walls. With respect to the positioning of the surfaces, Blass [B5] has remarked that if the tubes are not in a vertical position it will be impossible to obtain a stable film. In addition, if the density of the wetting weak solution is low, it will also lead to an unstable film, which will then break into individual jets. This is caused by the action of tension forces [H2] and several criteria have been derived for a "minimum wetting rate" from the forces acting on the liquid film.

"Although non-wet patches on surfaces were a frequent observation in the past, there was no definite explanation given, except that the premature breaking of the film was due to the latent heat of solution".

These observations were the forerunners in the process of gas absorption. Chilton, Duffey and Vernon [B8] found unpredictable results in the ammonia absorption due to channelling of the liquid, although the liquid was initially homogeneously distributed. Molstad and Parsly [M9] also experienced the same channelling problem during the absorption of ethanol vapour in water. They found that, when absorbing the vapour at different water rates the wetted area of a

packed tower varied from 10-20% to 80-90%. However, they provided no explanation for the phenomenon.

Only Bond and Donald [B8] established that hydrodynamic film stability depends on the heat and mass transfer processes. It was observed that during absorption the surface tension decreased leading, to film break-up and dry patches on the tube surfaces to appear. This phenomenon is known as the **Marangoni-effect** [B8] and results from the influence of both the temperature and concentration on the surface tension of the falling film, during the absorption process. The effect is stronger at the film entrance, where the change in temperature and concentration is the largest. It tapers off as the liquid approaches equilibrium with the gas stream, in which case its temperature and concentration is maximum and constant. The results of Haselden and Malaty [H3] contradict those by Bond and Donald. They experimented on the film stability on inner and outer tube surfaces and indicated that the inner tube surfaces contribute to a higher surface tension which favours a stable film formation. On the outer surfaces, a stable film may be achieved if the lower surface tensions can be combined with the "rippling" of the liquid film. But it was concluded that a considerable difference existed between natural rippling and rippling of the film caused by violent absorption. This statement is in agreement with the conclusions drawn by Bond and Donald that rippled surfaces can result in premature breakdown of the film.

Haselden and Malaty [H3] tested stainless steel and mild steel tubes in an attempt to find surface conditions which would maintain film stability at low flow rates. Experiments on the stainless steel surfaces showed early deterioration of the film. The film stability was improved temporarily if the tube was degreased, roughened, heated and quenched in water. Mild steel tubes proved to be more effective, even more so when rust was formed and heated and then quenched in water. Although rust formation proved to give good wetting properties, it contaminated the surface of the tube. In addition to the burning of organic deposits, the heating and quenching caused a layer of an oxide to be formed which had a greater affinity with the liquid. This oxide acted as a surface agent which suppressed the natural rippling. This is also in agreement with the observations by Bond and Donald who experimented with surface agents and found that extremely low water rates can be achieved without breaking down of the water film. With absorption, a violent rippling of the film reduced the minimum wetting rate.

2.3 HYDRODYNAMICS OF TWO-PHASE FLOW IN VERTICAL TUBES

According to Griffin and Wallis [G4] and Nicklin [N1] the two-phase flow patterns most frequently researched are bubble, slug and annular-flows through vertical tubes.

By definition of the void fraction “ ϵ ”, being the total volume of the gas bubbles per unit volume of tube, the rising velocity of the gas bubble in a liquid is given by:

$$u_s = \frac{Q_G}{\epsilon \cdot A}$$

where

Q_G = volume of gas-flow per unit time

A = cross-sectional area of the tube

This is a basic equation which can be applied to either bubble or slug-flows, regardless whether the liquid has a free surface, is stationary or has a finite velocity.

It has been shown by Nicklin et al. [N2] that for a slug-flow, the rising velocity of a single slug in a stationary liquid relative to the tube wall is:

$$(2.6) \quad u_s = \frac{Q_G}{\epsilon \cdot A} = 0.35\sqrt{gD}$$

It was also shown that this equation describes the rising velocity of the slug relative to the velocity of the liquid ahead. If the flow of the liquid has a velocity relative to the tube wall ahead of the nose of the slug, then the velocity of the slug relative to the tube wall is that described by eq. (2.6), plus a component to account for the net liquid velocity. If the net of liquid-flow is downwards, the slug shape is distorted in an effort to avoid the fastest moving liquid at the centre of the tube [S5, N2].

If multiple slugs exist in a tube, eq. (2.6) is modified to account for the non-uniform velocity of the liquid between the slugs [N1]. The average slug velocity becomes:

$$(2.7) \quad u_s = \frac{Q_G}{\epsilon \cdot A} = 1.2\bar{u}_{LS} + 0.35\sqrt{gD}$$

where

$\bar{u}_{LS}$ = average total liquid superficial velocity

This relation was also supported experimentally by Nicklin [N1] for turbulent upward liquid-flow, for superficial liquid Reynolds number over 8000.

The void fraction “ ϵ ” in eq. (2.7) can also be expressed in terms of the liquid and gas velocities as the ratio of the area occupied by the gas and the total area of the tube, that is:

$$(2.8) \quad \epsilon = \frac{A_G}{A} = \frac{Q_G u_G}{Q_G u_G + Q_L u_L} = \frac{1}{1 + \frac{Q_L}{Q_G} \frac{u_G}{u_L}}$$

The ratio u_G/u_L is termed slip ratio, while the value of $u_G - u_L$ is the slip velocity.

For a homogeneous flow (no slip between the phases, $u_G = u_L$), the void fraction becomes:

$$(2.9) \quad \epsilon = \frac{Q_G}{1.2(Q_G + Q_L) + 0.35A\sqrt{gD}}$$

and the average slug velocity is given by:

$$(2.10) \quad \bar{u}_s = \frac{Q_G}{\epsilon \cdot A} = 1.2 \frac{Q_G + Q_L}{A} + 0.35\sqrt{gD}$$

This equation reveals that if the liquid-flows co-currently with a slug or with a bubble, then the average velocity of the gas will be higher than the basic velocity given by eq. (2.6). Nicklin [N1] explained that a basic velocity is given to each phase as the gas flows into the tube at the entrance. In addition, there is the buoyancy velocity component which causes the bubble to rise faster than the basic velocity by an amount u_o . This velocity u_o has a magnitude:

$$u_o = 0.2 \frac{Q_G + Q_L}{A} + 0.35 \sqrt{gD}$$

This explains the observation at the entrance of a tube that bubbles tend to coalesce into bigger bubbles forming slugs which tend to coalesce further. Griffith and Wallis [G4] explained that at the tube entrance or at a non-fully developed slug-flow, the wake of the first bubble gives rise to a liquid velocity profile which causes the second bubble to rise faster. Moissis and Griffith [M8] took measurements of the velocity profile in the tube at the wake of a rising slug and found that at a distance between slugs of about 5 tube diameters all disturbances of the liquid velocity profile had died out. At this distance there will be no coalescence between the slugs which will rise with a constant velocity.

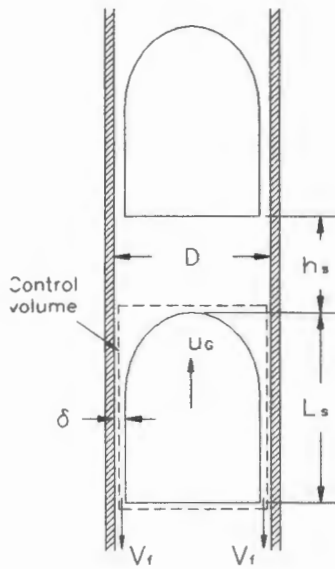


Figure 2.11 The slug-flow model in vertical tubes [M7]

In the paper by Moissis [M7], the rise velocity of a bubble was shown to be in agreement with that predicted from eq. (2.10). Moissis, extended the model further to include tube entrance effect, where each bubble is influenced by the bubble ahead. In his paper a developing flow where Taylor bubbles agglomerate is distinguished from a fully developed flow, with an equation for the velocity of the trailing slug:

$$(2.11) \quad \bar{u}_s = 1.2 \frac{Q_G + Q_L}{A} + C \sqrt{gD}$$

where C is a function dependent on the separation distance h_s between the bubbles and the tube diameter D , Figure 2.11. For a fully developed flow the function C equals 0.35, but for a developing slug-flow, C is given by:

$$C = 0.35 + 2.8e^{-1.06 \frac{h_s}{D}}$$

Moissis experimented using tubes with an internal diameter of 0.5in, 0.75in, 1in and 2in, but the viscosity and surface tension effects were more pronounced in small tubes than in larger.

2.3.1 Film thickness calculations

Between the rising slug and wall of the tube there is a film of descending liquid of thickness δ .

Moissis [M7] has expressed the absolute liquid film velocity as:

$$(2.12) \quad V_f = \frac{1}{4\delta(D-\delta)} \left[\left(12 \frac{Q_G + Q_L}{A} + C\sqrt{gD} \right) (D-2\delta)^2 - D^2 \frac{Q_G + Q_L}{A} \right]$$

Several relations have been proposed for the film thickness. Hubbard proposed that the film thickness occupies about 20% of the tube cross-sectional area but this is only a very rough approximation.

In the modelling of the slug-flow, Moissis [M7] proposed two methods to calculate δ .

1. The potential flow assumption is valid, only if the flow of liquid with respect to the slug is a potential flow. In this case the slug nose becomes a stagnation point and the film-slug interface becomes a constant pressure surface. For this to occur, the slug-flow must be fully developed and the length of the bubble must be shorter than the length at which the film velocity reaches a terminal value. In his model neither of the above could be satisfied.
2. The governing equation for the film thickness δ includes shear forces. The shear stress model at the slug-liquid interface (neglecting momentum changes and pressure forces) is consistent with the Nusselt film condensation model.

Thus the film thickness is given by:

$$(2.13) \quad \frac{\delta}{D} = f \cdot \frac{\frac{V_f}{\sqrt{gD}}}{2 \left(1 - \frac{\delta}{D} \right)}$$

where f = a friction factor dependent on the film liquid Reynolds number Re , given by:

$$(2.14) \quad Re = \frac{4(\sqrt{gD}) \cdot D \cdot \delta^* \rho_f \cdot V_f^* (1 - \delta^*)}{\mu_f}$$

where $\delta^* = \frac{\delta}{D}$ $V_f^* = \frac{V_f}{\sqrt{gD}}$

Rosehart and Jagota [R7] proposed a theoretical model predicting the film mass transfer coefficient in annular-flows. In this model, Rosehart and Jacota assumed that the average annular film thickness, (not including the droplets) can be represented by eq. (2.15)

$$(2.15) \quad \delta = \frac{D}{2} \left(1 - \sqrt{1 - \frac{u_G}{V_f}} \right)$$

2.3.2 Pressure drop calculations

There are a variety of methods to calculate the pressure drop for two-phase flows, depending on the specified conditions of the system, geometry, phase properties, etc. Most of the pressure drop correlations stem from the Martinelli et al. [M3] correlation revised by Lockhart-Martinelli [L3] which was based upon the following assumptions:

- The static pressure drop for the gas-phase equals that for the liquid-phase, irrespective of the flow pattern.
- The sum of the volumes of the gas and the liquid equals the volume of the tube.
- The momentum and hydrostatic pressure-drops were neglected. However, Martinelli and Nelson's approach [M4] includes the momentum and hydrostatic pressure losses.
- Lockhart-Martinelli defined four possible types of flow mechanisms which can exist in a simultaneous flow of gas and liquid. These are the four combinations between the gas and the liquid in turbulent and viscous flows, that is [M5]:

t-t = turbulent liquid-flow, turbulent gas-flow

t-v = turbulent liquid-flow, viscous gas-flow

v-t = viscous liquid-flow, turbulent gas-flow

v-v = viscous liquid-flow, viscous gas-flow

Lockhart-Martinelli [L3] expressed the pressure drop losses in two-phase flows with a relationship correlating the pressure drop losses in each phase.

$$\left(\frac{\Delta P}{\Delta L}\right)_{TP} = \left(\frac{\Delta P}{\Delta L}\right)_L \Phi_L^2 = \left(\frac{\Delta P}{\Delta L}\right)_G \Phi_G^2$$

where

$$\left(\frac{\Delta P}{\Delta L}\right)_L, \left(\frac{\Delta P}{\Delta L}\right)_G \text{ are the pressure drops per unit length, when each phase flows alone.}$$

The multipliers Φ_L and Φ_G are related to the Lockhart-Martinelli dimensionless function by the following relation:

$$(2.16) \quad X^2 = \frac{\Phi_L^2}{\Phi_G^2} = \frac{\left(\frac{\Delta P}{\Delta L}\right)_L}{\left(\frac{\Delta P}{\Delta L}\right)_G}$$

The above relationship is valid if, and only if the quantities $\left(\frac{\Delta P}{\Delta L}\right)_L, \left(\frac{\Delta P}{\Delta L}\right)_G$ are calculated from the basis that the liquid or gas is flowing at the same individual mass-flow rate as in the two-phase flow. This relationship is shown graphically in Figure 2.12.

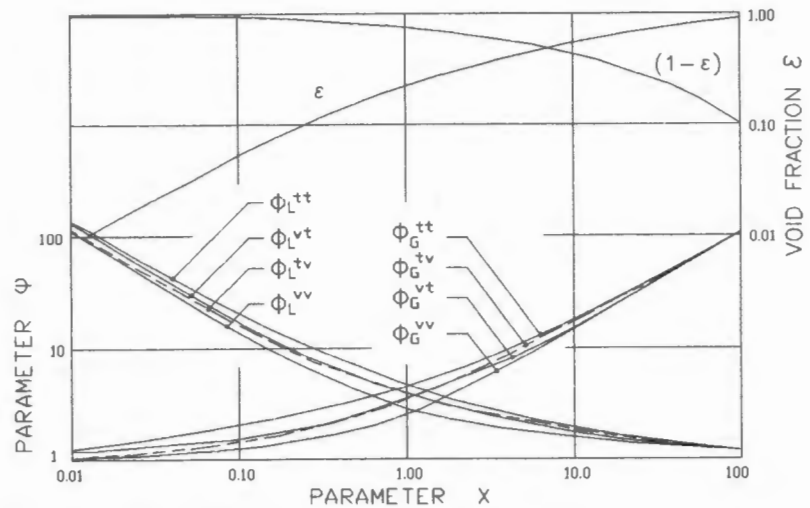


Figure 2.12 The relationship of the dimensionless parameter X with the void fraction ϵ and the parameter Φ for all flow mechanisms

The dimensionless variable X

is associated with the four types of flows and is a function of weight rates W , densities ρ , viscosities μ , and tube diameter D . These variables are expressed as follows [M3]:

$$(X_{t-t})^{1.11} = \left(\frac{W_L}{W_G} \right) \left(\frac{\rho_G}{\rho_L} \right)^{0.555} \left(\frac{\mu_L}{\mu_G} \right)^{0.111}$$

$$(X_{v-t})^2 = \left(\frac{W_L}{W_G} \right) \left(\frac{\rho_G}{\rho_L} \right) \left(\frac{\mu_L}{\mu_G} \right) \left(\frac{C_L}{C_G} \right) \text{Re}_G^{-0.8}$$

$$(X_{t-v})^2 = \left(\frac{W_L}{W_G} \right) \left(\frac{\rho_G}{\rho_L} \right) \left(\frac{\mu_L}{\mu_G} \right) \left(\frac{C_L}{C_G} \right) \text{Re}_G^{0.8}$$

$$(X_{v-v})^2 = \left(\frac{W_L}{W_G} \right) \left(\frac{\rho_G}{\rho_L} \right) \left(\frac{\mu_L}{\mu_G} \right)$$

where

C_G and C_L are constants for the gas and liquid friction factors respectively, with values

$C_G=0.184$ for turbulent-flow in smooth pipes

$C_L=64$ for viscous-flow.

According to the Fanning equation it can be shown that the static pressure drop due to the liquid-flow may be written as:

$$\left(\frac{\Delta P}{\Delta L} \right)_{TP} = 2f_L \cdot \frac{\rho_L \cdot V_L^2}{D_L \cdot g}$$

$$\left(\frac{\Delta P}{\Delta L} \right)_{TP} = 2 \cdot f_G \cdot \frac{\rho_G \cdot V_G^2}{D_G \cdot g}$$

where

D_L and D_G are the hydraulic diameters for the liquid and gas-flows respectively.

f_L and f_G are the friction factors which can be expressed in the Blasius form as:

$$f_L = \frac{C_L}{\text{Re}_L^n} = \frac{C_L}{\left(\frac{4W_L}{\pi \cdot \alpha \cdot \mu_L \cdot D_L} \right)^n}$$

$$f_G = \frac{C_G}{\text{Re}_G^m} = \frac{C_G}{\left(\frac{4W_G}{\pi \cdot \beta \cdot \mu_G \cdot D_G} \right)^m}$$

α , β are the ratios of the actual cross section area of the respective flow to the area of a circle with diameters D_L and D_G respectively, that is:

$$\alpha = \frac{A}{\left(\frac{\pi}{4}\right)D_L^2} \quad \beta = \frac{A}{\left(\frac{\pi}{4}\right)D_G^2}$$

The relation between the two unknown quantities D_L and D_G is:

$$\frac{D_L^{5-n}}{D_G^{5-m}} = \left(\frac{C_L}{C_G}\right) \left(\frac{W_L^{2-n}}{W_G^{2-m}}\right) \left(\frac{\mu_L^n}{\mu_G^m}\right) \left(\frac{\rho_G}{\rho_L}\right) \left(\frac{\beta^{2-m}}{\alpha^{2-n}}\right) \left(\frac{\pi}{4}\right)^{n-m}$$

The values of the exponents “n” and “m”, the constants “ C_L ” and “ C_G ” and the Reynolds numbers “ Re_L ”

and Re_G are given in Table 2.1 according to the type of flow.

TABLE 2.1				
	t-t	v-t	t-v	v-v
n	0.2	1.0	0.2	1.0
m	0.2	0.2	1.0	1.0
C_L	0.046*	16.0	0.046*	16.0
C_G	0.046*	0.046*	16.0	16.0
Re_L	>2000	<1000	>2000	<1000
Re_G	>2000	>2000	<1000	<1000

* = For smooth pipes

Davis [D2] modified the Lockhart-Martinelli parameter X for turbulent-flows and introduces the Froude number by which his model can describe gravitational and inertia effects.

$$(2.17) \quad X = 0.19 \left(\frac{W_L}{W_G}\right)^{0.9} \left(\frac{\rho_G}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_G}\right)^{0.1} \left(\frac{V_M^2}{gD}\right)^{0.185}$$

This relation, has a 20% agreement with data for Reynolds numbers above 8000, or above 6000 if the Froude number exceeds 100 [S5].

2.4 INTERFACIAL AREA IN CO-CURRENT GAS-LIQUID FLOW

Successful attempts to measure the interfacial area of two-phase flows have been reported by a number of researchers. The aim of their investigation was to produce correlations which could predict the mass transfer coefficients.

Banerjee et al. [B2] measured interfacial areas between gas and liquid in helical coils of 16.5mm internal diameter (ID) for annular-flow. Measurements also have been made by Gregory and Scott [G3] for slug-flow in a 19mm ID horizontal tube 8077mm long, by Kulic and Rhodes [K13] for slug-flow in helical tubes with 16.5mm ID and by Kasturi and Stepanek [K1] who experimented on flows in 6mm vertical tubes. Their work was further investigated by Shilimkan and Stepanek to include 10mm, 15mm and 20mm ID tubes for mainly annular-flows. Tomida et al. [T3, T4] report on slug, froth and annular-flows in vertical tubes of 10mm, 18mm and 24mm ID. These authors used chemical absorption of carbon dioxide into sodium hydroxide solutions. Their work is presented in a comparative form in APPENDIX K.

2.5 MASS TRANSFER IN TWO-PHASE GAS-LIQUID-FLOW

Two-phase gas/liquid-flow has been the study of many researchers since the 1950s. During the 1960s heat transfer investigations increased, because of the great importance of the two-phase flow occurring in the coolant channels in reactor cores. It was only during the 1970s when mass transfer studies became more popular in terms of molecular diffusivity. In respect to this, most of the studies are based on existing theoretical models which describe the mass transfer at a fluid-fluid interface. For example, the film theory proposed by Whitman in 1923, the two-film theory developed by Lewis and Whitman in 1924, the penetration theory proposed by Higbie in 1935, the surface renewal theory proposed by Danckwerts in 1951, and the film penetration proposed by Toor and Marchello in 1958 [H4]. A simple model of turbulent heat and mass transfer based on a modified form of the Reynolds analogy, is proposed by Kropholler and Carr [K11]. Equations have been derived from which heat and mass transfer coefficients, and temperature or concentration profiles may be predicted at any value of Prandtl or Schmidt number, APPENDIX L. It has been assumed that there is no significant molecular transport and that eddy transport is a function of the flow pattern only [I1, K11].

2.6 OPERATIONAL ARGUMENTS BETWEEN FILM AND VERTICAL TUBULAR BUBBLE ABSORBERS.

The film absorbers are the most commonly applied contactors in the chemical industry. The vertical tubular bubble absorbers are also gaining interest lately due to the increased turbulence in the co-current two-phase flow.

In the previous sections it was established that the requirements which have to be satisfied in an absorber are those of good heat and mass transfer. However, these requirements could render the absorber expensive. It was customary to calculate the area required for heat transfer and assume the same will be sufficient for mass transfer as well. It was only the work by Haselden and Malaty [H3] that considered a simultaneous heat and mass transfer.

Film absorbers are prone to hydrodynamic instability triggered by a slight deviation from the vertical [B5] or by scale deposits on the wall. Hydrodynamic instability is also enhanced by the process of heat and mass transfer [B8, H3]. Due to this film instability the area available for heat and mass transfer is greatly reduced. With respect to this, methods of improving hydrodynamic stability have been proposed but no literature is known to the author for a concrete solution.

Nevertheless, improvements on the transfer rates can be expected if:

- the liquid distribution is improved aiming to increased film stability.
- turbulent-flow exists in the film. This can be caused by an appropriate wall geometry over which the film is flowing, although rough or uneven surfaces can also increase film instability.

In both cases, the heat and mass transfer coefficients are not expected to improve.

Moreover, film absorbers for absorption refrigeration machines lack the ability to fulfil certain requirements due to the film instability.

- They are unable to survive movement. This limits the absorption machines to be land based and immobile.

- Due to **Marangoni-effect** [B8] film absorbers cannot handle a concentration difference more than 8% between the strong and weak solutions. This argument is valid only if the temperature of the heat supply in the generator is adequate to produce a large concentration difference between the two solutions. This condition is desired as it would result in reduced circulation rates and reduced sizes of the components involved [H6, V3].
- It has been the trend for the film absorbers to have the film flowing on the outer surfaces of tubes. This prohibits the absorbers from being air-cooled which requires surface extensions, such as fins etc.

Tubular bubble absorbers do not exhibit problems with the above conditions, and in addition:

- they are simpler and more compact in construction
- they allow very high flow rates which eliminate the possibility of flooding.
- they facilitate temperature control.

For these reasons the following paragraphs will focus onto the design of tubular bubble absorbers and in particular on the prediction of the height of the tubes as a parameter of the flow conditions.

2.7 DESIGN CALCULATIONS OF THE TUBULAR BUBBLE ABSORBER

The study on vertical bubble absorbers is focused on the simultaneous heat and mass transfer calculations. For this it is necessary to obtain values for the heat and mass transfer coefficients and also their dependence on parameters which enable the design of the bubble absorber.

From the literature cited, studies on the mass transfer were concerned with annular-flow, and few with slug-flow in horizontal tubes. The only literature known to the author which refers to vertical slug-flow with absorption taking place, is that by Infante Ferreira et al. [I2]. In their study, only the co-current upward flow was considered, although their apparatus was capable of offering co-current upward and downward, and counter-current flows. The limitation was

due to the practicality of the experimentation. That is, the conditions with respect to the mass ratio of the gas to the liquid which were required to obtain co-current downward and counter-current flows, were different to the conditions prevailing in absorption refrigeration machines.

In addition, they only considered a liquid-phase controlled mass transfer, ignoring the gas-side mass transfer coefficient due to the use of a pure gas. From literature it is not clear whether the mass transfer is governed by the gas-phase, or the liquid-phase, or both phases. Hughmark [H10] remarked that during the absorption of air-ammonia mixtures in water-ammonia solutions the liquid-phase resistance cannot be neglected. The absorption of pure ammonia gas in water-ammonia solutions is controlled by both the gas and liquid-phases.

The experiment by Infante Ferreira et al. [I2] involved absorption of pure ammonia gas into ammonia-water solution and it aimed to establish the relationship of the height of the absorber tubes with various operational variables. A preliminary study was performed at adiabatic conditions in order to determine the slug rise velocity at different absorber heights. The set-up contained a tube of 1m height and 16.5mm ID made of glass, to allow observation of the absorption process. These observations revealed two distinguished flow patterns. At the entrance region there was froth-flow, which then changed to a developed slug-flow.

Measurements were taken for the rise velocity of the slugs, the pressure drop and the void fraction. From these, the film thickness and the overall mass transfer coefficient were calculated.

According to Nicklin [N1, N2] the rise velocity of the slug is given by eq.(2.7). Similar relations were derived for different heights z in the absorber.

According to [I2]:

$$(2.18) \quad u_s = 1.2(u_{GS} + u_{LS}) + 0.36\sqrt{gD} \quad \text{for } z=0.2\text{m}$$

$$(2.19) \quad u_s = 1.4(u_{GS} + u_{LS}) + 0.29\sqrt{gD} \quad \text{for } z=0.5\text{m}$$

The coefficients of eq. (2.18) and (2.19) are in agreement with those proposed by Nicklin [N1], that is $C_1=1.2$ and $C_2=0.35$, eq. (2.10).

The deviation on the C_1 and C_2 coefficients is mainly due to Nicklin's relation being valid only for flows without absorption. When average values were considered over the entire height of the absorber, the deviations were much stronger with $C_1=-0.2$ and $C_2=0.89$

This was due to the flow pattern being unstable and not fully developed slug-flow within the first 0.2m of the absorber's height.

Prediction of the pressure drop in the system using the relations of Nicklin [N1, N2] and of Hughmark and Pressburg [H11] showed the closest agreement, with deviations resulting due to calculated values being referenced to the inlet conditions. Large deviations were shown when the pressure drop was calculated by the relations proposed by Lockhart and Martinelli [L3] and Davis [S5]. This is because both relations do not take into account the hydrostatic pressure drop and the slip into account.

The mean values of the pressure drop are shown in Table 2.3.

Table 2.3		
<i>Comparison of experimental pressure drop per unit height of absorber with some of the correlating methods</i>		
		Mean Pressure drop
		$\Delta P/\Delta z$ [kPa/m]
Infante Ferreira (experimental)	[I2]	5.075
Nicklin	[N1]	2.9875
Hughmark and Pressburg	[H11]	2.4375
Davis	[S5]	0.325
Lockhart and Martinelli	[L3]	0.3375

The void fraction was also calculated according to methods described by [N1], [H11], [S5] and [L3]. The mean void fraction values are shown in Table 2.4

Table 2.4		
<i>Comparison of experimental void fraction with some of the correlating methods</i>		
		Mean void fraction
Infante Ferreira (experimental)	[I2]	0.423
Nicklin	[N1]	0.59
Hughmark and Pressburg	[H11]	0.99
Davis	[S5]	1.15
Lockhart and Martinelli	[L3]	0.66

According to Jagota et al. [J1] the overall mass transfer coefficient K_L is expressed by:

$$K_L = \frac{m_L}{\rho_L \cdot A_s} \cdot \frac{\frac{c_o - c_i}{(c_i^* - c_i) - (c_o^* - c_o)}}{\ln \frac{(c_i^* - c_i)}{(c_o^* - c_o)}}$$

where A_s is the surface area of the slug which is also the area for mass transfer

$[c] = \text{kmol/m}^3$ concentration of solute in liquid

$[c^*] = \text{kmol/m}^3$ equilibrium concentration of solute in liquid

This expression gave an overall mass transfer coefficient value $K_L \cdot 2 \times 10^{-4} \text{ms}^{-1}$

When predicting the individual mass transfer coefficient k_L with relations similar to eq. (L2) in APPENDIX L, the values obtained were between 10^{-6} to 10^{-2} m/s. This reveals that on certain occasions $k_L < K_L$, a fact which contradicts eq. (2.4). In case $k_L = K_L$ the mass transfer would be governed by the liquid-side resistance. In case $k_L > K_L$ the mass transfer would be governed by the gas-side resistance. These contradictory results which were caused mainly by severe temperature variations at the inlet of the absorber, led Infante Ferreira et al. [I2] to analyse the behaviour of the system involving heat and mass transfer simultaneously.

In the proposed model by Infante Ferreira et al. [I2] the independent variables necessary for the design of the absorber of an absorption machine are:

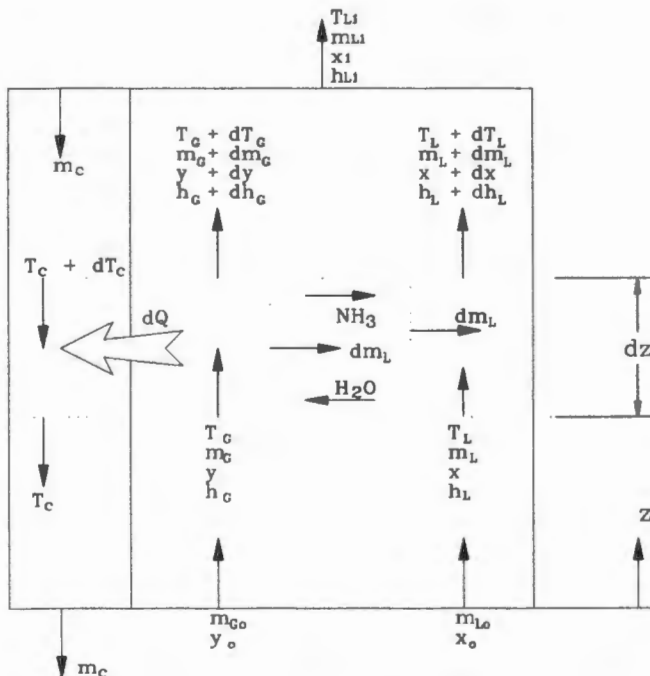
- the total pressure in the absorber P
- the concentrations of the weak or strong solutions, x^{WE} or x^{ST} and of the refrigerant vapour y_{ref}
- the mass flow rates of the refrigerant vapour m_G and the weak solution and m_L
- the height and the diameter of the tubes z and D respectively
- the temperatures of the solution T_L and the gas T_G in the tubes.
- the heat transferred to the coolant Q .

All these parameters can be grouped in an equation which could solve the height of the absorber tubes:

$$z = f(x^a \cdot y_{mf}^b \cdot Q^c \cdot T_L^d \cdot T_G^e \cdot m_G^f \cdot m_L^g \cdot P^h \cdot D^i)$$

The parameter x is the concentration of the bulk solution in the absorber.

It was assumed that heat and mass transfer coefficients were constant along the absorber. The model is based upon a differential element dz at any arbitrary height z , Figure 2.13, for the following flow conditions:



- co-current upward flow for the ammonia gas and ammonia-water solution
- cooling medium flows in the counter-current direction
- the ammonia concentrations are uniform in both phases and in equilibrium after each differential element
- the temperatures in both phases are uniform and equal to each other

Figure 2.13 The differential element for the formulation of heat and mass balances [12]

At an element dz , the entering and leaving variables at constant total pressure, are as follows:

	<i>ENTERING VARIABLES</i>	<i>LEAVING VARIABLES</i>
mass flow of liquid	m_L	$m_L + dm_L$
mass flow of gas	m_G	$m_G + dm_G$
mass flow of coolant	m_C	m_C
temperature of liquid	T_L	$T_L + dT_L$
temperature of gas	T_G	$T_G + dT_G$
concentration of NH_3 in liquid	x	$x + dx$
concentration of NH_3 in gas	y	$y + dy$

The boundary conditions at the inlet and outlet of the absorber are indicated with the indices "0" and "1" respectively. Also at the top of the absorber all gas should be absorbed and therefore $m_{G1}=0$.

From their model description, it is deduced that the absorbed mass of the gas is:

$$dm_G = -K_L \alpha (x^* - x) \frac{\pi}{4} D^2 dz$$

where

$x^* - x$ = the concentration potential

x^* = the concentration in the liquid which is in equilibrium with the gas concentration

The height of the differential element is also deduced by re-arrangement of the above equation.

Thus:

$$(2.20) \quad dz = - \frac{dm_G}{K_L \alpha (x^* - x) \frac{\pi}{4} D^2}$$

The values of the variables for the gas, liquid and the coolant leaving the element can be taken as the values of the variables entering the next element. If "n" is the necessary number of elements needed to absorb all the gas (that is $dm_G=0$) then the height of the tube is $z = n \cdot dz$.

2.7.1 Heat transfer calculations

The rate of heat transferred in the absorber dQ , is evaluated as follows:

$$dQ = U \cdot A_i \cdot (T_L - T_C)$$

where

T_C is the temperature of the coolant

$$A_i = \pi \cdot D_i \cdot dz$$

U = the overall heat transfer coefficient:

$$(2.21) \quad \frac{1}{U} = \frac{1}{\alpha_i} + \frac{D_i}{2k_t} \ln\left(\frac{D_o}{D_i}\right) + \frac{D_i}{D_o \cdot \alpha_c}$$

α_i and α_c are the heat transfer coefficients at the solution-side and at the coolant-side of the absorber tube respectively and k_t is the heat conductivity of the tube wall.

The temperature of the coolant leaving the element can be calculated from:

$$(T_C - dT_C) = T_C - \frac{dQ}{m_c \cdot Cp_c}$$

Estimation of the heat transfer coefficients α_i and α_c

The heat transferred from the solution to the coolant is the difference between the heats involved in the gas and liquid before mixing and in the homogeneous solution at the exit after mixing.

$$Q = m_{Go} \cdot h_{Go} + m_{Lo} \cdot h_{Lo} - m_{L1} \cdot h_{L1}$$

With the temperatures of the solution and of the coolant at the entrance and exit of the tube being known, the heat transfer coefficient U is related to Q by:

$$Q = U \cdot A_i \cdot \Delta T_{Ln}$$

where

ΔT_{Ln} = the logarithmic mean temperature difference

From Welty, Wicks and Wilson [W4] the coefficient α_c is evaluated from:

$$(2.22) \quad \alpha_c = 0.86 \left(\frac{k_w}{D_H} \right) \cdot \left(\frac{D_2}{D_o} \right)^{0.16} \frac{\xi_w \cdot Pr_w (Re_H - 1000)}{1 + 12.7 \left[(Pr_w)^{\frac{2}{3}} - 1 \right] \cdot \xi_w^{0.5}} \quad \text{for } Re_F > 2300$$

$$\alpha_c = 0.86 \left(\frac{D_2}{D_o} \right)^{0.16} \left(\frac{k_w}{D_H} \right) \cdot \left[3.66^3 + 1.61^3 Re_H Pr_w \cdot \left(\frac{D_H}{z} \right) \right]^{\frac{1}{3}} \quad \text{for } Re_F < 2300$$

The friction factor for the water-side is defined as:

$$(2.23) \quad \xi_w = \frac{(1.82 \log_{10}(Re_H) - 1.64)^{-2}}{8}$$

According to Horn [H9] the two-phase flow coefficient α_i can be expressed as the sum of the coefficients for the two regions which prevail in the absorber tube. That is, for that part of the absorber where the slug and froth-flow dominate, α_i is identical to the heat transfer coefficient in a thin liquid film. This portion of the flow corresponds with the average void fraction "ε". For the remaining part of the flow, (liquid slugs), α_i corresponds with the heat transfer coefficient for a single-phase liquid flow. Thus:

$$(2.24) \quad \alpha_i = \varepsilon \cdot \alpha_{LF} + (1 - \varepsilon) \cdot \alpha_{LP}$$

where α_{LF} and α_{LP} are the film and the single-phase (liquid plug) heat transfer coefficients respectively.

The characteristic length in Horn's relations is the film thickness calculated by the equation proposed by Brauer [B11]:

$$(2.25) \quad \delta = \left(\frac{3 \mu D_i u_G}{4 \rho g} \right)^{\frac{1}{3}} \quad \text{for } Re_{LF} < 200$$

$$\delta = (Re_{LF})^{\frac{1}{3}} \left(\frac{3 \cdot \left(\frac{\mu}{\rho} \right)^2}{g} \right)^{\frac{1}{3}} \quad \text{for } 200 < Re_{LF} < 400$$

$$\delta = 0.302 \left(Re_{LF} \right)^{\frac{8}{15}} \left(\frac{3 \left(\frac{\mu}{\rho} \right)^2}{g} \right)^{\frac{1}{3}} \quad \text{for } Re_{LF} > 400$$

The film velocity V_f is given by Brauer's expression:

$$(2.26) \quad V_f = \frac{\rho_L \cdot \delta^2 \cdot g}{3\mu_L}$$

2.7.2 The overall volumetric mass transfer coefficient $K_L\alpha$

The overall volumetric mass transfer coefficient $K_L\alpha$ was defined by eq.(2.20).

The parameters involved in the determination of the conditions of the solution vary with the height z of the absorber, such as the concentrations x and x^* , as well as the temperature T_L and the pressure P . The coefficient $K_L\alpha$ will also vary from very large values at the entrance of the absorber, to very small values at the exit. Therefore a simple integration of eq. (2.20) will give erroneous results in the prediction of the gas mass flow rate m_G through the absorber.

The volumetric overall mass transfer coefficient can be calculated from Higbie's theory applied to laminar falling film. Thus if the local gas-phase mass flow rate m_{GL} is known:

$$K_L\alpha = 0.283 \left(\frac{m_{GL}}{\rho_G} \right)^{0.33} \cdot \left(\frac{\rho_L^{1.17}}{D_L^{0.5} \nu_L^{0.17}} \right) \cdot \frac{1}{z_f^{0.5} D^{0.83}}$$

Alternatively, Infante Ferreira's et al. [I2] approach was to determine $K_L\alpha$, by using dimensional analysis, based on the Chilton's and Colburn's [C3] relation on the Reynolds analogy between mass and momentum transfer.

Their investigation showed that $K_L\alpha$ can be related to the superficial gas velocity:

$$(2.27) \quad K_L\alpha = 22.44 \frac{\rho_G \cdot u_{Gs}}{\mu_G} \cdot \rho_L^{0.6} \cdot D_L^{0.6} \cdot \mu_L^{0.4} \cdot \frac{D^{0.1}}{z}$$

This indicates that the gas velocity and the height of the tube are the strongest influences on the mass transfer, while the diameter of the tube is hardly of any importance. However, to achieve high gas velocities the tube diameter must be small, but to achieve high heat transfer rates the tube diameter must be large.

Therefore, if mass transfer resistance is controlling the absorption process, the tube diameter must be small and if heat transfer resistance is controlling the process, the tube diameters must be large.

In addition, according to Keizer [K5] it was concluded that for a tube diameter of 20 mm:

- if the heat transfer coefficient $U > 1000 \text{ W/m}^2\text{K}$ then mass transfer controls the absorption process
- if the heat transfer coefficient $U < 100 \text{ W/m}^2\text{K}$ then heat transfer controls the absorption process
- if the heat transfer coefficient $\sim 170 < U < \sim 950 \text{ W/m}^2\text{K}$ then both heat and mass transfers control the absorption process.

2.8 THE INFLUENCE OF VARIOUS PARAMETERS ON THE PREDICTION OF HEIGHT OF THE ABSORBER TUBES

Further results from a multiple regression analysis by [I2] relate the height of the absorber tubes with the most significant variables:

The conclusions which were drawn were:

- smaller absorbers will be required if there is an increase of:
 - the system pressure
 - the tube diameters in the absorber
 - the liquid mass flow rate

- larger absorbers will be required if there is an increase of:
 - the liquid-phase concentration
 - the liquid-phase temperature
 - the gas-phase mass flow rate
 - the transferred heat

A multiple regression, with emphasis on the concentration difference between the strong and the weak solutions, revealed that the number of tubes was also a significant variable which could influence the height of the absorber.

It was proposed by Keizer [K5] that the height of the tube can be expressed in terms of the number of tubes and the tube diameter and the concentration difference between the weak and strong solutions.

$$(2.28) \quad z = \frac{1040}{N \cdot D_i} \left(1 - \Delta x^{\frac{1}{3}} \right) \quad [z] \text{ and } [D_i] = \text{cm} \quad [x] = \text{mass\% NH}_3$$

From the above equation the value of the product ZND_i remains constant for a constant Δx , which means that for a fixed diameter, the effect would be the same for a small number of long tubes, as for a larger number of shorter tubes. From this observation and the closing statement of section 2.7.2, it implies that the absorption process in vertical tubular absorbers is controlled by heat transfer. Therefore, since the surface available for heat transfer is linear to the tube diameter, the tube height z is strongly dependent on the tube diameter D_i .

It can also be observed that the height z and the quantity $\left(1 - \Delta x^{\frac{1}{3}} \right)$ are linearly related

This indicates that the ratio $z/\Delta x$ decreases as the concentration difference increases. Therefore if higher Δx values can be obtained, the size of the absorber will diminish. Furthermore, as the mass flow rates of the strong and the weak solutions depend upon the concentration difference, higher values of x would result in a smaller size absorption unit.

The dependence of the tube height on the heat transfer is indicated in two ways:

- the dependency on the coolant-side heat transfer coefficient α_c :

$$(2.29) \quad z = 4645 \alpha_c^{-0.591}$$

This is a fictitious relationship, because it was formulated for a simulated constant temperature T_c along the absorber. However, it gives an indication on the behaviour of the absorber's height in conditions of low and high heat transfers. It can be observed that for values of $\alpha_c > 2000 \text{ W/m}^2\text{K}$ a change in α_c produces an insignificant change in the absorber's height and on the overall heat transfer coefficient U . For values of $\alpha_c < 2000 \text{ W/m}^2\text{K}$, changes in α_c greatly affect the tube-height and the value of U in eq.(2.21). This is because it has a direct influence on the temperature difference T_c of the coolant. This in turn is directly linked with the solution temperature and hence with the concentration and the absorber's efficiency.

- the dependency on the coolant temperature difference ΔT_c

$$(2.30) \quad z = 270 \Delta T_c^{-0.72}$$

This expression, although it cannot be used to solve for the temperature difference it shows a moderate dependency on ΔT_c , which if decreased would lead to an increase in the absorber's height.

The above relations, eq. (2.29) and (2.30) can be applied only as a rough estimate of the absorber's size. For a more accurate prediction, the tube height "z" is correlated with the parameters which have a significant effect on both the heat and mass transfer.

$$(2.31) \quad z = 133.6(x)^{2.1} \cdot \left(\frac{m_G^{0.7}}{m_L^{0.04}} \right) \cdot \left(\frac{T_L^{0.4} \cdot h_{GL}^{0.6}}{P \cdot D^{0.5}} \right)$$

However, knowledge of the quantity h_{GL} (heat of solution plus vaporisation), is required.

CHAPTER 3

ANALYSIS OF THE SIMULATION MODEL

PRESENT WORK

3.0 GENERAL

The method of analysis presented in this chapter makes use of the ammonia-water enthalpy-concentration diagram. The enthalpy-concentration diagrams stem from the original development of the treatment of binary mixtures by Mollier and Merkel, and are unique for every binary absorbent-refrigerant mixture.

The following discussion on the graphical presentation of the absorption refrigeration process is based on the method developed by Bosnjakovic [B10]. He was an associate of Mollier, who researched the thermodynamic diagrams of binary mixtures and their application to a variety of technological applications. This method can be found in a number of publications [C2, T2, W2], but up until now it has not been utilised to its full potential, as for example in designing and optimising an absorption refrigeration machine.

In the following sections, the cycle is analysed graphically and it will be shown that the cumbersome and complex mathematical analysis presented in textbooks and journals [R2, W3] can be substituted by a much easier to follow diagrammatic approach.

Literature search established that there were not available computerised equations estimating the enthalpies of the aqua-ammonia mixture, that could compare accurately with the well-established enthalpy-concentration diagram.

Under the assumption that Bosnjakovic's chart of aqua-ammonia [K10] represents the true values of the mixture, his chart was traced and the resulting curves were fitted with Fourier equations (Figure C1, APPENDIX C) [V2]. These equations are bounded between pure water and pure ammonia concentrations and form an integral part of the computerised model.

3.1 PERFORMANCE OF THE ABSORPTION REFRIGERATION MACHINES

The absorption refrigeration machine could be treated theoretically as a combination of an engine producing work and as a refrigerator absorbing the same amount of work. The theoretical cycle analysis for an absorption machine, shows that the coefficient of performance (COP) values obtained are much higher than the COP values obtained in practice.

With reference to the absorption refrigeration machine shown in Figure 3.1 (neglecting the work done by the solution pump W_p) the overall heat balance can be written as:

$$Q_G + Q_E \geq Q_C + Q_A + Q_D$$

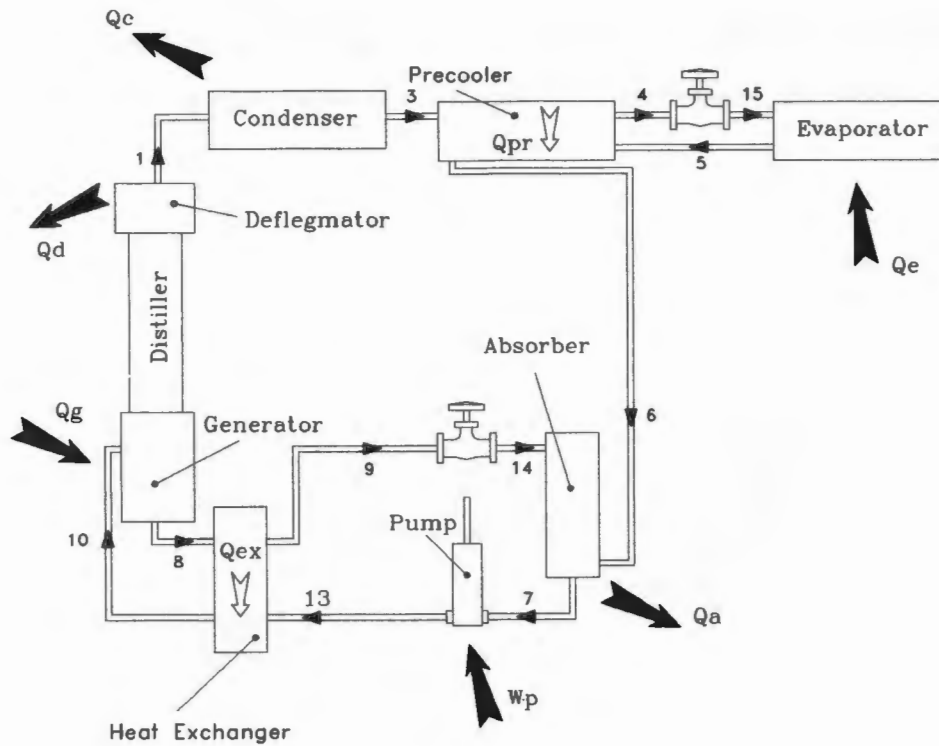


Figure 3. 1 Component arrangement of the single stage two-fluid absorption system

A corollary of the second law states that the COP of an ideal Carnot cycle operating between two heat reservoirs, depends only upon the temperatures of the heat reservoirs.

Assuming that the cooling temperatures for the distiller, condenser and absorber are equal to a sink temperature T_s , that is $T_D = T_C = T_A = T_s$, and also that the heat quantity Q_D can be incorporated in the heat rejected by the condenser, then for a generator's temperature T_G :

$$G_G \left(\frac{T_G - T_s}{T_G} \right) \leq Q_E \left(\frac{T_s - T_E}{T_E} \right)$$

For any absorption refrigeration system, the COP is defined as the ratio of the refrigeration capacity to the amount of heat supplied to the generator.

Therefore the COP of the reversible cycle is given by:

$$COP = \frac{Q_E}{Q_G} \leq \frac{T_E}{T_G} \cdot \frac{T_G - T_S}{T_S - T_E}$$

If values are assigned to the generator, evaporator and sink temperatures T_G , T_E and T_S respectively, then the coefficient of performance (COP) of the reversible cycle can be represented in a graph similar to Figure 3.2. For example, if the generator is at 110°C and the evaporator and sink temperatures are at -20°C and 15°C respectively, then the theoretical COP is 1.79.

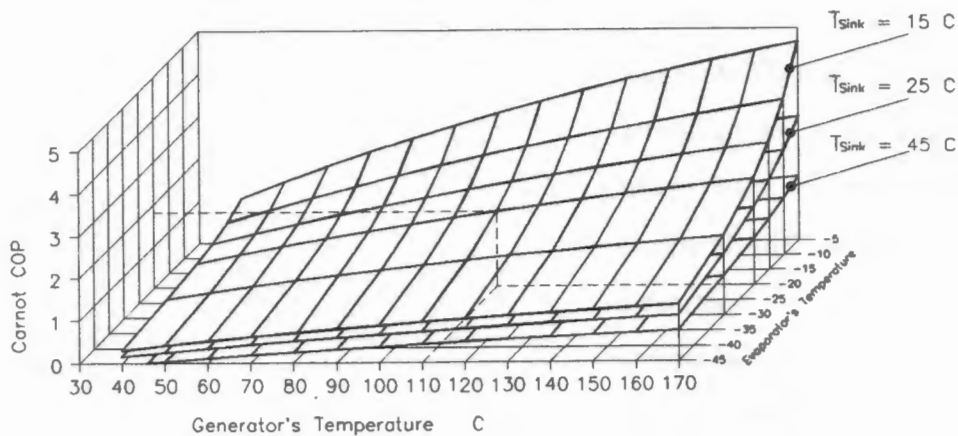
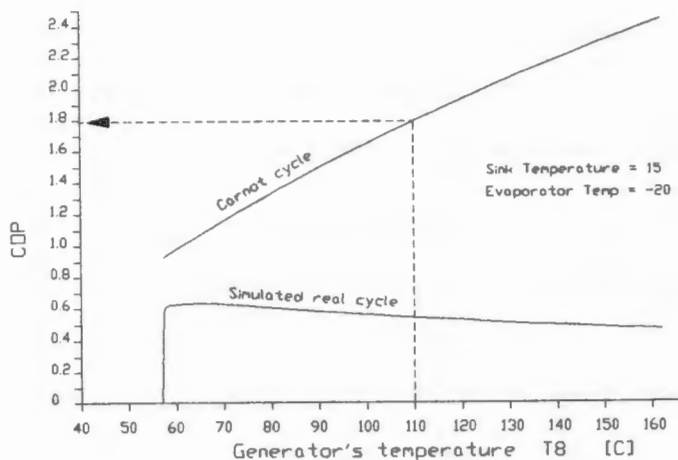


Figure 3. 2 The effect of the temperature conditions on the COP of the reversible absorption refrigeration cycle.



The simulation which predicts the performance of a real machine (discussed later in this chapter) for various temperature conditions, produces curves as shown in Figures 3.3 and 3.4. The COP of a real cycle drops off after it has reached a maximum value.

Figure 3. 3 Comparison of the COPs between a real and a reversible cycle. Real cycle's data were obtained from author's computerised simulation model.

Figure 3.3 shows an increasing gap between the reversible and real cycle COPs.

As the temperature of the heat source increases, more of the volatile absorbent is released together with the refrigerant. For this mixture to be purified to the required concentration, more heat is rejected from the deflegmator. Concurrently the heat of reaction involved in both the generation and the absorption processes are highly irreversible. Therefore adopting the reversible cycle procedure to estimate the COP of the real absorption cycle is a totally impractical way.

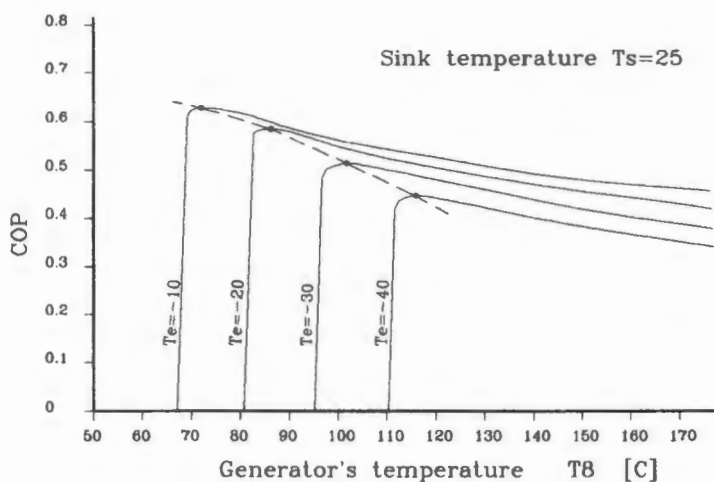


Figure 3.4 Typical COP curves, produced by the simulation model, for a variety of evaporator and generator conditions

For each sink temperature T_s it is possible to construct diagrams similar to Figure 3.4. Each curve is produced for a specific evaporator temperature, by simulating different temperature conditions in the generator. As the generator temperature increases, each of these curves indicates a steep increase in the coefficient of performance until a maximum value is reached. This maximum point is identified by a

particular set of sink, evaporator and generator temperature conditions.

Thus, at a given sink temperature T_s , a family of the evaporator temperature curves depicts COP maxima, as shown by the dots in Figure 3.4. Through these maxima, a curve is fitted which represents the maximum performance of the plant.

If this procedure is further extended to a range of sink temperatures, a surface of optimal conditions is produced. This surface has the characteristic that for every sink and evaporator temperatures set, T_s and T_e respectively, an optimum generator's temperature is identified which leads to a maximum coefficient of performance, Figure 3.5. These optimal generator temperatures and COP maxima were obtained from simulated data which were produced by varying the generator temperature within a range of evaporator temperatures within a range of sink temperatures.

For example, if the sink temperature (temperature of the condensate) is 30°C and refrigeration is required at -15°C, then the optimum temperature of the weak solution in the generator is 92°C. These conditions predict a maximum COP of 0.586, (see Figure 3.5).

The equations which predict the optimum values of the coefficient of performance and generator temperature have the form of polynomials in eq. (3.1) and (3.2) respectively.

These relations apply for the temperature ranges: $T_s=5^\circ\text{C}$ to 45°C and $T_E=-55^\circ\text{C}$ to -5°C

$$(3.1) \quad COP_{max} = f_{(T_s)}^0 + f_{(T_s)}^1 \cdot (T_E) + f_{(T_s)}^2 \cdot \frac{1}{(T_E)} + f_{(T_s)}^3 \cdot \frac{1}{(T_E)^2}$$

$$(3.2) \quad T_s^{opt} = g_{(T_s)}^0 + g_{(T_s)}^1 \cdot (T_E) + g_{(T_s)}^2 \cdot (T_E)^2 + g_{(T_s)}^3 \cdot (T_E)^3 \quad T_s \text{ is taken to be the generator's representative temperature}$$

where

$$(3.3) \quad f_{(T_s)}^j = a_0^j + a_1^j \cdot (T_s) + a_2^j \cdot (T_s)^2 + a_3^j \cdot (T_s)^3 \quad \text{for } j=0 \dots 3$$

$$(3.4) \quad g_{(T_s)}^j = b_0^j + b_1^j \cdot (T_s) + b_2^j \cdot (T_s)^2 + b_3^j \cdot (T_s)^3 \quad \text{for } j=0 \dots 3$$

Further the coefficients for eq. (3.3) and (3.4) above are tabulated below in Table 3.1.

Table 3.1			
	COP _{max}		T _s ^{opt}
a ₀ ⁰	6.55132 x 10 ⁻¹	b ₀ ⁰	-7.26209
a ₁ ⁰	-1.56484 x 10 ⁻²	b ₁ ⁰	2.91205
a ₂ ⁰	9.30096 x 10 ⁻⁴	b ₂ ⁰	-1.86364 x 10 ⁻²
a ₃ ⁰	-1.36564 x 10 ⁻⁵	b ₃ ⁰	1.85319 x 10 ⁻⁴
a ₀ ¹	4.41648 x 10 ⁻³	b ₀ ¹	-2.58944
a ₁ ¹	-2.77987 x 10 ⁻⁴	b ₁ ¹	1.24279 x 10 ⁻¹
a ₂ ¹	1.98907 x 10 ⁻⁵	b ₂ ¹	-4.42468 x 10 ⁻³
a ₃ ¹	-2.66527 x 10 ⁻⁷	b ₃ ¹	4.29658 x 10 ⁻⁵
a ₀ ²	-6.19823	b ₀ ²	-6.52162 x 10 ⁻²
a ₁ ²	5.77539 x 10 ⁻²	b ₁ ²	6.98244 x 10 ⁻³
a ₂ ²	7.47419 x 10 ⁻³	b ₂ ²	-2.38075 x 10 ⁻⁴
a ₃ ²	-1.36508 x 10 ⁻⁴	b ₃ ²	2.26658 x 10 ⁻⁶
a ₀ ³	-32.7816	b ₀ ³	-8.99504 x 10 ⁻⁴
a ₁ ³	4.40948 x 10 ⁻¹	b ₁ ³	9.7008 x 10 ⁻⁵
a ₂ ³	1.80843 x 10 ⁻²	b ₂ ³	-3.27227 x 10 ⁻⁶
a ₃ ³	-3.80018 x 10 ⁻⁴	b ₃ ³	3.05454 x 10 ⁻⁸

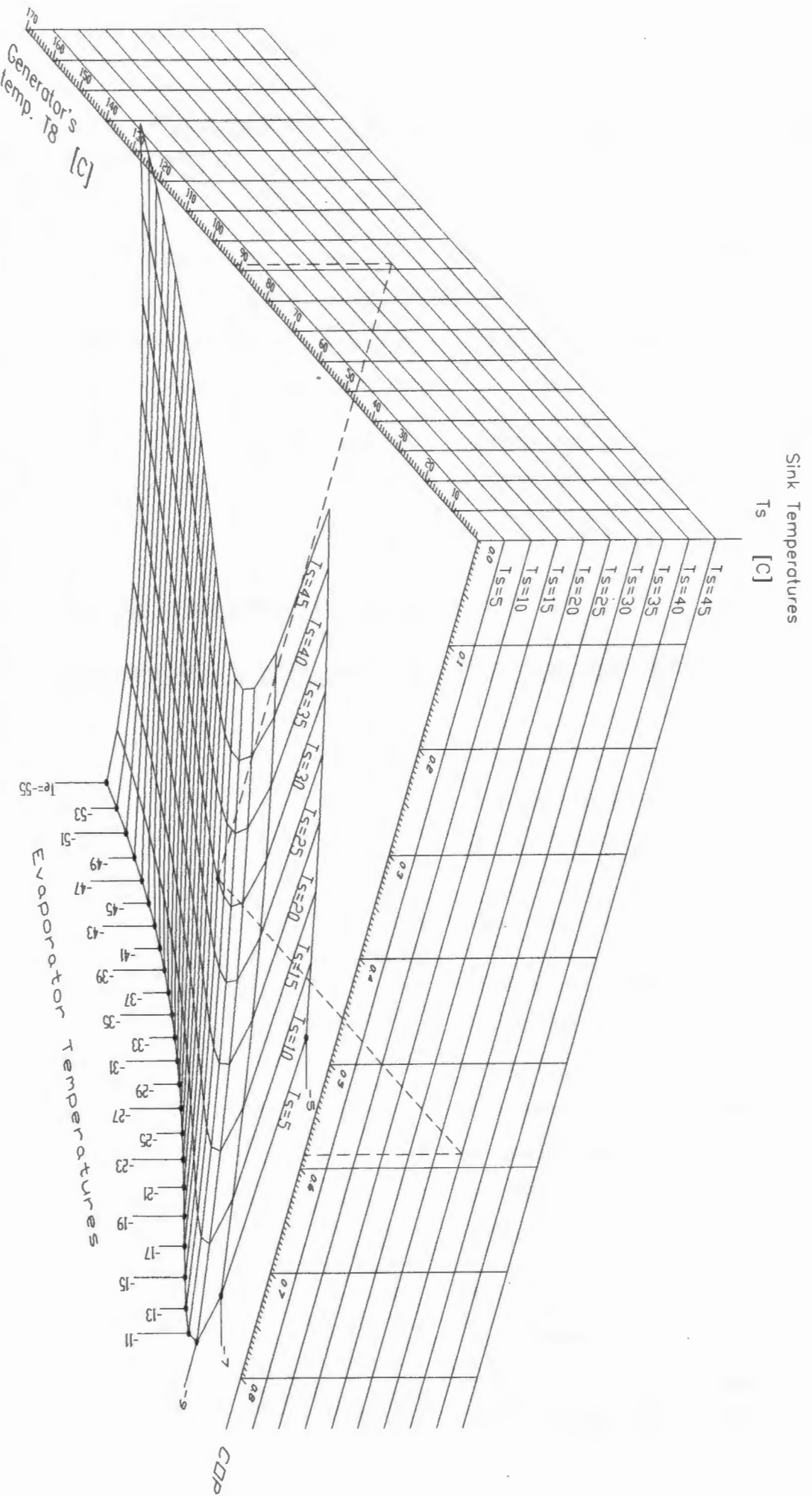


Figure 3. 5 *Optimisation curves for the ammonia-water absorption refrigeration machines*

3.2. GRAPHICAL EVALUATION OF THE HEAT EFFECTS OF THE ABSORPTION CYCLE.

3.2.1. General

The analysis presented in this chapter, is based on the design of an aqua-ammonia absorption refrigeration plant given a desired evaporator temperature and refrigeration capacity. The design will be explained with the aid of the enthalpy-concentration (h-x) diagram for the aqua-ammonia mixture as proposed by Bosnjakovic [B10].

In an attempt to support the proposed procedure in designing and optimising an ammonia-water absorption machine, certain methods from literature will be reviewed and discussed.

3.2.2 Individual component design of the plant

3.2.2.1 The evaporation process and estimation of the operating pressures

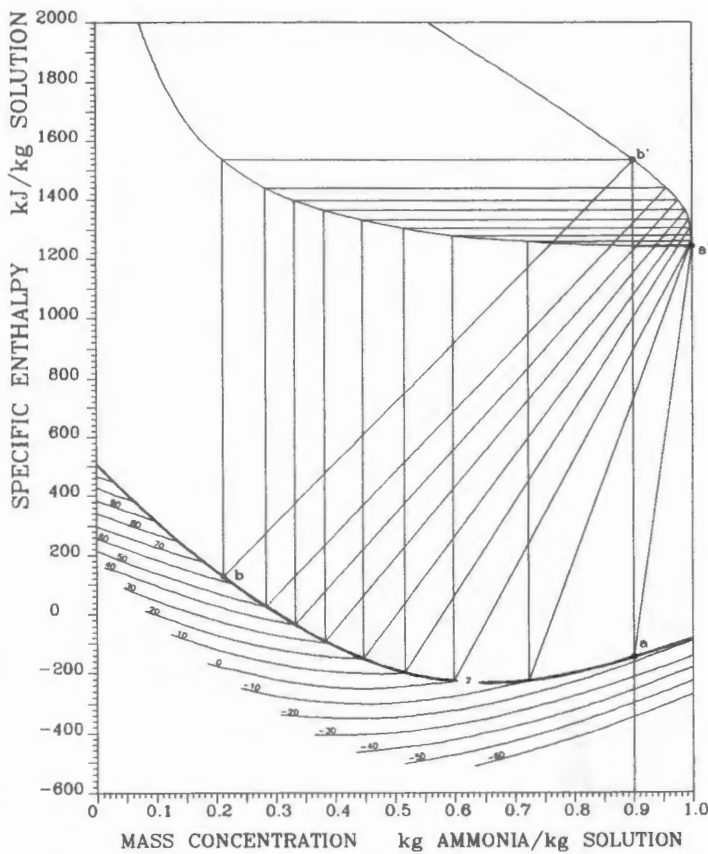


Figure 3. 6

In designing the evaporator of an absorption refrigeration machine, the following statement should be considered:

“Once the refrigerant enters the evaporator, it receives heat and evaporates. The condition in the evaporator depends upon the pressure-temperature-concentration equilibrium of the ammonia-water mixture.”

The diagram in Figure 3.6 shows liquid-vapour equilibria during the evaporation of a 0.900 concentration mixture at 2bar vapour pressure.

The saturation temperature of the liquid with concentration 0.900 is approximately -18°C . The construction diagram shows the concentration of its vapour at position a' . The remaining liquid (lower in concentration) receives heat at a higher temperature and its vapour is at a lower concentration too. This remaining liquid could only evaporate completely at point b' where its saturation temperature is approximately 62°C . This is shown by the isotherm $b-b'$.

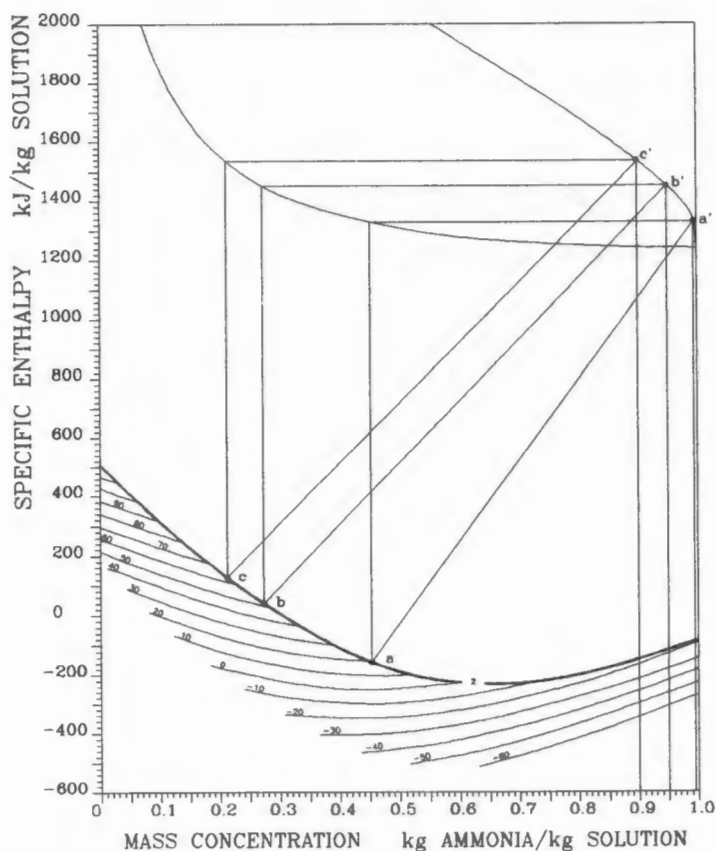


Figure 3. 7

The construction lines in Figure 3.7 show complete vaporisation of different concentration aqua-ammonia mixtures. The vapour at point b' has concentration 0.950 and its saturation temperature is approximately 50°C . A vapour with concentration 0.995 at point a' is saturated at a temperature approximately 20°C .

This illustrates that even at high concentrations approaching pure ammonia, complete evaporation of the refrigerant liquor, will require high saturation temperatures which are undesirable in the evaporator.

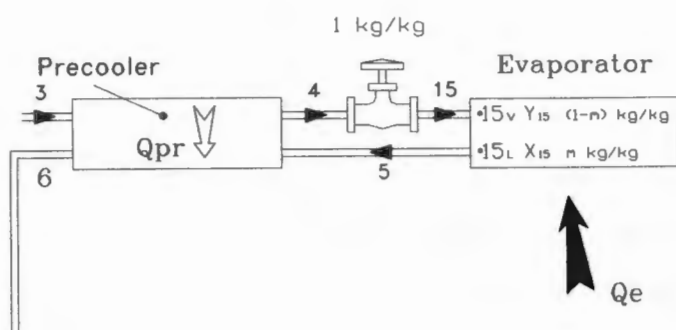


Figure 3. 8

Despite the above, literature search shows that it is customary, as an exercise, to assume saturated vapour conditions at the exit of the pre-cooler, such as points a' , b' and c' , as shown in Figure 3.7. According to Figure 3.8 which shows a schematic evaporator and pre-cooler assembly, it is claimed

by Threlkeld [T2], that the refrigeration effect is the enthalpy difference either (h_5-h_4) or (h_6-h_3) , and assumes that point 6 is on the saturated vapour curve. Thus the refrigeration effect is maximised.

This is exactly one of the stumbling points, where theoretical analyses presented in publications [R2, T2, W2] design the evaporator from data collected from existing plants and not from the refrigeration requirements.

However, it is correctly assumed that at points 3 and 15 the liquid is saturated. As shown in Figure 3.9(a), the high and low pressure curves are drawn through the intersection points between an ammonia concentration x_3 (a design parameter) and given sink T_s and evaporator T_{15} temperatures respectively. But what Figure 3.9(a) also shows is, that point 6 and its saturation temperature T_6 are established independently to the location of point 3, and hence to the sink temperature T_s . Because points 3 and 6 belong to the “hot-side” streams of the pre-cooler, Figure 3.8, the exit temperature T_6 must be always less or equal to the sink temperature T_s . Because this condition is violated in Figure 3.9(b), the assumption that point 6 is saturated,

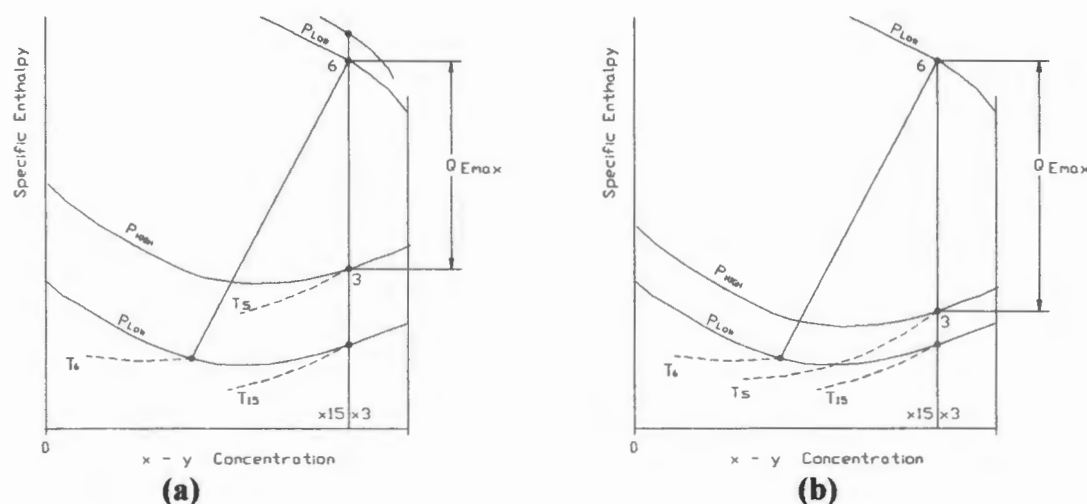


Figure 3.9

is invalidated.

Of course there is a concentration $x_3=x_{15}$, which together with the temperature T_{15} will produce a low pressure P_{LOW} , such that the vapour at point 6 will have a saturation temperature T_6 less or at least equal to the sink temperature T_s . But because $P_{\text{LOW}}-T_{15}-x_{15}$ are

linked with an equilibrium constraint, assuming a value for either P_{LOW} or x_3 renders the procedure redundant.

PROPOSED PROCEDURE

Before any other process is analysed, a decision must be made on the purity of the refrigerant/absorbent mixture leaving the distillation column. The literature search on the subject established that a concentration within the range of 0.990 to 0.995 is quite common. This mixture in the condenser and evaporator establishes the working pressures of the system.

The analysis of the evaporator (without a pre-cooler) using the enthalpy-concentration diagram, begins by assuming that the concentration of the ammonia is 0.990 and the average evaporator temperature T_E is given. The average evaporator temperature T_E , is bounded between two limits, $T_{Emin}=T_{15}$ and $T_{Emax}=T_5$. The low temperature limit, T_{15} is the temperature which the ammonia/water mixture attains at the entrance to the evaporator at point 15. The upper temperature limit T_5 occurs at the exit of the evaporator, point 5. It is the designer's task to define these two temperature limits.

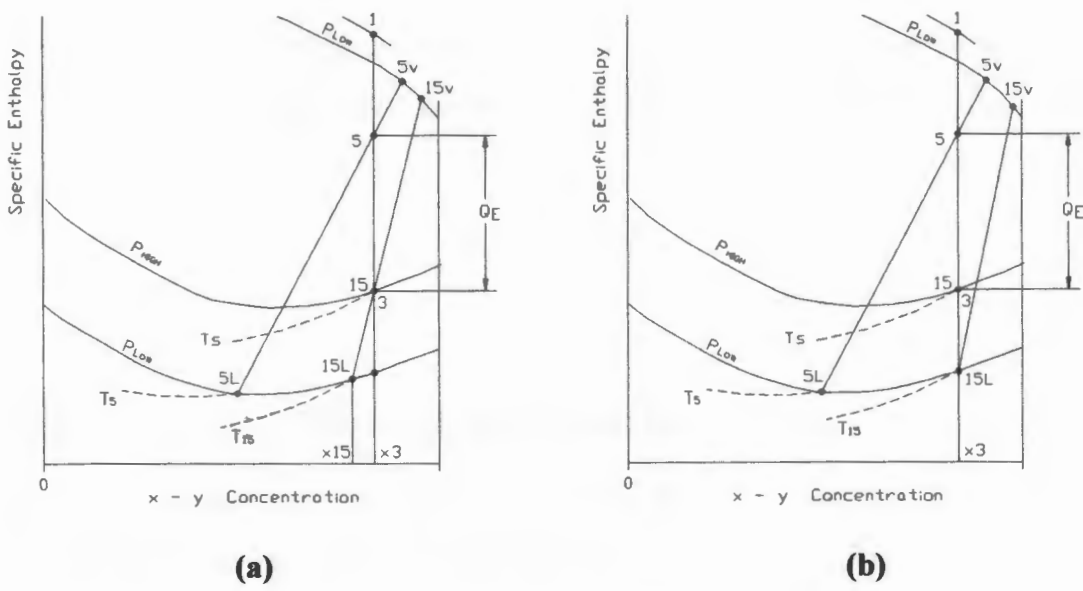


Figure 3.10 Schematic h - x diagram of the evaporation process

From point 15, which is the inlet to the evaporator, and onward, the mixture receives heat at constant pressure.

As the mixture passes through the throttling valve, it “flushes” and the first vapour produced will have a concentration equivalent to the saturation point 15_v along the isotherm T₁₅ in equilibrium with the liquid at point 15_L, Figure 3.10(a).

At ammonia concentrations such as 0.990 and higher, the isotherms are almost vertical lines. Also because the mass of the vapour produced at the throttling valve is negligible compared to the liquid it can be assumed, within the accuracy of the enthalpy-concentration diagram, that the concentrations x_{15L} and x₃ are identical.

The construction of Figure 3.10(b) starts with the location of point 15_L at the intersection of the temperature curve T₁₅ and the concentration line x₃ = 0.990. At this intersection, the low pressure P_{LOW} of the cycle is indicated.

The vapour point 15_v is at the intersection of the isotherm T₁₅ and the vapour low pressure curve. With a similar construction, the isotherm T₅ locates the points 5_L and 5_v at the low pressure liquid and vapour curves respectively.

The environmental sink temperature T_s intersects the 0.990 concentration line at point 3. The liquid pressure curve which passes through point 3 is the high pressure of the cycle.

The concentration line x₃ intersects the isotherm 5_L-5_v at point 5 which represents the condition of the mixture at the exit of the evaporator. The refrigeration effect is the enthalpy difference (h₅-h₁₅).

3.2.2.2 Vapour-liquid heat exchanger (pre-cooler)

Since it is desirable that the enthalpy of the refrigerant be lower than it is when it leaves the condenser, the use of a pre-cooler is indicated.

An energy balance around the pre-cooler yields:

$$h_6 - h_5 = h_3 - h_4 = Q_{PR}$$

The maximum heat transfer Q_{PRmax} is:

$$C_{min} \cdot (T_3 - T_5) = Q_{PRmax}$$

and the effectiveness is:

$$\varepsilon = \frac{Q_{PR}}{Q_{PRmax}} = \frac{(T_6 - T_5) \cdot C_c}{(T_3 - T_5) \cdot C_{min}}$$

Since $C_c = C_{min}$ the effectiveness becomes:

$$\varepsilon = \frac{(T_6 - T_5)}{(T_3 - T_5)}$$

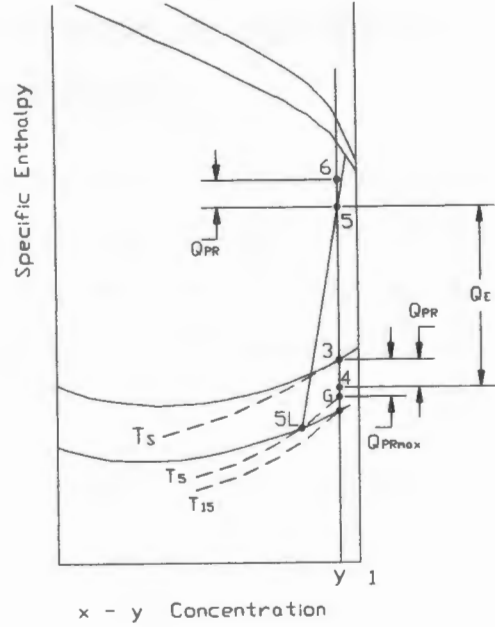


Figure 3. 11

For an effectiveness $\varepsilon=1$, the above equation gives:

$$T_6 = T_3 = T_5.$$

Notwithstanding the comments in paragraph 3.2.2.1, the pre-cooler is designed to receive vapour at temperature T_5 (evaporator's design specification) and deliver vapour at a temperature approaching T_5 (condenser's conditions).

According to Bosnjakovic [B10], the construction of Figure 3.11 shows the quantity Q_{PRmax} as the enthalpy difference between point 3 and point G at the intersection of the temperature curve T_5 with the 0.990 line. By assuming an effectiveness, $\varepsilon=0.7$ the pre-cooler's load is set as the enthalpy difference:

$$(h_3 - h_4) = \varepsilon \cdot Q_{PRmax}$$

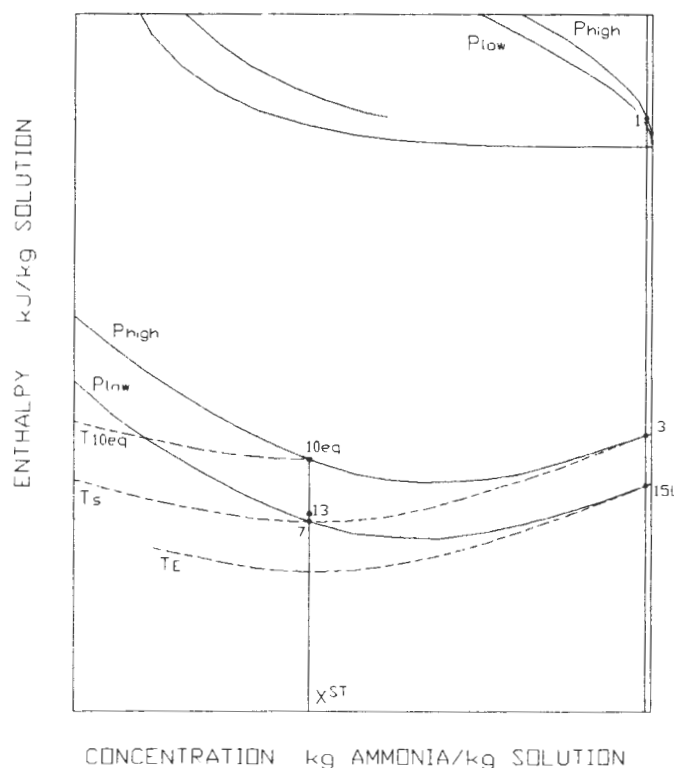
Figure 3.11 shows a condition in which point 5 is in the wet vapour region.

At concentrations near pure ammonia where the isotherms are almost vertical, the temperature difference $(T_5 - T_{15})$ will approach zero and the vapour at point 5 will approach saturation conditions.

3.2.2.3 Determining the conditions at the condenser and the strength of the strong solution required for the plant

With reference to Figure 3.1, the condenser receives vapour at point 1. This point represents a high pressure saturated vapour which has the same ammonia concentration as the saturated liquid at point 3. In Figure 3.12, the vapour high pressure curve together with the concentration 0.990 locates point 1. The enthalpy difference ($h_1 - h_3$) is the condenser's load Q_C per kg of refrigerant.

The low pressure curve and the sink temperature T_s intersect at point 7 which fixes the concentration of the strong solution x^{ST} .



After the action of the pump, the strong solution at point 13 (see Figure 3.1) maintains its concentration and it is at high pressure, but with hardly any change in temperature. The position of point 13 can be established, once the increase of the solution's enthalpy due to the work W_P of the pump is calculated. Because W_P is small compared to the other heat effects in the cycle, point 13 is expected to be very close and above point 7. The process to locate point 13 is described in paragraphs 3.2.2.6 and 3.2.2.7.

Figure 3. 12 *h-x diagram*

The strong solution concentration line intersects the high pressure curve at point 10eq. Through this point the temperature curve T_{10eq} passes which represents the state at which vapour liberation is initiated.

3.2.2.4 Determining the conditions at the generator

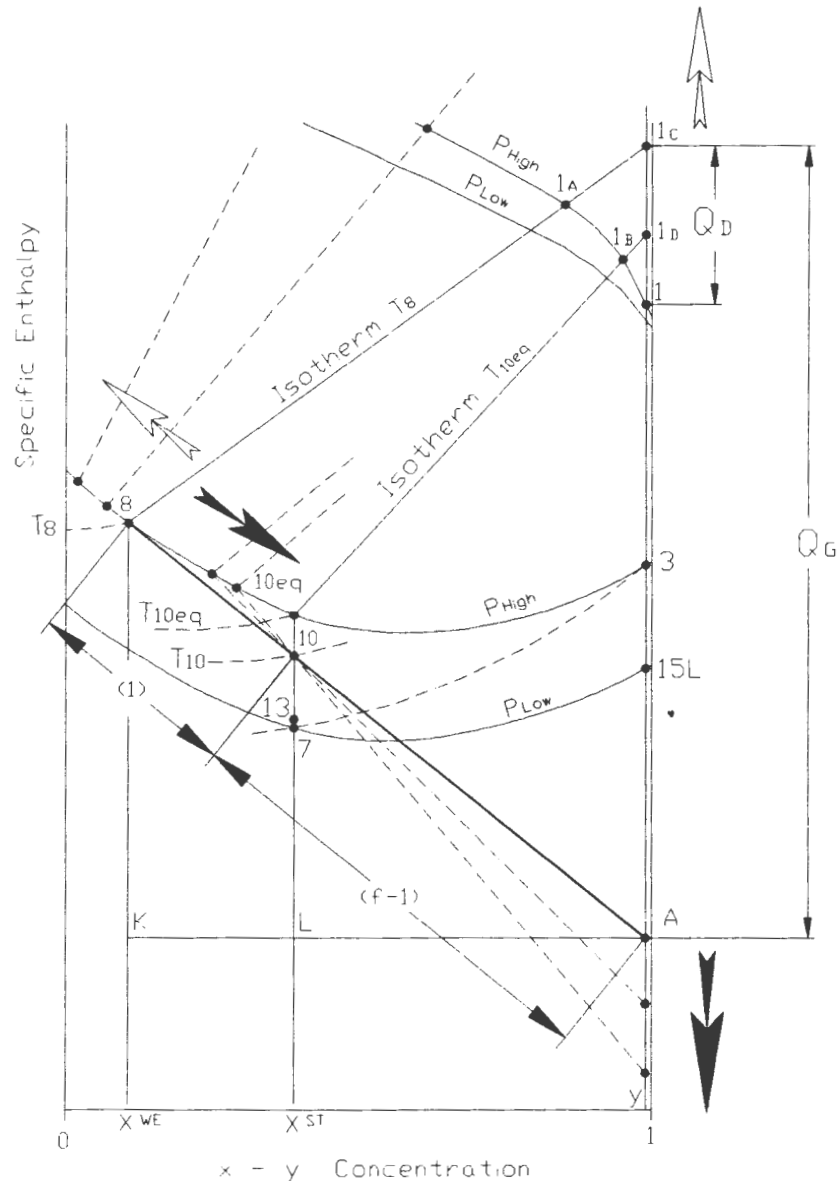


Figure 3. 13

The next step aims at establishing the concentration of the weak solution leaving the generator on its way to the absorber at point 8. This is achieved in the following way:

With reference to Figure 3.13, Point 7 represents the conditions of the strong solution as it leaves the absorber. The strong liquor is pumped through the heat exchanger, (see Figure 3.1) from a non-equilibrium state at point 13 and arrives at the generator at a condition described by point 10. Heat supplied to the generator raises the temperature of this liquor to its

saturation condition at point 10_{eq} . At this point extra heat supplied to the generator will liberate ammonia and will increase the saturation temperature of the solution to values higher than $T_{10_{eq}}$. Any of these temperatures could be the representative generator temperature T_8 , which immediately establishes the concentration of the weak solution x^{WE} .

It is precisely at this point that the procedure demands improvement. Surely this is the point where the designer requires the optimum generator conditions. If the conditions in the generator are not optimised, then the heat demand in the generator is not minimal. This is explained by the following step-by-step procedure:

The isotherms through points 8 and 10_{eq} intersect the vapour high pressure curve at points 1_A and 1_B . These are the vapour points in equilibrium with the weak solution and strong solutions respectively. The T_8 and $T_{10_{eq}}$ isotherms, when extended intersect the refrigerant ammonia concentration line at points 1_C and 1_D .

When the generator operates at a temperature T_8 , according to distillation theory [F1], point 1_C represents the point of minimum reflux. The enthalpy difference $(h_{1_C} - h_1)$ is the heat Q_D rejected from the deflegmator.

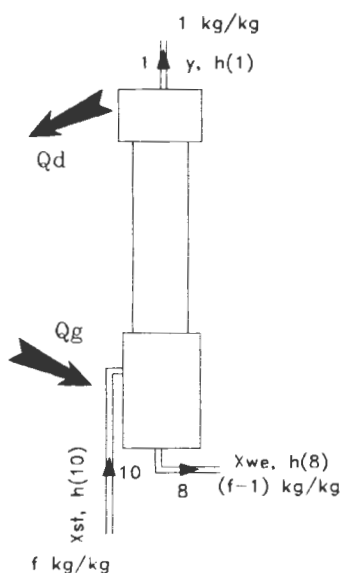


Figure 3.14 Mass and heat fluxes around the generator and deflegmator. Refer to Figure 3.1 for actual position within the plant.

Assume that the strong solution is heated in the heat exchanger from its condition at point 13 to a temperature T_{10} which passes through point 10 on the x^{ST} concentration line, Figure 3.13.

The generator receives strong solution with concentration x^{ST} at a mass flow rate of (f) kg so that 1kg of vapour with concentration y is released, Figure 3.14.

At the same time $(f-1)$ kg of weak solution with concentration x^{WE} is returned to the absorber.

A heat balance on the ammonia flow through the generator/distiller yields:

$$(3.5) \quad Q_G - Q_D = f \cdot (h_8 - h_{10}) + (h_1 - h_8)$$

and the mass balance gives:

$$y \cdot 1 + x^{WE} \cdot (f - 1) = f \cdot x^{ST}$$

or

$$(3.6) \quad f = \frac{y - x^{WE}}{x^{ST} - x^{WE}}$$

On the h-x diagram of Figure 3.13, a point "A" can be located on the y concentration such that:

$$h_A = h_8 - f \cdot (h_8 - h_{10})$$

This is the equation of a straight line of negative slope (f), joining point 8 to point A and passing through point 10.

Substituting into eq. (3.5) the heat required by the generator becomes:

$$(3.7) \quad Q_G = Q_D + h_1 - h_A$$

Eq. (3.6) is represented in Figure 3.13 by the ratio of the linear segments that constitute the line 8-A. If a unit length is assigned to the length $(x^{ST} - x^{WE})$, then $(y - x^{WE}) = f$ and $(y - x^{ST}) = f - 1$. From the similar triangles 8-K-A and 10-L-A it can be observed that the ratio (8-10):(10-A) represents the ratio of the mass flow rates of the vapour to the weak solution. That is, $1:(f-1)$

Eq. (3.6) also indicates that the rate of relative circulation (f), is inversely proportional to the concentration difference $(x^{ST} - x^{WE})$. Since the concentration x^{ST} is fixed, it can be depicted from Figure 3.13, that $(x^{ST} - x^{WE})$ depends on the temperature difference $(T_8 - T_{10eq})$.

If the generator's temperature T_8 is reduced towards the limit temperature T_{10eq} , point 1C will also reduce towards the low limit point 1D, and the difference $(x^{ST} - x^{WE})$ will diminish. The former reflects to the deflegmator's heat Q_D approaching a minimum value $(h_{1D} - h_1)$, and the latter reflects to a rapid increase in the heat rate through the generator, due to an excessively high rate of relative circulation (f).

This is shown by the big solid arrows on the enthalpy-concentration diagram of Figure 3.13, which together with eq. (3.7) indicates that larger amounts of heat have to be supplied to the generator in order to maintain a constant mass rate of vapour.

On the other hand an increase of the generator temperature leads to a steeper isotherm T_8 which in turn corresponds to a larger amount of heat Q_D rejected from the deflegmator. This also increases rapidly the heat demand in the generator which is shown by the big hollow arrows on the enthalpy-concentration diagram.

In addition to the above, for a particular temperature T_8 , a reduction of the heat demand Q_G is indicated as T_{10} approaches T_{10eq} .

In order to minimise the heat Q_G required by the generator¹ the temperature T_8 must be optimised to T_8^{opt} and the heat exchanger must be designed to supply the solution at its saturation temperature T_{10eq} .

¹

The term "heat required by the generator" represents the net heat received by the fluids in the generator/distiller assembly, for the production of a constant vapour mass rate at concentration y at the exit of the distillation column.

3.2.2.5 Purification of the refrigerant mixture

Rectification is a well-known process and literature is available for a vast number of mixtures.

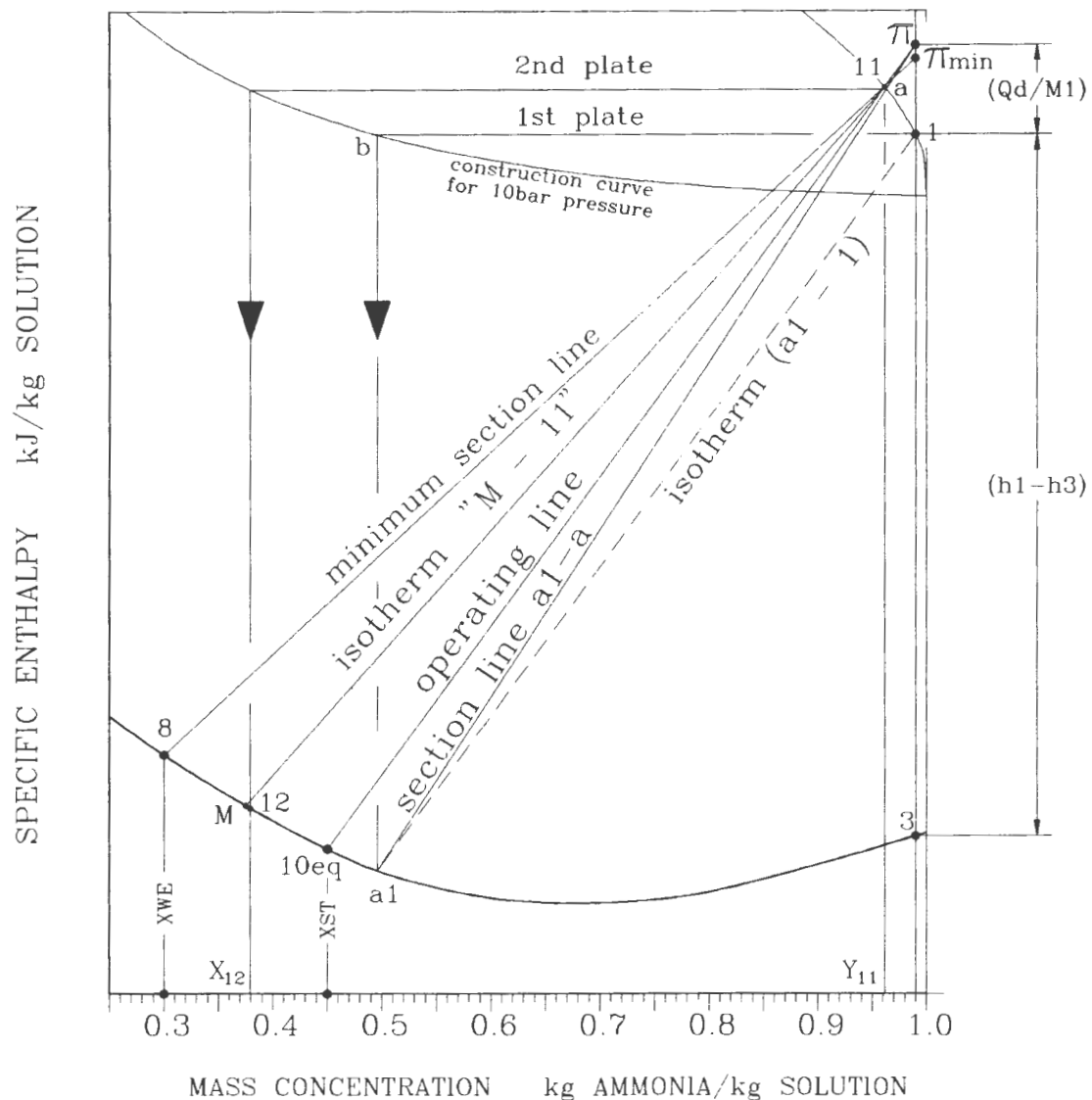


Figure 3.15 The Ponchon-Savarit diagram depicting the generator's and the concentrating column's working conditions for an aqua-ammonia mixture at 10 bars pressure

Due to mixing of the solutions in the generator, it is assumed that the bulk solution has a concentration which is the average between the weak and strong solutions. This solution has concentration x^M and is in equilibrium with its vapour of concentration y_{11} leaving the generator, Figure 3.15. Since the vapour concentration y_{11} is less than the ammonia concentration 0.990 at point 1, vapour purification is required.

The purity of the refrigerant ammonia/water vapour mixture leaving the distillation column at point 1, can be achieved by a number of plates in the column determined by either of the two well known methods:

- the McCabe-Thiele method, which uses a graphical approach plotting values on a vapour concentration versus liquid concentration diagram. This method requires a prior knowledge of the required reflux ratio.
- the Ponchon-Savarit method, in which specific enthalpy values of the binary mixture are plotted against liquid-vapour concentrations. This is a most popular approach as in deriving the number of theoretical plates, the reflux ratio can also be established. This method is illustrated in Figure 3.15 and supported by the explanation to follow:

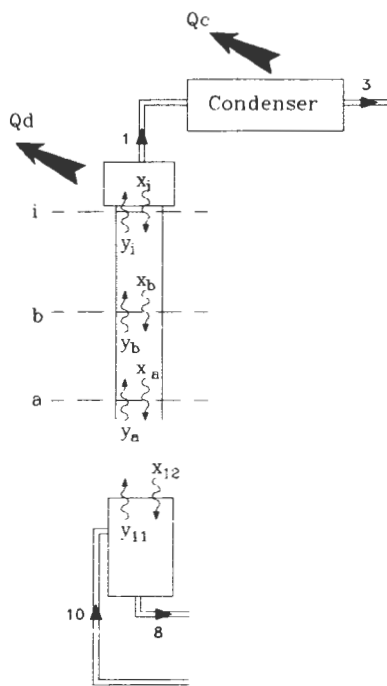


Figure 3.16

For any part of the column between two cross-sections, such as “a” and “b” in Figure 3.16, the mass flux entering the control section equals that leaving:

$$M_{ya} - M_{xa} = M_{yb} - M_{xb}$$

The same relationship holds for any other control section along the column and therefore it must equal to a constant. If the control section includes the deflegmator, that is between sections “1” and “i” the constant will be the mass flux of the distillate.

Thus the mass balance equation becomes:

$$M_{yi} - M_{xi} = M_1$$

The conservation of ammonia mass between sections “i” and “1” yields:

$$M_{yi} \cdot y_i - M_{xi} \cdot x_i = M_1 \cdot x_1$$

The heat balance around the deflegmator yields:

$$M_{yi} \cdot h_{yi} - M_{xi} \cdot h_{xi} = M_1 \cdot h_1 + Q_D$$

A combination of the above equations gives the governing equation for the deflegmator's heat per kg of distillate.

$$\frac{M_{yi}}{M_1} = \frac{y_1 - x_i}{y_i - x_i}$$

$$\frac{M_{yi}}{M_1} \cdot (h_{yi} - h_{xi}) = (h_1 - h_{xi}) + \frac{Q_D}{M_1}$$

If the control section includes the whole distillation column, then x_i and y_i are substituted by x_{12} and y_{11} respectively.

The distillation column receives vapour of concentration y_{11} at point 11 and returns to the generator solution of concentration x_{12} at point 12, Figure 3.16.

The concentration x_{12} is identified by constructing the Ponchon-Savarit diagram of Figure 3.15 in the following way:

The vapour at point 11 is saturated and in equilibrium with the mixture with concentration x^M .

The position of the pole $\pi_{\min}$ is located at the extension of the line joining points 8 and 11 to the ammonia concentration 0.990 line [B10, F1]. An operation under this pole position would require an infinite number of plates in the column. The position of the operating pole π is at the extension of the line joining points 10_{eq} and 11 to the ammonia concentration 0.990, [F1]

Starting from point 1 ($y_1=0.990$), a horizontal line meets the construction curve at point "b" from which a vertical line locates point "a₁" on the liquid high pressure line. The reflux liquid at point "a₁" is in equilibrium with the vapour at point 1 and these two points are joint with the isotherm (a₁-1). The section line from the pole π to point "a₁" intersects the vapour high pressure line at a point "a". Since the liquid at point "a₁" has concentration higher than that of the strong solution (which is considered the "feed" to the column), a second plate is required.

A similar construction for the second plate locates point 12 on the liquid high pressure line. Since the reflux liquid at point 12 has concentration less than that of the feed (point 10_{eq}) a third plate is not required.

It can be seen from Figure 3.15 that the location of the reflux liquid at point 12 is very close to that of the bulk solution at point M.

Construction of diagrams similar to that of Figure 3.15 for different pressures and combination of weak and strong solutions, also indicate the same result.

It was felt appropriate to assume that the concentrations of the reflux liquid x_{12} and of the bulk solution x^M are equal.

Thus the reflux heat Q_D is measured in relation to known quantities:

$$(3.8) \quad Q_D = M_1 \cdot \left[\frac{y_1 - x_{12}}{y_{11} - x_{12}} \cdot (h_{11} - h_{12}) - (h_1 - h_{12}) \right]$$

The quantity $\frac{Q_D}{M_1}$ is the heat rejected from the deflegmator for the production of 1kg of distillate of concentration y_1 .

The amount of the reflux liquid M_R to the top plate of the column is calculated by:

$$(3.9) \quad M_R = M_1 \cdot R$$

where R is called Reflux Ratio, which is defined as:

$$(3.10) \quad R = \frac{\frac{Q_D}{M_1}}{h_1 - h_3}$$

The above quantities can be evaluated graphically on the enthalpy-concentration diagram as shown in Figure 3.15.

3.2.2.6 The solution pump

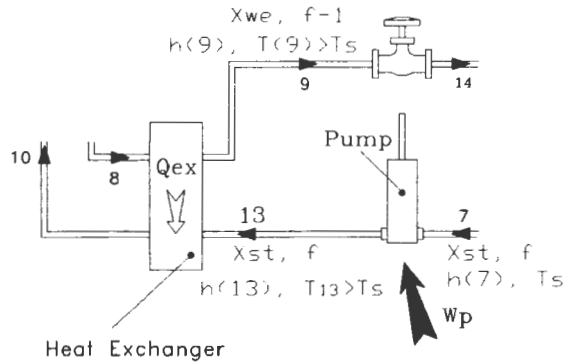


Figure 3.17

A heat balance around the pump gives:

$$W_P = f \cdot (h_{13} - h_7)$$

This value is reflected by the enthalpy difference ($h_B - h_A$) in Figure 3.15. The enthalpy difference ($h_{13} - h_7$) is equal to the heat supplied to the strong solution. The pump load W_P is proportional to the specific volume of the solution v_f :

$$(3.11) \quad W_P = \frac{1}{\eta_P} \cdot f \cdot (P_{High} - P_{Low}) \cdot v_f$$

where η_P is the mechanical efficiency of the pump.

and v_f is calculated by the polynomials described in section 3.4.4

3.2.2.7 Optimising the solution heat exchanger

The strong solution in the stream between points 13 and 10 receives heat from the hot solution in the streams between points 8 and 9, Figure 3.17.

Thus, the temperature of the strong solution will rise along the same concentration x^{ST} , from point 13 to point 10, while the temperature of the weak solution will fall along the same concentration x^{WE} , from point 8 to point 9, Figure 3.18.

An energy balance around the heat exchanger in Figure 3.17 would give:

$$\frac{h_8 - h_9}{h_{10} - h_{13}} = \frac{f}{f - 1}$$

The strong solution at point 13 is not saturated, hence the location of point 13 is found indirectly after the pump's load is evaluated by eq. (3.11).

On the y concentration line of Figure 3.18, a point B is located such that the enthalpy difference $(h_B - h_A) = W_P$. A line joining point B to point A, intersects the strong solution concentration line at point 13.

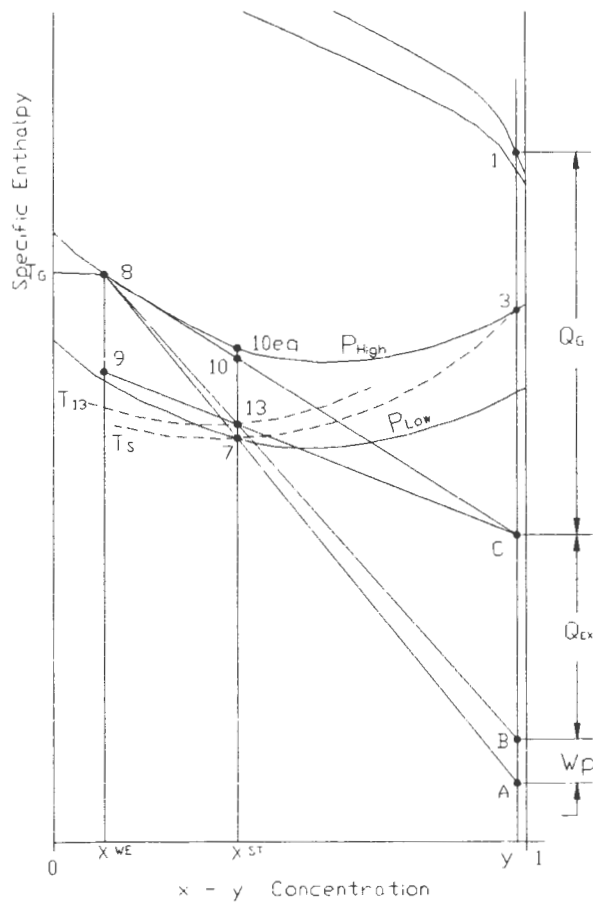


Figure 3. 18

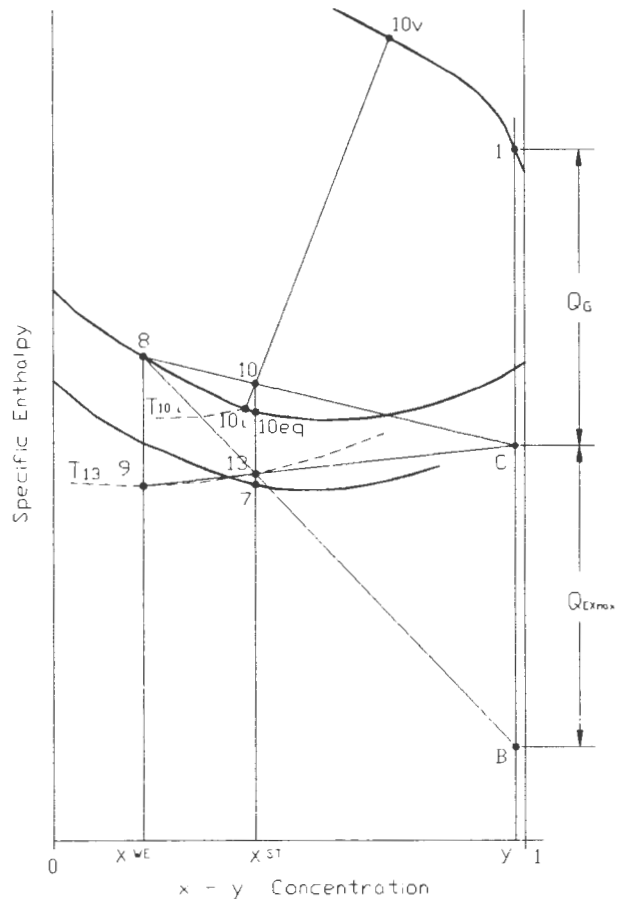


Figure 3. 19

If the heat exchanger supplies the strong solution at a condition described by point 10, then point C is located at the extension of the line 8-10 to the y concentration line. The enthalpy difference $(h_{10} - h_{13})$ is the heat received by the strong solution per kg of strong solution. This quantity is equivalent to the enthalpy difference $(h_B - h_C)$ per kg of vapour of concentration y and it is shown by the triangles 8-10-13 and 8-C-B.

It is also the same heat supplied by the weak solution. A line drawn from point C through point 13, intersects the weak solution concentration line at point 9. The enthalpy difference (h_8-h_9) is the heat supplied by the weak solution per kg of weak solution.

From Figure 3.18, the triangles C-8-9 and C-10-13 show that the heat $(f-1).(h_8-h_9)$ is the equal to the heat $(f).(h_{10}-h_{13})$.

The limit position of point 9 is shown in Figure 3.19 where $T_9=T_{13}$ and the weak solution imparts the maximum heat to the strong solution.

If the amount of heat exchanged is sufficient to raise the temperature of the strong solution beyond the equilibrium point 10_{eq} , that is $(h_{10}-h_{13}) > (h_{10eq}-h_{13})$, then boiling will occur and there will be a change in the concentration of the strong solution. The total concentration of point 10 in Figure 3.19, is x^{ST} , but the liquid and the vapour have concentrations corresponding to the equilibrium points 10_L and 10_V respectively.

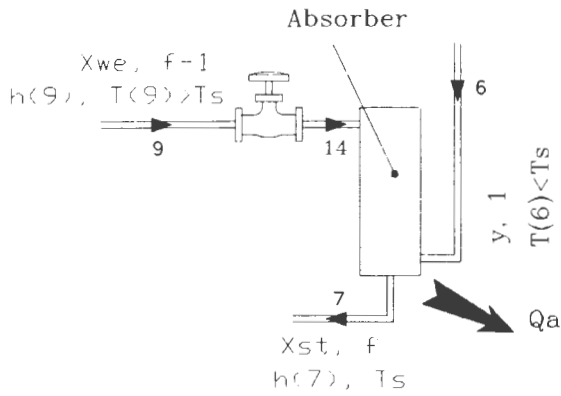
Because in the presence of a two-phase flow in tubes there is a notable reduction of the heat transfer coefficient, this is an undesirable condition in the heat exchanger. Therefore the heat exchanger should be designed to deliver the strong solution at its equilibrium temperature marked by point 10_{eq} .

3.2.2.8 The absorption process

The absorber receives a weak solution at point 14, the properties of which depend upon the throttling of the solution at point 9. A heat and mass balance around the throttle, Figure 3.20 would give:

$$h_9 = h_{14} \quad x_9 = x_{14} = x^{WE}$$

The throttling of the weak solution is shown in the h-x diagram of Figure 3.21 where, points 9 and 14 coincide, but the respective streams have different state properties.



A heat balance around the absorber in Figure 3.20 gives:

$$(3.12) \quad Q_A = h_6 - h_9 + f \cdot (h_9 - h_7)$$

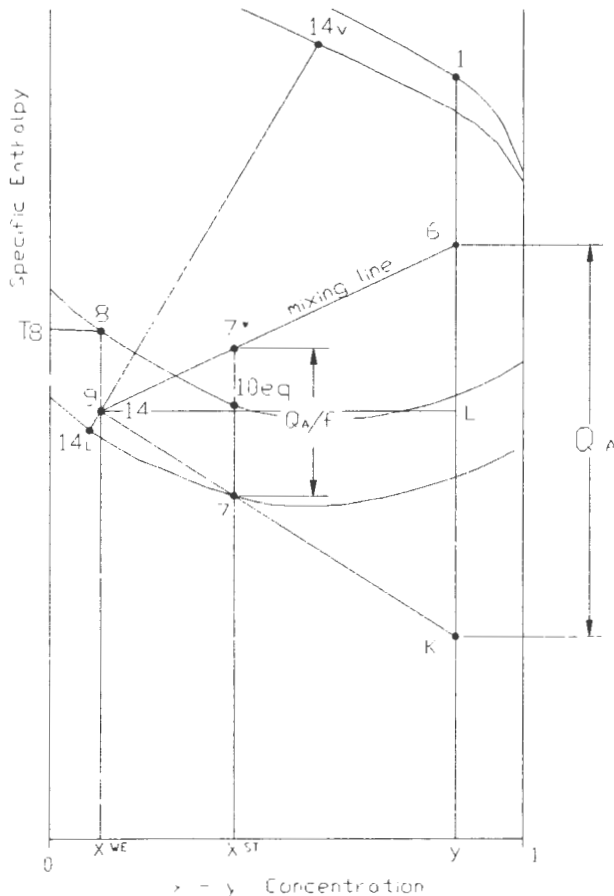
Which is represented by the line

$$Q_A = h_6 - h_K$$

Figure 3. 20

Point K is located on the y concentration line such that:

$$h_K = h_9 - f \cdot (h_9 - h_7)$$



The line 14-6 represents the mixing line between the weak solution and the vapour. If the mixing is done adiabatically, the resulting solution will have concentration x^{ST} and enthalpy h_7^* . The process between points 7 and 7^* is entirely in the wet vapour region and because the absorber delivers liquid and not vapour, the wet vapour mixture at point 7^* must be liquefied. The total heat Q_A rejected from the absorber during this process is the sum of the heat $(h_L - h_K)$ involved to cool the weak solution and the heat of mixing $(h_6 - h_L)$. This is equivalent to the heat Q_A/f rejected from the absorber per unit mass of strong solution.

Figure 3. 21 The throttling of the weak solution and the absorption processes

3.3 A CASE STUDY

The examples discussed in references [R2] and [T2] are typical cases of analyses for existing units. The present example will demonstrate the usage of the h-x diagram for a possible solution to a refrigeration demand. The h-x diagram of Figures 5.22 to 5.25 is that of [K10].

3.3.1 Problem statement

It stems from the discussion of section 3.2.2.1, that the evaporator temperature will not be constant. Therefore a range of working temperatures must be specified rather than a single evaporator temperature.

Assume that, due to refrigeration requirement the cooling coils must be designed to operate between -15°C and -12°C . Also due to cooling in the condenser and absorber the temperature of the condensate and of the solution is 30°C .

3.3.2. A step-by-step analysis

Step-1 Determining the operating pressures of the system

The analysis begins with an assumed value of the concentration of the condensate (which is usually quite high) 0.990

On the h-x diagram, Figure 3.22, a vertical line is drawn at the 0.990 concentration value which intersects the 30°C and the -15°C temperatures, which constitute the design conditions. The intersection points 3 and 15_L determine the high pressure (11.34bar) and low pressure (2.2bar) respectively.

Step-2 Determining the inlet and outlet conditions of the cooling coils

The evaporator exit temperature -12°C , also a design condition, marks point C on the liquid low pressure line and point E on the vapour low pressure line, Figure 3.22. The isotherm C-E crosses the 0.990 concentration line at point 5 which marks the state of the wet vapour at the evaporator's exit.

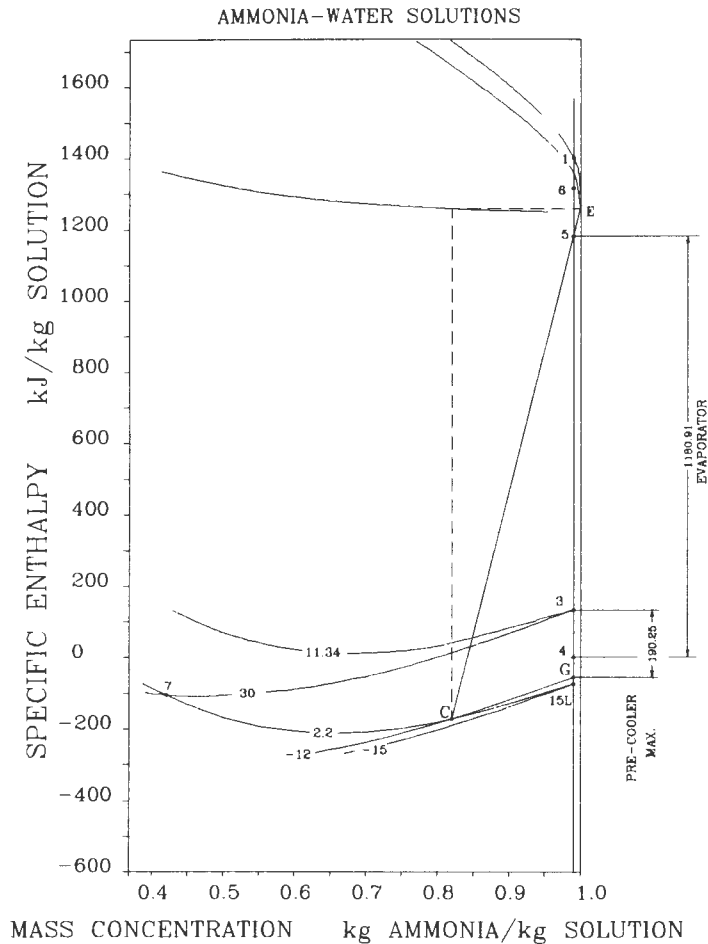


Figure 3.22

Step-3 Determining the condenser's condition

Vapour from point 1 condenses to liquid at point 3. Point 1 is located at the intersection of the vapour high pressure line and the concentration 0.990. The enthalpy difference (h_1-h_3) is the heat rejected from the condenser (1265 kJ/kg)

Step-4 Determining the strong solution concentration

The strong solution concentration is obtained by drawing a vertical line from the intersection of the low pressure and the design sink temperature $T_s=30^\circ\text{C}$, to the base, Figure 3.23. The strong solution concentration line intersects the liquid high pressure line at point 10_{eq}. This point marks the minimum temperature for vapour liberation from the generator.

The -12°C temperature curve intersects the 0.990 concentration line at point G. The enthalpy difference (h_3-h_G) indicates the theoretical maximum heat transfer through the pre-cooler. In this procedure, 70% of this maximum heat is transferred from the condensed liquid to the cold vapour.

Thus point 4 is established and the heat (h_5-h_4) specifies the refrigeration effect (1180.91 kJ/kg). The exit of the pre-cooler, point 6 can be located on the $y=0.990$ concentration vertical line such that $(h_6-h_5)=(h_3-h_4)$.

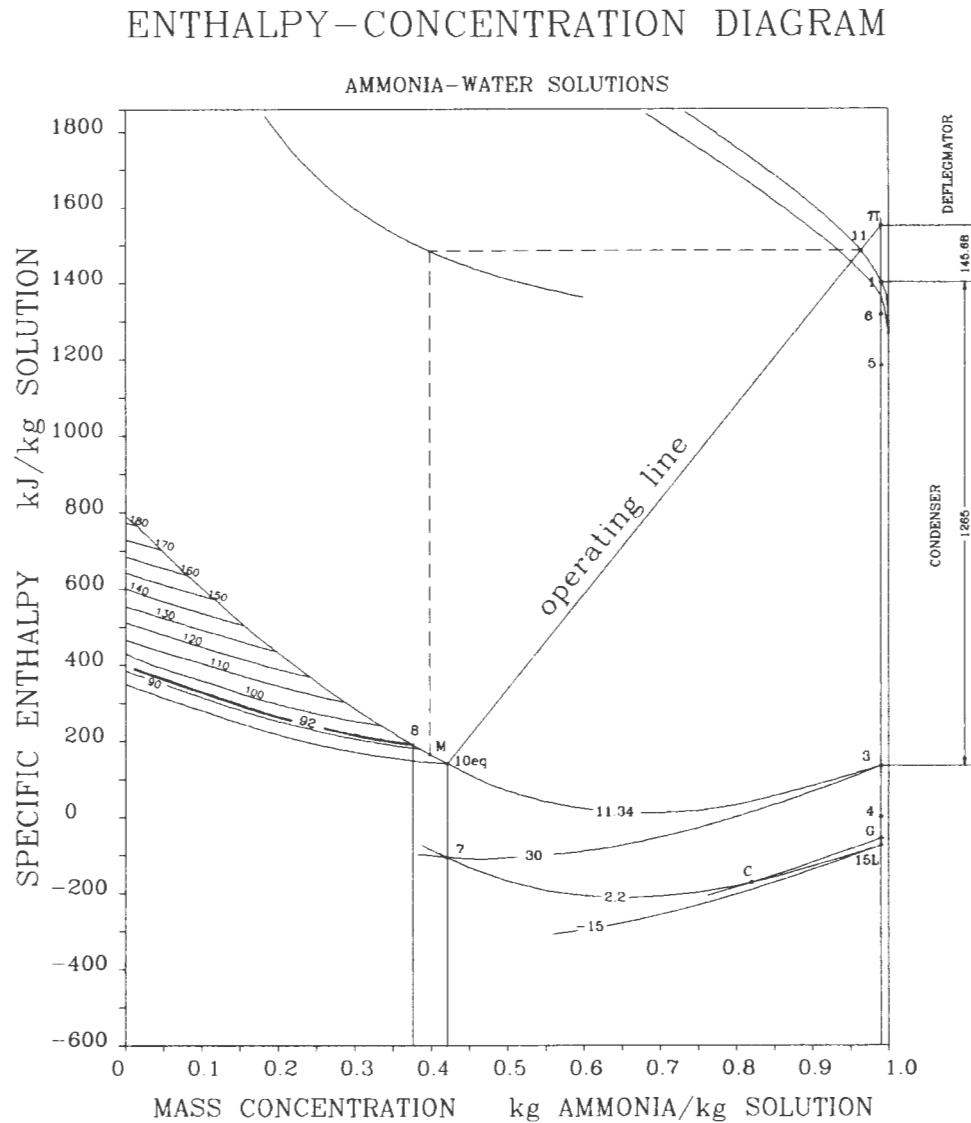


Figure 3. 23

The temperature contours above T_{10eq} mark the range of possible generator working temperatures which will produce the same refrigeration effect and heat rejection from the condenser.

Step-5 Optimisation of the generator's temperature

From Figure 3.5, the design sink and evaporator temperatures $T_5=30^\circ\text{C}$ and $T_{15}=-15^\circ\text{C}$ respectively, give the optimum weak solution temperature in the generator $T_8^{opt}=92^\circ\text{C}$. This temperature curve is highlighted in Figure 3.23.

Step-6 Determining the weak solution concentration

The weak solution concentration is obtained by drawing a vertical line from the intersection of the high pressure line and the weak solution temperature T_8^{opt} , to the base.

Step-7 Determining the deflegmator's heat effects and the Pole position π

Point M on the high pressure line is the average concentration of the bulk solution in the generator and is at equilibrium with its vapour at point 11, Figure 3.23. The extension of the operating section-line 10_{eq}-11 of the distillation column intersects the 0.990 concentration line at the pole position π . The heat $(h_\pi - h_1) = 145.66 \text{ kJ/kg}$ is rejected from the deflegmator necessary to produce vapour at concentration 0.990, at point 1.

The ratio of the enthalpy differences $(h_\pi - h_1)/(h_1 - h_3)$ is the reflux ratio.

Step-8 Determining the circulation rates of the solutions and the heat effects of the pump, heat-exchanger, absorber and generator

The extensions of the lines 8-10_{eq} and 8-7 meet the 0.990 concentration line at points F and D respectively, Figure 3.24. The work required by the pump W_P is calculated according to eq. (3.11) and the polynomials (3.15) and (3.16) in section 3.4.4. This value is estimated as $W_P = 1.38 \text{ kJ}$ per kg of strong solution. Thus point 13 could be located above point 7, such that $(h_{13} - h_7) = 1.37 \text{ kJ/kg}$. Because this is such a small quantity, it is difficult to locate point 13 directly.

Since the line 8-D represents the ratio of the mass rates, the lever rule is applied to locate point L on the 0.990 concentration line, such that the ratio $(8-7):(8-D) = (7-13):(D-L) = 1:f$.

This ratio establishes the mass flow rate of the strong solution $f = 13.76 \text{ kg/kg}$ of refrigerant. The work required by the pump is then $(f) \cdot W_P = 19 \text{ kJ/kg}$ of refrigerant and this locates point L above point D. The line 8-L intersects the strong solution concentration line at point 13.

Point 9 is located at the intersection of the extension of the F-13 line and the weak solution concentration line. The weak solution gives up heat which is measured along its concentration line as the difference $(h_8 - h_9)$ per kg of weak solution. This heat is gained by the strong solution

and is measured along its concentration line as the difference $(h_{10eq}-h_{13})$ per kg of strong solution.

The heat difference $(h_{10eq}-h_{13})$ per kg of strong solution is equivalent to the enthalpy difference $(h_F-h_L)=3367.74\text{kJ/kg}$ of refrigerant. The heat required by the generator is the enthalpy difference $(h_\pi-h_F)=2029.47\text{kJ/kg}$ and the heat rejected from the absorber is the enthalpy difference $(h_6-h_K)=1818.76\text{kJ/kg}$.

It can be observed from Figure 3.24 and the above analysis, that the absorber and the heat exchanger are the only two components to be affected if the work of the pump is not included. The error involved in the heat exchanger in this case will be 0.56% and for the absorber 1.04%

Step-9 Determining the COP

The coefficient of performance of the above optimised cycle is calculated as the ratio of the refrigeration effect (measured in step-2) and the heat required by the generator (measured in step-8). This ratio is 0.582 and it verifies the predicted maximum coefficient of performance ($\text{COP}_{\max} = 0.586$) obtained by the optimisation curves of Figure 3.5

3.3.3 Comments on the optimisation procedure

Figures 3.24 and 3.25 show the final steps towards the optimisation of the plant. However, it can be observed that a large amount of heat is exchanged in the heat exchanger.

In an attempt to justify the above, let it be assumed that points 8 and 10_{eq} coincide, in which case points 9 and 13 will coincide too. In the diagram of Figure 3.24, this will reflect to the line 8-D being parallel to the concentration line 0.990. This will translate to an “infinite” mass flow rate through the heat exchanger and of course an “infinite” heat exchanged between the solutions. Apart from the heat exchanger becoming 100% efficient, the heat supplied to the generator (Q_G+Q_{EX}) will only maintain a hot solution circulation through the heat exchanger without any vapour being produced. This condition refers to the “cut-off” point or the

“minimum” temperature T_8 depicted in the performance curves of Figures 3.3 (real cycle) and 3.4.

Because of the almost vertical rise in the coefficient of performance at temperatures close to $T_8 = T_{10eq}$, the mass and heat rates in the heat exchanger are still excessively high, which renders its size much larger than any other component in the plant. For this purpose a limit has to be imposed to the relative size of the heat exchanger.

In the production of data for the optimisation curves of Figure 3.5, the heat exchanger's heat is limited to twice the heat required by the generator. Hence, all the generator's temperatures below the one corresponding to this limit are omitted. Of course this limitation does not apply when the full trend of performances is required. In the above example, the heat exchanger's heat is about 1.66 times the heat required by the generator (see Figure 3.24).

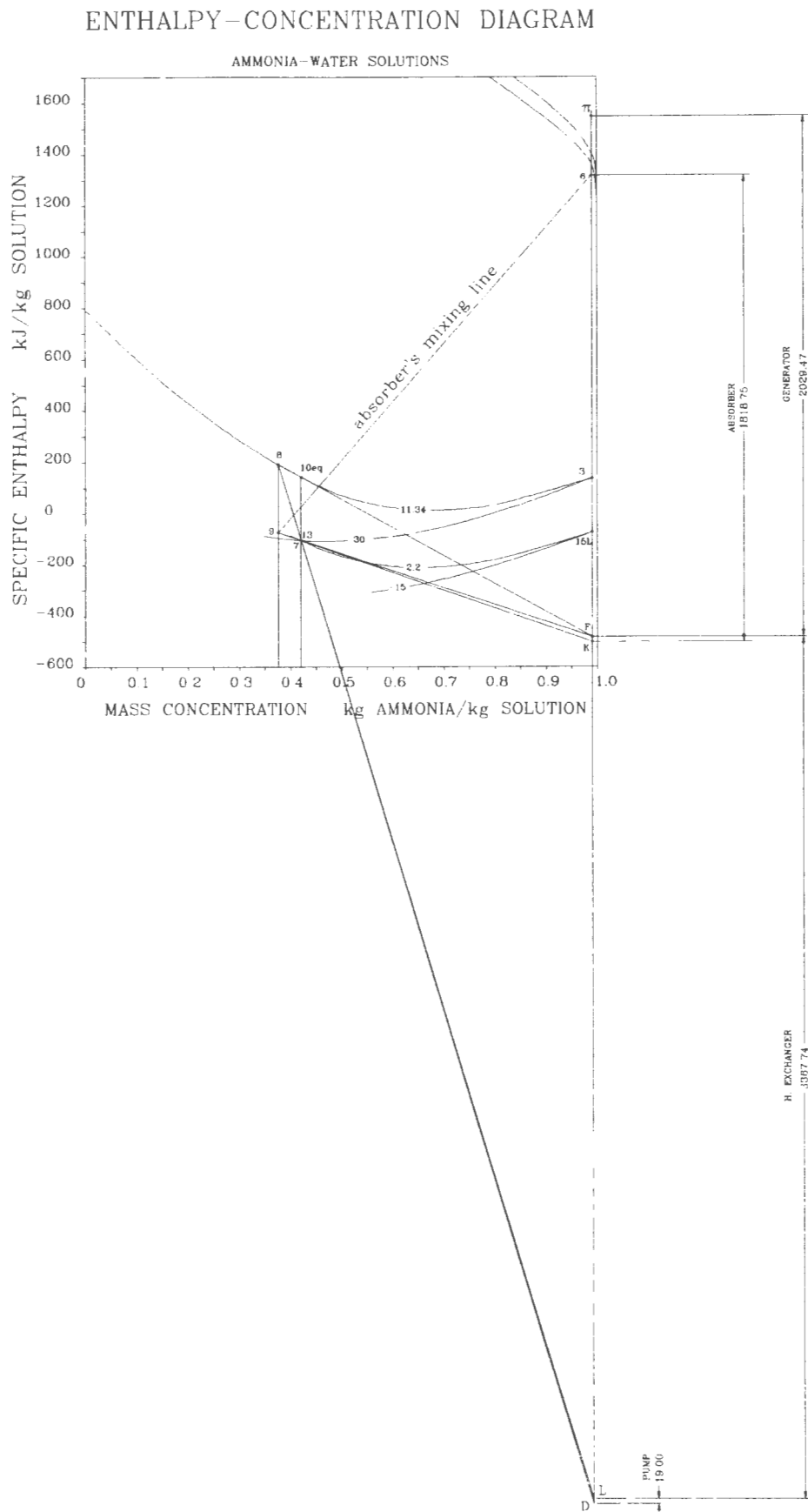


Figure 3. 24

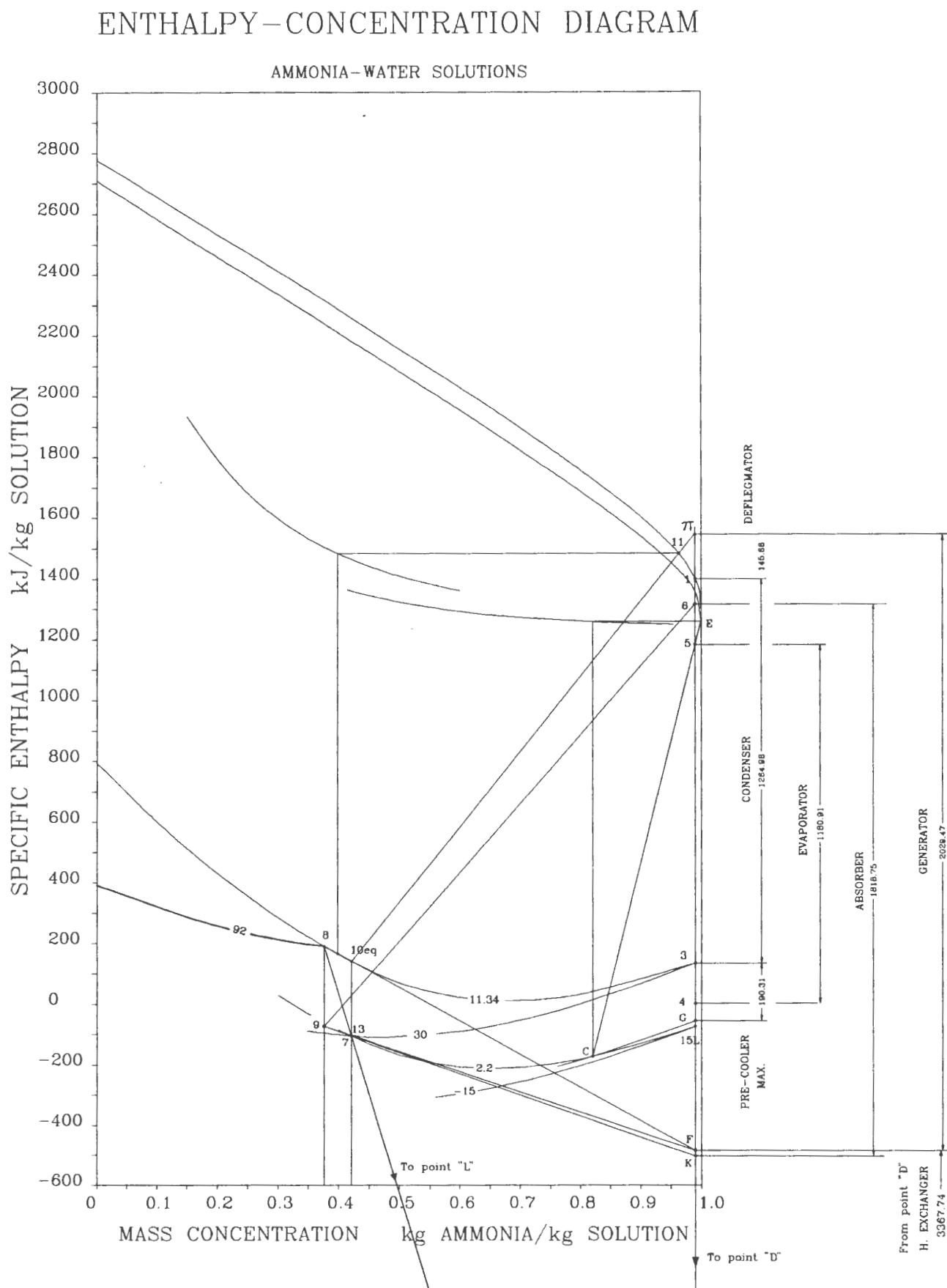


Figure 3. 25 The complete graphical presentation of the heat effects of the optimised absorption cycle

3.4 DESCRIPTION OF THE SIMULATION MODEL

The proposed method designs and optimises an absorption refrigeration plant, in terms of a known refrigeration requirement (evaporator coil temperature, and cooling capacity) and an environmental cooling temperature.

As discussed in section 3.2, the graphical analysis enables one to understand how the performance of the absorption refrigeration machine is affected by variations in the temperatures of the generator, evaporator, and cooling environment.

The proposed simulation model is written for a typical absorption unit as shown in Figure 3.1, and is based on the step-by-step graphical analysis of the “CASE STUDY” presented in the previous section. The equations used for the model are empirical functions, developed by the author, for the heat effects of the working fluid and for the Pressure-Temperature-Concentration (P-T-x) relationship.

The assumptions that were considered for the purpose of the calculations, are as follows:

- Heat and pressure losses in the piping between components are neglected.
- The mixture in the streams “1”, “2”, “3”, “7” and “10” is saturated.
- The absorber and the evaporator have the same pressure.
- The generator and the condenser have the same pressure.
- The concentration of the refrigerant is known (usually 0.990 to 0.995) [K8].
- All calculations are performed per kg of refrigerant mixture.

The parameters required by the simulation model are:

- T_E Average evaporator temperature
- R.C. Refrigeration capacity
- T_S Environmental temperature
- x_{15} Concentration of the refrigerant at the beginning of the evaporator tubes.
- T_8 Generator's temperature (or T_8^{opt} , if optimisation of the plant is required)

For the purpose of demonstrating the simulation model the experimental plant will be designed in a step-by-step procedure, using relevant equations for temperatures, pressures, concentrations and heat effects per kg of refrigerant for the following conditions:

- T_E $-15^{\circ}\text{C} < T_E < -12^{\circ}\text{C}$
- T_S 25°C
- x_3 0.990
- T_g 100°C

Then the experimental plant will be sized for 1.0 kW refrigeration capacity.

This chapter will close with the comparison of the optimum cycle characteristics.

3.4.1 Pressure evaluation

The saturation pressure of the mixture is determined by the P-T-x relation which is the key to solving the equilibrium properties of any binary saturated mixture. The graphical representation of the P-T-x relation is based on the data published by Scatchard et al. [S1] and is shown in APPENDIX A.

Equations have been developed by the author to predict the equilibrium vapour pressure of the water-ammonia mixture at a given temperature and concentration. These equations are presented in APPENDIX A.

The two main pressure domains are specified by the saturated pressures of the refrigerant-absorbent mixture at the condenser's and the evaporator's temperatures.

3.4.1.1 Low Pressure

The proposed simulation procedure starts with the concentration of the refrigerant mixture and the evaluation of the low pressure in the evaporator. As it is discussed earlier in this chapter the temperature of the mixture in the evaporator is not constant, but changes as the mixture

receives heat at constant pressure and evaporates [V3]. Therefore, T_{15} is the temperature of the refrigerant mixture at the beginning of the coil, immediately after the expansion valve.

<i>Known parameters</i>	<i>Unknown parameters</i>
coil inlet temperature - T_{15}	low pressure - P_{LOW}
concentration of mixture at coil's inlet - x_{15}	

The low pressure is calculated according to the P_{LOW} - T_{15} - x_{15} relationship.

CALCULATIONS

With $T_{15}=258.15$ K and $x_{15}=0.990$, the P-T-x relation from APPENDIX A:

eq.(a1) gives: **$P_{LOW}=2.198$ bars** {1}

concentration line x_3' . The enthalpy h_A is evaluated at a fictitious saturated pressure P_A , using a P_A - T_5 - x_3' relationship.

It is upon the designer's decision to predetermine the effectiveness of the pre-cooler. An acceptable value for a vapour-liquid heat exchanger is $\mathcal{E}=0.7$. Therefore point "4" can be located on the x_3' concentration line, such that $h_4=h_3'-(h_3'-h_A) \cdot \mathcal{E}$

The concentration x_3' and the enthalpy h_4 are calculated within an iterative loop until the concentration x_3' converges.

As a by-product of this iterative calculation, the saturated high pressure P_{HIGH} of the solution with concentration x_3 is also estimated and updated.

CALCULATIONS

Applying eq. (3.13), the following values are calculated:

With $P_{\text{LOW}}=2.198$ bars and $x_{15}=0.990$

$$\begin{aligned} \text{eq. (b1) gives:} & \quad h_{15x} = -77.89 \text{ kJ/kg} \\ \text{eq. (b3) gives:} & \quad h_{15y} = 1243.42 \text{ kJ/kg} \end{aligned}$$

With $T_{15}=258.15$ K and $x_{15}=0.990$

$$\text{eq. (c1) gives:} \quad y_{15} = 0.9999992$$

$T_5 = -12^\circ\text{C} = 261.15$ K (assume 3°C increase in refrigerant temperature inside the evaporator) and with the assumption that $x_3'=0.990$,

$$\text{eq. (a1) gives:} \quad P_A = 2.5016 \text{ bars}$$

With $P_A=2.5016$ and $x_3'=0.990$

$$\text{eq. (b1) gives:} \quad h_A = -63.964 \text{ kJ/kg}$$

With $T_3=25^\circ\text{C}=298.15$ K and $x_3'=0.990$,

$$\text{eq. (a1) gives:} \quad P_{\text{HIGH}} = 9.718 \text{ bars}$$

With $P_{\text{HIGH}}=9.718$ and $x_3'=0.990$

$$\text{eq. (b1) gives:} \quad h_3' = 109.767 \text{ kJ/kg}$$

$$Q_{PRmax} = h_3' - h_A = 173.73 \text{ kJ/kg}$$

With the assumption that the pre-cooler's effectiveness is $\mathcal{E}=0.7$,

$$Q_{PR} = 0.7 \times 173.73 = 121.612 \text{ kJ/kg}$$

$$h_4 = h_3' - Q_{PR} = -11.844 \text{ kJ/kg}$$

Then

$$\text{eq. (3.13) gives: } x_3' = 0.99049$$

After a multiple iteration the end values are:

$$P_{HIGH} = 9.731 \quad \{2\}$$

$$h_3 = 110.11 \text{ kJ/kg} \quad \{3\}$$

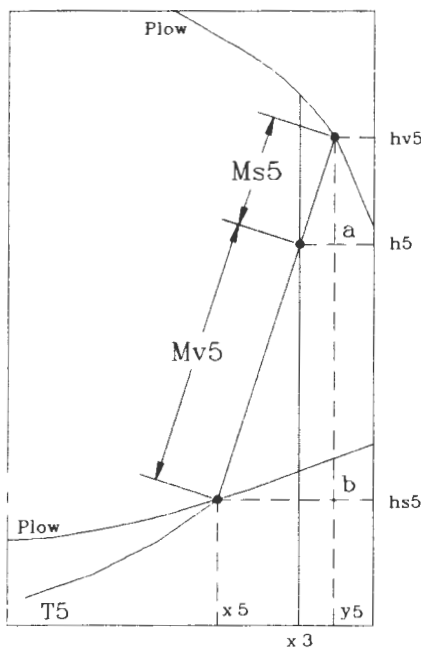
$$h_4 = -11.44 \text{ kJ/kg} \quad \{4\}$$

$$x_3 = 0.99050 \quad \{5\}$$

$$Q_{PR} = 121.549 \text{ kJ/kg}$$

3.4.2 Determination of the evaporator coil and pre-cooler conditions

<i>Known parameters</i>	<i>Unknown parameters</i>
total mass of refrigerant - M_5	total mass of saturated solution - M_{S5}
total concentration of refrigerant x_3	total mass of saturated vapour - M_{V5}
coil exit temperature - T_5	enthalpy of saturated solution - h_{S5}
low pressure - P_{LOW}	enthalpy of saturated vapour - h_{V5}



The mixture at point 5 is a wet vapour. The total enthalpy h_5 will be constituted by the proportions of saturated vapour and saturated liquid, Figure 3.28.

The enthalpy of the vapour at the exit of the pre-cooler is evaluated using the heat balance equation $h_6 = h_3 + h_5 - h_4$. The temperature T_6 of the vapour however, is calculated iteratively. It is adjusted until a concentration-pressure combination $x_{S6} \cdot P_{LOW}$ will give the total enthalpy of the mixture h_6 .

Figure 3. 28

CALCULATIONS

With $T_5 = 261.15$ K and $P_{LOW} = 2.198$ bars and using Simpson's rule on eq (a1) to evaluate x_5 ,

$$\text{eq. (a1) gives:} \quad x_5 = 0.81886$$

With $x_5 = 0.81886$ and $T_5 = 261.15$ K,

$$\text{eq.(c1) gives:} \quad y_5 = 0.999968$$

A check whether the mixture is a superheated or a wet vapour is based on Figure 3.28 and is as follows:

$$m_{s5} = \frac{y_5 - x_3}{y_5 - x_3} \cdot m_3 = 0.052235 \text{ kg/s}$$

where m_{s5} is mass of the entrained liquid in the mixture and it must be positive for a wet vapour condition.

Then the mass of the vapour is: $m_{v5} = m_5 - m_{s5} = 1 - m_{s5} = 0.94776 \text{ kg/s}$

With $P_{\text{LOW}} = 2.198 \text{ bars}$ and $y_5 = 0.999968$

eq. (b3) gives: $h_{v5} = 1243.86 \text{ kJ/kg}$

With $P_{\text{LOW}} = 2.198 \text{ bars}$ and $x_5 = 0.81886$

eq. (b1) gives: $h_{s5} = -180.470 \text{ kJ/kg}$

The total enthalpy at point 5 is:

$$h_5 = m_{s5} \cdot h_{s5} + m_{v5} \cdot h_{v5}$$

$$\mathbf{h_5 = 1169.461 \text{ kJ/kg} \quad \{6\}}$$

The total enthalpy at point 6 is:

$$\mathbf{h_6 = h_3 + h_5 - h_4 = 1291.01 \text{ kJ/kg} \quad \{7\}}$$

And the refrigeration effect is given by:

$$\mathbf{R.E. = h_5 - h_4 = 1180.9 \text{ kJ/kg} \quad \{8\}}$$

The temperature at the exit of the pre-cooler is found iteratively as follows:

Assume $T_6 = T_5 = 261.15 \text{ K}$ and $x_6 = x_5 = 0.81886$

With $x_6 = 0.81886$ and $P_{\text{LOW}} = 2.198 \text{ bars}$,

eq. (b1) gives: $h_{x6} = -180.470 \text{ kJ/kg}$

With $x_6 = 0.81886$ and $P_{\text{LOW}} = 2.198 \text{ bars}$ and using Simpson's rule on eq. (a1) to evaluate T_6 ,

eq. (a1) gives: $T_6 = 261.15 \text{ K}$

With $x_6 = 0.81886$ and $T_6 = 261.15 \text{ K}$

$$\text{eq.(c1) gives: } y_6=0.9999682$$

With $y_6=0.9999682$ and $P_{\text{LOW}}=2.198$ bars,

$$\text{eq.(b3) gives: } h_{v6}=1243.861 \text{ kJ/kg}$$

Similarly a check whether the mixture is a superheated or a wet vapour, is as follows:

$$m_{s6} = \frac{y_6 - x_3}{y_6 - x_6} \cdot m_3 = 0.052263 \text{ kg/s}$$

$$\text{Then the mass of the vapour is: } m_{v6}=m_6 - m_{s6}=1 - m_{s6}=0.94774 \text{ kg/s}$$

$$\text{A test total enthalpy at point 6 is: } h_{6\text{test}}= m_{s6} \cdot h_{s6} + m_{v6} \cdot h_{v6}=1169.42 \text{ kJ/kg}$$

The concentration x_6 is adjusted until $h_{6\text{test}}=h_6$ from eq. {7}

After a multiple iteration the end values are:

$$T_6=296.14 \text{ K}$$

$$h_{6\text{test}}=1291.006 \text{ kJ/kg}$$

3.4.3 Absorber

The operating conditions in the absorber are such that a strong solution is produced at temperature T_7 and pressure P_{LOW} ¹.

With reference to the PhD thesis by Keizer [K5], an absorber could be constructed to deliver a saturated solution at a pressure lower than the evaporator's pressure.

Therefore, in the proposed simulation model the concentration x^{ST} of the strong solution can be calculated under the saturated conditions of temperature T_7 and pressure P_{LOW} .

CALCULATIONS

With $P_{LOW} = 2.198$ bars and $T_7 = 298.15$ K, a Simpson's rule iteration on eq. (a1) evaluates the concentration of the strong solution x^{ST}

$$\text{eq. (a1) gives: } x^{ST} = 0.433 \quad \{9\}$$

With $x^{ST} = 0.433$ and $P_{LOW} = 2.198$ bars,

$$\text{eq. (b1) gives: } h_7 = -124.90 \text{ kJ/kg} \quad \{10\}$$

¹ With reference to APPENDIX III, the pressure drop in the evaporator tubes due to friction and change of momentum is small. When calculations are performed per kg of refrigerant, this pressure drop is neglected. However, when sizing of a plant and designing of the components is required, a pressure drop ΔP_E is subtracted from the low pressure P_{LOW} . The pressure ($P_{LOW} - \Delta P_E$) in the absorber will determine the concentration of the strong solution.

3.4.4 Pump

Once the strong solution passes through the pump to the high pressure side, it behaves as a sub-cooled solution. Therefore the properties of the fluid at point 13, cannot be found by equilibrium relationships.

The work of the pump W_P is proportional to the specific volume of the solution v_7 and is calculated as:

$$(3.14) \quad W_P = M_7 \frac{P_{high} - P_{low}}{\eta_P} \cdot v_7$$

The specific volume of the solution is given by:

$$v_7 = x_7 \cdot v_7^a + (1 - x_7) \cdot v_7^w$$

where

v_7^a is the specific volume of the saturated liquid ammonia

v_7^w is the specific volume of the saturated water

The specific volumes of ammonia and water are given by the following polynomials as suggested by [H5] and [K1] respectively.

$$(3.15) \quad v_7^a = v_{(0)} + v_{(1)} \cdot T + v_{(2)} \cdot T^2 + v_{(3)} \cdot T^3 + v_{(4)} \cdot T^4$$

$$(3.16) \quad v_7^w = a + b \cdot (c - t)^d - e \cdot [(P_{low} - \Delta P_E) - f] \cdot (g - t)^{-h}$$

where

$[v_7^a] = \text{m}^3/\text{kg}$, $[v_7^w] = \text{cm}^3/\text{g}$, $[P] = \text{int Atm}$

The coefficients of the above equations are given in Table 3.2

Table 3.2			
Saturated liquid			
Coeff.	Ammonia	Coeff	Water
$v_{(0)}$	2.941×10^{-3}	a	3.086
$v_{(1)}$	-2.840×10^{-5}	b	0.899
$v_{(2)}$	1.774×10^{-7}	c	374.100
$v_{(3)}$	-5.000×10^{-10}	d	0.147
$v_{(4)}$	4.955×10^{-13}	e	0.400
		f	218.500
		g	385.000
		h	1.600

The enthalpy of the solution at point 13 is the sum of the solution at point 7 and the work done by the pump:

$$h_{13} = h_7 + W_p$$

It is assumed that the temperature increase is insignificant. Therefore $T_{13}=T_7$.

CALCULATIONS

The mass rate of the strong solution is calculated according to eq. (3.6).

Thus

$$m_7 = 5.332 \text{ kg/s} \quad \{11\}$$

With $T_7 = 298.15 \text{ K}$ and $P_{\text{Low}} = 2.198/1.013 = 2.1697 \text{ Atm}$

$$\text{eq. (3.15) gives: } v_7^a = 9.059 \times 10^{-4} \text{ m}^3/\text{kg}$$

$$\text{eq. (3.16) gives: } v_7^w = 9.6487 \times 10^{-4} \text{ m}^3/\text{kg}$$

The total specific volume of the mixture is:

$$v_7 = x^{ST} \cdot v_7^a + (1 - x^{ST}) \cdot v_7^w = 8.805 \times 10^{-4} \text{ m}^3/\text{kg}$$

The work by the pump is given by eq. (3.14).

Thus

$$W_p = 0.961 \text{ kJ/kg} \quad \{12\}$$

The heat rate of steam 13 is:

$$Q_{13} = (h_7 + W_p) \cdot m_7 = -660.80 \text{ kJ/kg} \quad \{13\}$$

3.4.5 Solution heat exchanger

With reference to section 3.2.6, the strong solution in stream 10 is saturated at P_{high} . Therefore the temperature T_{10} is calculated by a $P_{\text{HIGH}}-T_{10\text{eq}}-x_{10\text{eq}}$ relation.

The heat transferred from the weak to the strong solution is expressed in terms of the properties of the weak solution:

$$(3.17) \quad Q_{EX} = \overline{Cp}^{WE} \cdot M_8 \cdot (T_8 - T_9) = \overline{Cp}^{ST} \cdot M_7 \cdot (T_{10} - T_{13})$$

where

$$\overline{Cp}^{WE} = x^{WE} \cdot \overline{Cp}^a + (1 - x^{WE}) \cdot \overline{Cp}^w$$

and

$$(3.18) \quad \overline{Cp}^a = a + b \cdot T_b + c \cdot T_b^2 + d \cdot T_b^3 + e \cdot T_b^4$$

$$(3.19) \quad \overline{Cp}^w = f + g \cdot T_b + h \cdot T_b^2$$

for the temperature range

$$225 < \left[T_b = \frac{T_8 + T_9}{2} \right] < 320$$

Table 3.3			
Cp for the pure component			
Coeff.	Ammonia	Coeff	Water
a	19.577576	f	5.508798
b	-0.25297	g	8.24476×10^{-3}
c	1.536×10^{-3}	h	1.27576×10^{-5}
d	4.1×10^{-9}		
e	4.955×10^{-13}		

The coefficients for the above relations are given in Table 3.3

CALCULATIONS

With $P_{\text{high}}=9.731$ bars and $x^{\text{ST}}=0.4328$, a Simpson's rule iteration on eq. (a1) evaluates the saturation temperature of the strong solution at point 10.

$$\text{eq. (a1) gives:} \quad T_{10\text{eq}} = 347.04 \text{ K} \quad \{14\}$$

With $x^{ST}=0.4328$ and P_{high} bars,

$$\text{eq. (b1) gives: } h_{10eq} = 99.58 \text{ kJ/kg} \quad \{15\}$$

The heat exchanged within the solution is:

$$Q_{EX} = h_{10eq} \cdot m_7 - Q_{13} = 1191.71 \text{ kJ/kg} \quad \{16\}$$

This heat must be equal to the quantity expressed by eq. (3.17):

Assuming $T_9=T_7$, $T_8=373.15$ K the bulk temperature of the weak solution is given by:

$$T_b = (T_8 + T_9)/2 = 335.65 \text{ K}$$

$$\text{eq. (3.18) gives: } C_p^a = 4.9303 \text{ kJ/kg K}$$

$$\text{eq. (3.19) gives: } C_p^w = 4.1812 \text{ kJ/kg K}$$

and

$$\text{eq. (3.17) gives: } Q_{EXtest} = 1427.37 \text{ kJ/kg}$$

Iterations between Q_{EX} and Q_{EXtest} converge giving:

$$T_9 = 310.75 \text{ K}$$

3.4.6 Generator

Earlier in this chapter, it was shown graphically the way by which the absorption refrigeration cycle is optimised to a particular generator's temperature T_8^{opt} .

This optimum temperature is calculated by eq. (3.2) and (3.4) and it is set as the generator's operating temperature unless a different temperature T_8 is desired. Then the concentration x^{WE} of the weak solution is determined by a $P_{HIGH}-T_8-x^{WE}$ relationship, APPENDIX A.

The bulk solution in the generator is a mixture of the incoming strong solution, the returned liquid from the distillation column and the outgoing weak solution. The concentration of this mixture depends on the construction of the generator. For example if the mixture remains for a long time in the generator, its concentration will be close to the weak solution. In this simulation procedure, it is assumed that the concentration of the bulk solution is the average

between the strong and weak solutions: $x_b = \frac{x^{WE} + x^{ST}}{2}$

This solution is saturated and its temperature T_b is calculated by the $P_{HIGH}-T_b-x_b$ relationship.

The vapour in equilibrium with this bulk liquid-mixture has a concentration y_{11} given by:

$$y_{11} = \frac{\alpha_{(T)} \cdot x_b}{1 + (\alpha_{(T)} - 1) \cdot x_b}$$

where

$\alpha_{(T)}$ = the relative volatility between water and ammonia at the prevailing temperature

The values of $\alpha_{(T)}$ are taken from the publication by Scatchard et al. [S1] and have been modelled into Fourier equations for the whole range of concentrations and temperatures. The equations and coefficients of the T-x-y routine are described in APPENDIX C.

CALCULATIONS

The minimum weak solution temperature is when $T_8 = T_{10eq} = 347.11$ K

The design is for a temperature **$T_8 = 100^\circ\text{C} = 373.15$ K** **{17}**

With $T_8 = 373.15$ K and $P_{HIGH} = 9.731$ bars, a Simpson's rule iteration on eq. (a1) evaluates the saturation concentration of the weak solution at point 8.

$$\text{eq. (a1) gives: } x^{\text{WE}}=0.3044 \quad \{18\}$$

With $x^{\text{WE}}=0.3044$ and $P_{\text{HIGH}}=9.731$ bars,

$$\text{eq. (b1) gives: } h_8=246.88 \text{ kJ/kg} \quad \{19\}$$

The mass rate of the weak solution is given by (f-1) where f is the mass rate of the strong solution given by eq (3.6).

$$\text{Thus } m_8=4.3315 \text{ kg/s} \quad \{20\}$$

The generator's bulk solution concentration is $x_b=0.3686$

$$\text{and } x_{12}=0.3686 \quad \{21\}$$

With $x_{12}=x_b=0.3686$ and $P_{\text{HIGH}}=9.731$ bars,

$$\text{eq. (b1) gives: } h_{12}=165.72 \text{ kJ/kg} \quad \{22\}$$

With $x_b=0.3686$ and $P_{\text{HIGH}}=9.731$ bars, a Simpson's rule iteration on eq. (a1) evaluates the saturation temperature of the weak solution at point 11.

$$\text{eq. (a1) gives: } T_{11}=359.79 \text{ K}$$

With $T_{11}=359.79 \text{ K}$ and $x_b=0.3686$,

$$\text{eq. (c1) gives: } y_{11}=0.9602 \quad \{23\}$$

The assumption that $x_{12}=x^{\text{ST}}$ leads to $T_{12}=T_{11}$

With $y_{11}=0.9601$ and $P_{\text{HIGH}}=9.731$ bars.

$$\text{eq. (b3) gives: } h_{11}=1484.90 \text{ kJ/kg} \quad \{24\}$$

3.4.7 Distillation Column

The equations used for the simulation of the distillation column are presented in section 3.2.2.5

CALCULATIONS

The vapour concentration of stream 1 is the same as the concentration of the condensate of stream 3, Thus:

$$y_1 = x_3 = 0.99051 \quad \{25\}$$

The concentration of the liquid at point 1 is calculated iteratively.

Assume a liquid concentration $x_1 = 0.5$

With $x_1 = 0.5$ and $P_{\text{HIGH}} = 9.731$, a Simpson's rule iteration on eq. (a1) evaluates the saturation temperature of the liquid in equilibrium with the vapour at point 1.

$$\text{eq. (a1) gives:} \quad T_1 = 331.063 \text{ K}$$

With $x_1 = 0.5$ and $T_1 = 331.063 \text{ K}$,

$$\text{eq. (c1) gives:} \quad y_{1\text{test}} = 0.99374$$

The concentration of x_1 is adjusted iteratively until $y_1 = y_{1\text{test}} = 0.99051$

The iterations converge, giving the following results:

$$x_1 = 0.491$$

$$T_1 = 336.497 \text{ K}$$

With $T_1 = 336.497 \text{ K}$ and $y_1 = 0.99051$

$$\text{eq. (b3) gives:} \quad h_1 = 1395.16 \text{ kJ/kg} \quad \{26\}$$

The heat rejected from the distillation column is given by eq (3.10). Thus:

$$\text{eq. (3.10) gives:} \quad Q_D/m_1 = 57.31 \text{ kJ/kg} \quad \{27\}$$

$$\text{eq. (3.12) gives:} \quad \text{Reflux} = m_R = 0.122 \text{ kg/s} \quad \{28\}$$

3.4.8. Mass and heat flow rates

The heat effects are calculated per kg/s of refrigerant. Therefore the mass flow rates and total heat within each component are determined as follows:

- the mass rate of the mixture at the streams 3, 5, 6, 4 and 15 is set equal to $M_1=1\text{kg/s}$
- the mass rate of the vapour from the generator to the distillation column, stream 11:

$$M_{11} = M_1 \frac{y_1 - x_{12}}{y_{11} - x_{12}}$$

- the mass rate of the returning liquid returning to the generator from the distillation column. stream 12:

$$M_{12} = M_1 \frac{y_1 - y_{11}}{y_{11} - x_{12}}$$

- the mass rate of the weak solution from the generator to the absorber, stream 8:

$$M_8 = M_1 \frac{y_1 - x^{ST}}{x^{ST} - x^{WE}}$$

- the mass rate of the strong solution to the generator from the absorber, stream 7:

$$M_7 = M_1 + M_8$$

- the mass rate of the mixture at the streams 13 and 10 is set equal to M_7
- the mass of mixture at the stream 9 is set equal to M_8

The enthalpies of the mixtures at the numerals are evaluated either through a saturated mixture equation or through a pseudo-temperature or pseudo-pressure procedure. Then the total heat carried by the fluid at a particular “j” point is $Q_j = h_j \cdot M_j$

The heat balance within each component is as follows:

- Generator (Q_{in}) : $Q_8 + Q_{11} - Q_{12} - Q_{10}$
- Absorber: $Q_6 + Q_9 - Q_7$
- Condenser: $Q_1 - Q_3$

- Pump: W_P eq. (3.14)
- Deflegmator Q_D eq. (3.10)
- Refrigeration effect (R.E.): $Q_5 - Q_4$

and the coefficient of performance is:

$$COP = \frac{Q_5 - Q_4}{Q_8 + Q_{11} - Q_{12} - Q_{10} + W_P} = \frac{R.E.}{Q_{in} + W_P}$$

The simulation program is shown in a flow-chart form in Figure 3.29.

CALCULATIONS

The mass fluxes are as follows:

$$m_1 = 1.0 \text{ kg/s} \quad \{29\}$$

$$\text{From } \{21\}, \{23\}, \{25\}, \{29\} \quad m_{12} = m_1 \cdot \frac{y_1 - y_{11}}{y_{11} - x_{12}} = 0.0512 \text{ kg/s} \quad \{30\}$$

$$\text{From } \{21\}, \{23\}, \{25\}, \{29\} \quad m_{11} = m_1 \cdot \frac{y_1 - x_{12}}{y_{11} - x_{12}} = 1.0512 \text{ kg/s} \quad \{31\}$$

The heat fluxes are as follows:

$$\text{From } \{25\}, \{26\} \quad Q_1 = 1395.16 \text{ kW} \quad \{32\}$$

$$\text{From } \{3\}, \{29\} \quad Q_3 = 110.11 \text{ kW} \quad \{33\}$$

$$\text{From } \{7\}, \{29\} \quad Q_6 = 1291.01 \text{ kW} \quad \{34\}$$

$$\text{From } \{10\}, \{11\} \quad Q_7 = -665.92 \text{ kW} \quad \{35\}$$

$$\text{From } \{18\}, \{19\} \quad Q_8 = 1069.40 \text{ kW} \quad \{36\}$$

$$\text{From } \{16\}, \{36\} \quad Q_9 = -122.31 \text{ kW} \quad \{37\}$$

$$\text{From } \{11\}, \{15\} \quad Q_{10} = 530.91 \text{ kW} \quad \{38\}$$

$$\text{From } \{23\}, \{31\} \quad Q_{11} = 1570.96 \text{ kW} \quad \{39\}$$

$$\text{From } \{21\}, \{30\} \quad Q_{12} = 8.489 \text{ kW} \quad \{40\}$$

Therefore the heat effects on the components are as follows:

From {36}, {38}, {39}, {40}	$Q_{in}=2091.0\text{kW}$
From {34}, {35}, {36}	$Q_{ABS}=1834.6\text{kW}$
From {32}, {33}	$Q_{CON}=1285.05\text{ kW}$
From {37}	$Q_{DIST}=157.3\text{ kW}$
From {12}	$W_P=5.136\text{ kW}$

The refrigeration effect and the COP of the plant are:

From {8}	$R.E.=1180.9\text{ kW}$
From {8}, {12}, {36}, {38}, {39}, {40}	$COP=0.563$

The complete simulated performance of the plant is presented in Tables 3.4 to 3.7

Table 3.4						
Position	Pressure	Temperature	Mass flow rate	Sp. Heat	Heat	Concentration
	Bar	°C	kg/s	kJ/kg	kW	
1	9.73	63.4	1.0	1395.2	1395.2	0.9905
R*	9.73	25.0	0.122	110.1	13.5	0.9905
3	9.73	25.0	1.0	110.1	110.1	0.9905
4	9.73	-0.7	1.0	-11.4	-11.4	0.9905
5	2.20	-12.0	1.0	1169.5	1169.5	0.9905
6	2.20	23.0	1.0	1291.0	1291.0	0.9905
7	2.20	25.0	5.332	-124.9	-665.85	0.433
8	9.73	100.0	4.332	246.9	1069.36	0.305
9	9.73	37.7	4.332		-122.3	0.305
10	9.73	73.9	5.33	99.91	530.9	0.433
11	9.73	86.6	1.051	1485.0	1561.0	0.960
12	9.73	86.6	0.051	165.7	8.5	0.369
15	2.20	-15.0	1.0	-11.4		0.990

Table 3.5		
Refrigeration capacity	kW	1180.9
Input heat to generator	kW	2091.0
Work done by the pump	kW	5.136
Heat rejected from the condenser	kW	1285.05
Heat rejected from the deflegmator	kW	157.3
Heat rejected from the absorber	kW	1834.6
Refrigeration Effect	kJ/kg	1180.66
Coefficient of performance		0.563

R* indicates the position of the reflux in the deflegmator

The experimental plant is sized according to the heat effects presented in Tables 3.6 and 3.7

Table 3.6						
Position	Pressure	Temperature	Mass flow rate	Sp. Heat	Heat	Concentration
	Bar	°C	kg/s x 10³	kJ/kg	kW	
1	9.73	63.4	0.847	1395.2	1.182	0.9905
R	9.73	25.0	0.104	110.1	0.0123	0.9905
3	9.73	25.0	0.847	110.1	0.093	0.9905
4	9.73	-0.7	0.847	-11.4	-0.0096	0.9905
5	2.20	-12.0	0.847	1169.5	0.991	0.9905
6	2.20	23.0	0.847	1291.0	1.093	0.9905
7	2.20	25.0	4.515	-124.9	-0.564	0.433
8	9.73	100.0	3.668	246.9	0.906	0.305
9	9.73	37.7	3.668		-0.103	0.305
10	9.73	73.9	4.515	99.91	0.451	0.433
11	9.73	86.6	0.890	1485.0	1.329	0.960
12	9.73	86.6	0.0434	165.7	0.0072	0.369
15	2.20	-15.0	0.847	-11.4		0.990

Table 3.7		
Refrigeration capacity	kW	1.000
Input heat to generator	kW	1.771
Work done by the pump	kW	0.004
Heat rejected from the condenser	kW	1.088
Heat rejected from the deflegmator	kW	0.133
Heat rejected from the absorber	kW	1.554
Refrigeration Effect	kJ/kg	1180.9
Coefficient of performance		0.563

With the use of the optimisation curves of Figure 3.5 the $T_{15} = -15^{\circ}\text{C}$, $T_s = 25^{\circ}\text{C}$ pair gives $T_8^{\text{opt}} = 80.44^{\circ}\text{C}$

The maximum performance plant would have the working parameters as indicated in Tables 3.8 and 3.9

Table 3.8						
Position	Pressure	Temperature	Mass flow rate	Sp. Heat	Heat	Concentration
	bar	$^{\circ}\text{C}$	kg/s	kJ/kg	kW	
1	9.73	63.4	1.0	1395.16	1395.16	0.99050
R	9.73	25.0	0.061	110.11	6.87	0.99050
3	9.73	25.0	1.0	110.11	110.11	0.99050
4	9.73	-0.75	1.0	-11.44	-11.44	0.99050
5	2.2	-12.0	1.0	1169.46	1169.46	0.99050
6	2.2	23.0	1.0	1291.01	1291.01	0.99050
7	2.2	25.0	17.566	-124.90	-2194.01	0.433
8	9.73	80.4	16.566	130.93	2168.91	0.400
9	9.73	26.6	16.566		-1757.38	0.400
10	9.73	73.9	17.566	99.58	1749.16	0.433
11	9.73	77.1	10.254	1440.24	1477.96	0.976
12	9.73	77.1	25.4	115.1	2.9	0.433
15	2.2	-15.0	1.0	-11.41		0.99

Table 3.9		
Refrigeration capacity	kW	1180.9
Input heat to generator	kW	1893.7
Work done by the pump	kW	16.9
Heat rejected from the condenser	kW	1285.0
Heat rejected from the deflegmator	kW	78.7
Heat rejected from the absorber	kW	1727.6
Refrigeration Effect	kJ/kg	1180.9
Coefficient of performance		0.618

The maximum COP value predicted by the optimisation curves is $\text{COP}_{\text{max}} = 0.620$

FLOWCHART FOR THE COMPUTER MODEL

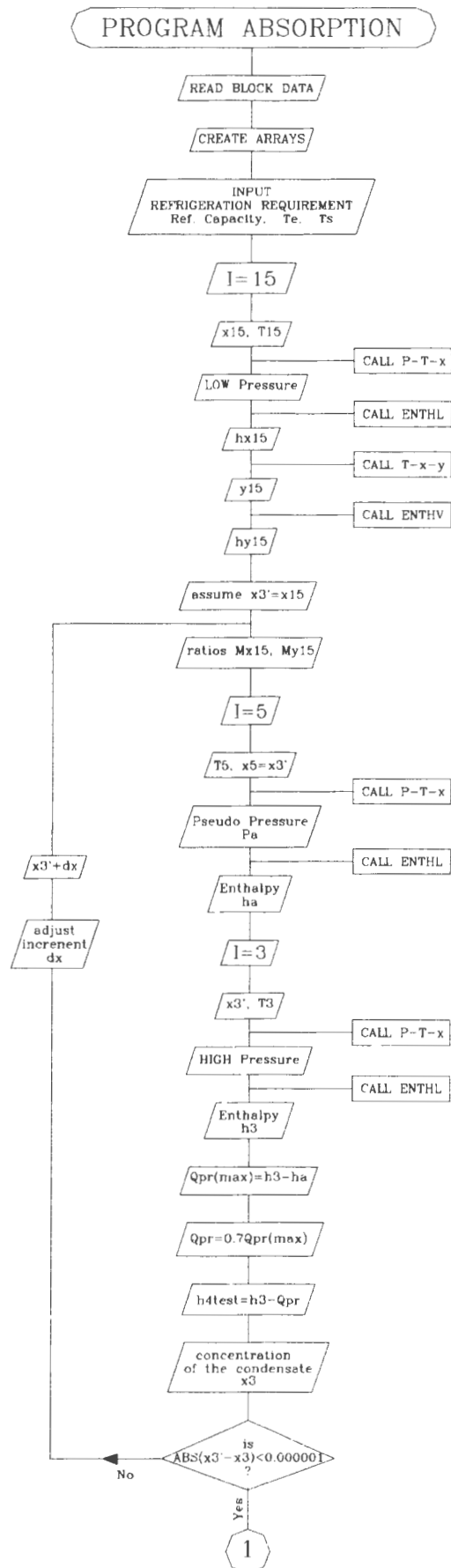


Figure 3. 29

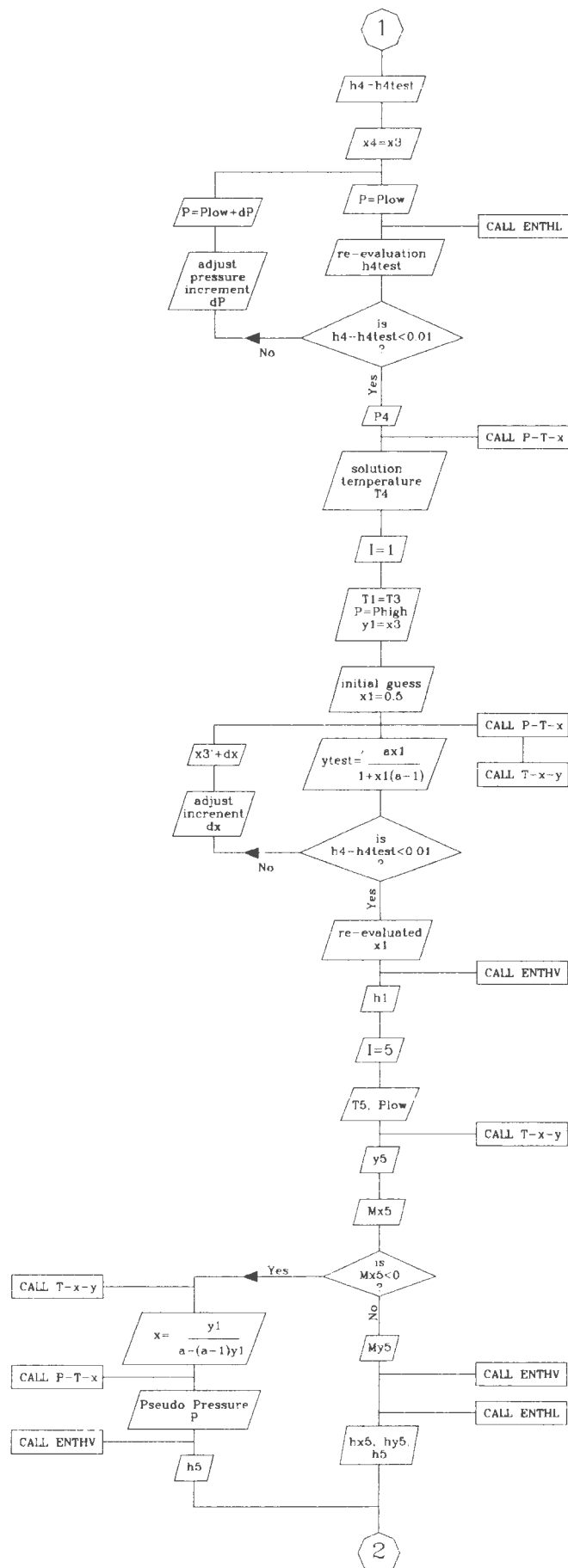


Figure 3. 29 (continued)

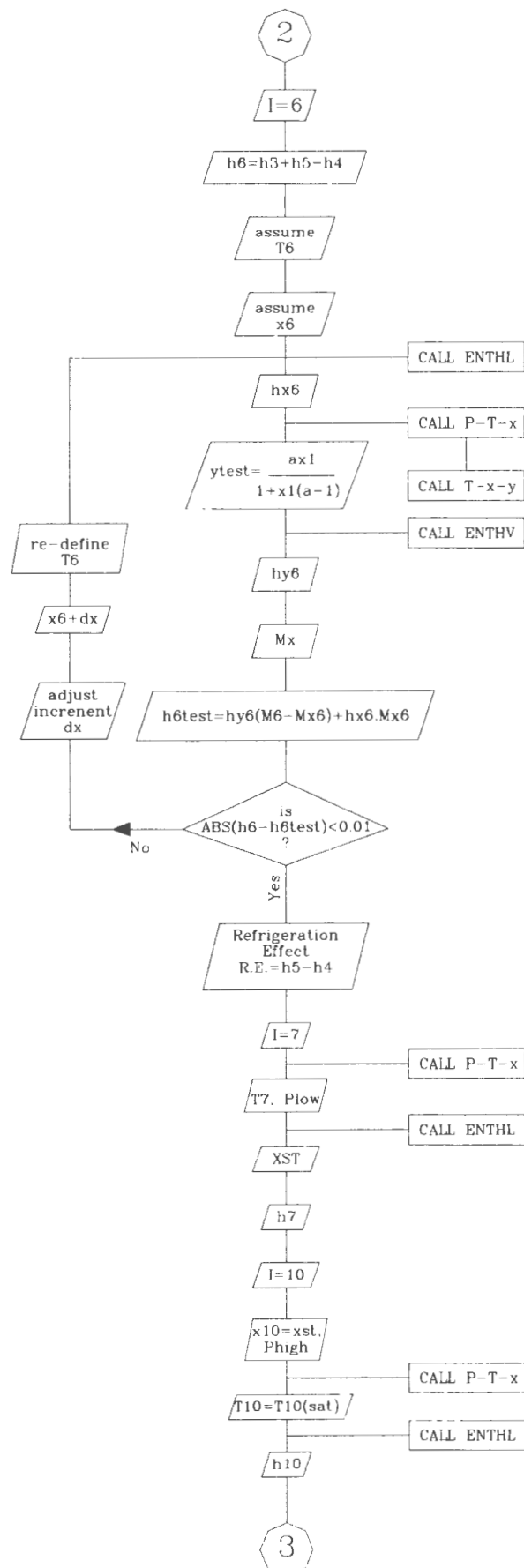


Figure 3. 29 (continued)

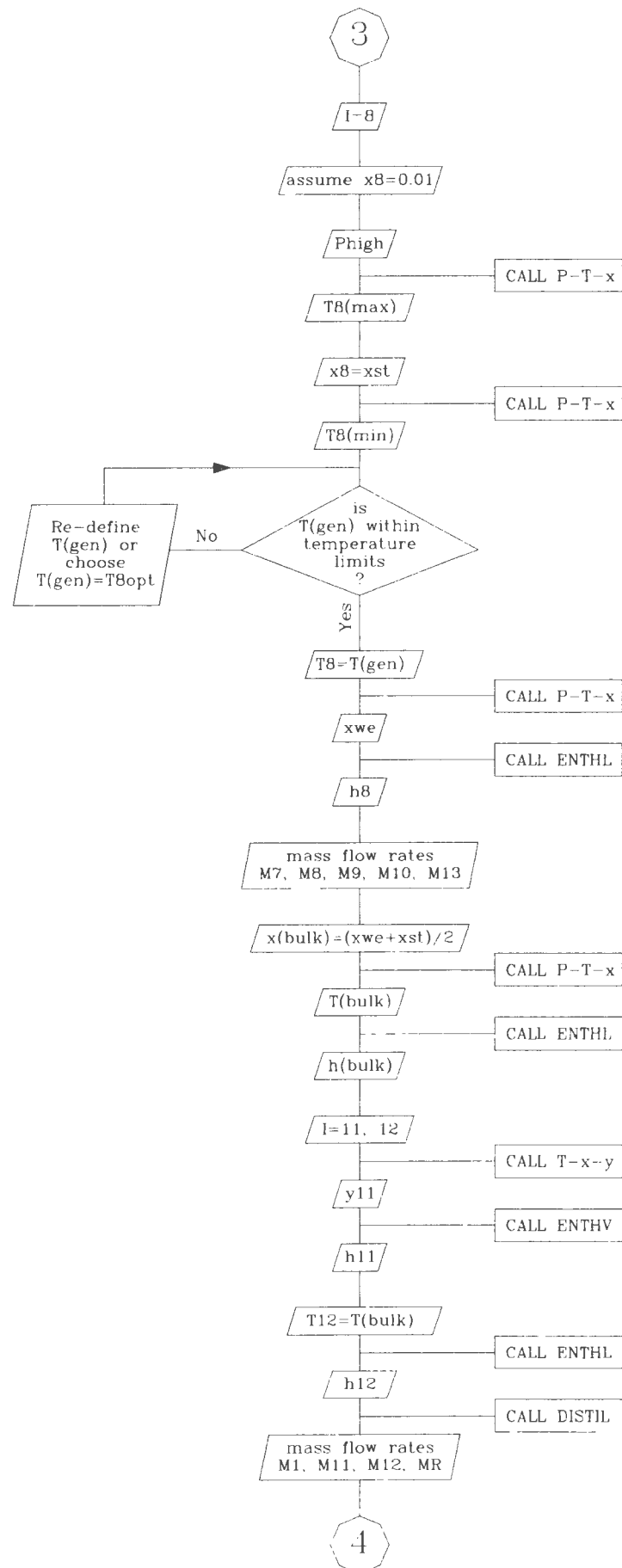


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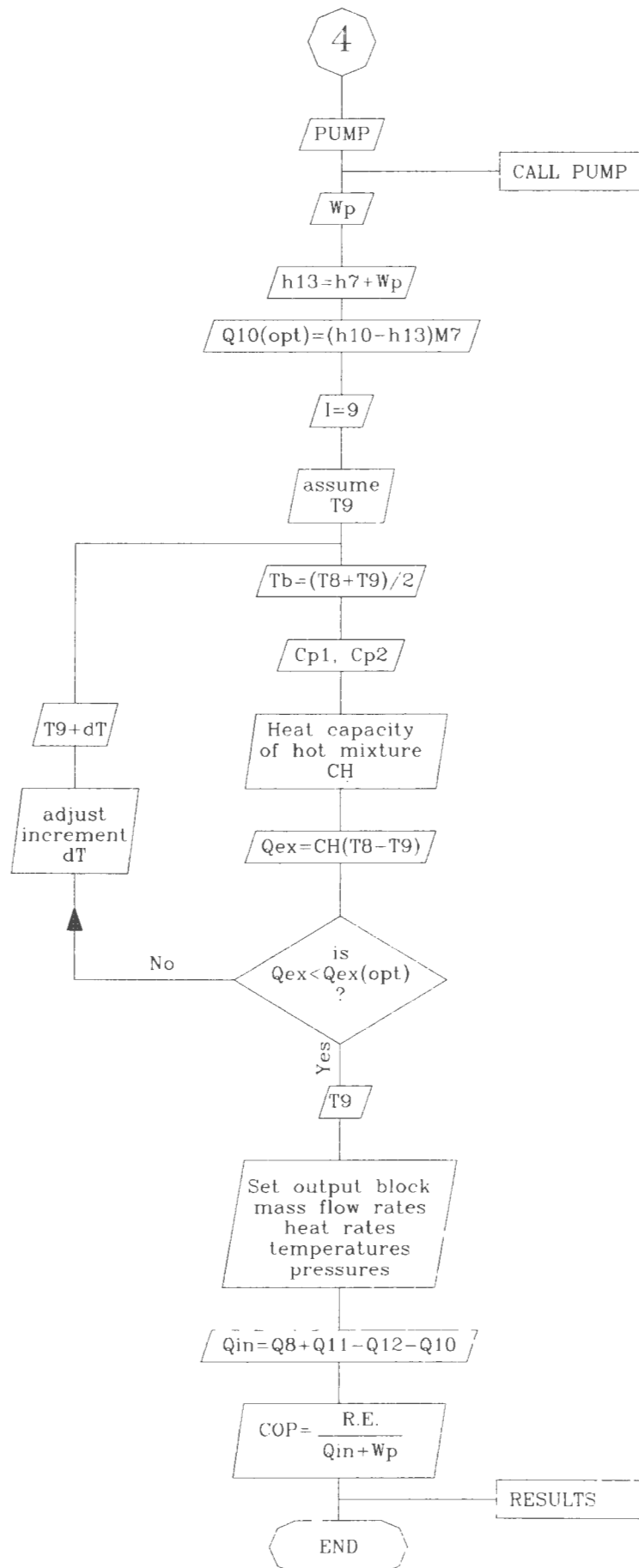


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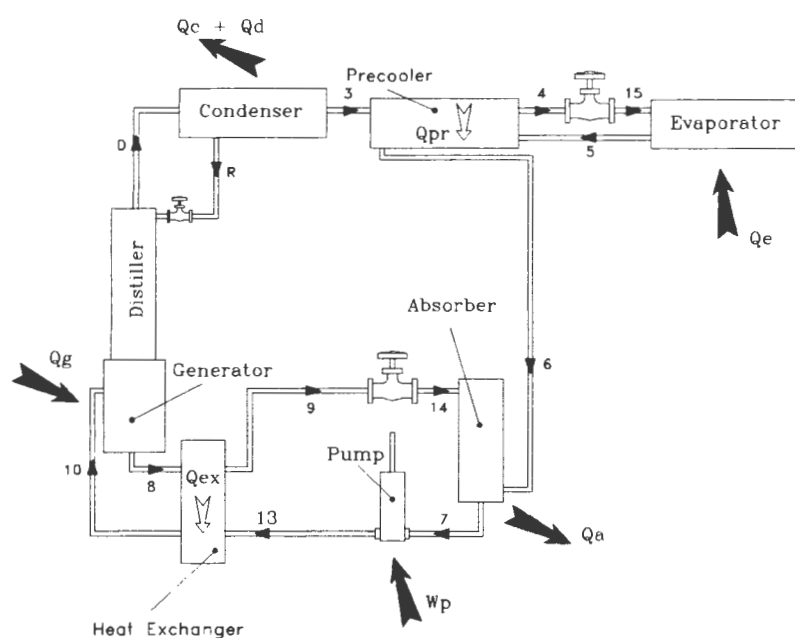
CHAPTER 4

DESIGN OF THE ABSORPTION REFRIGERATION EXPERIMENTAL PLANT

4.0 GENERAL

The purpose of building the experimental plant was to obtain sufficient data covering a large range of working conditions in order to verify the proposed simulation program.

The experimental unit is shown in a schematic diagram in Figure 4.1 with selected components shown in Figures 4.3 to 4.9.



The components were designed and sized to accommodate the relative mass and heat rates. These components are:

- condenser
- heat exchanger
- absorber
- evaporator/pre-cooler
- generator
- distillation column

Figure 4. 1 Schematic diagram of the experimental plant

The plant was designed for an average refrigeration capacity of 1kW at -15°C coil and 25°C condensate temperatures respectively. A test-run of the simulation program as demonstrated in Chapter 3 using the above requirements, produced the predicted working conditions which were used to design the components.

In this chapter the main results of the calculations as well as the design of the components are presented. The equations and calculations used for the sizing of each component are presented in their respective appendix.

4.1 COMPONENT DESIGN OF THE EXPERIMENTAL PLANT

4.1.1 Condenser-deflegmator combined design

The condenser is of the shell and tube type in which the refrigerant vapour is condensing over a bank of water-tubes.

The heat of the deflegmator is removed in the condenser. The modification steps to the design of the condenser are shown in Figure 4.2 [F1, T2].

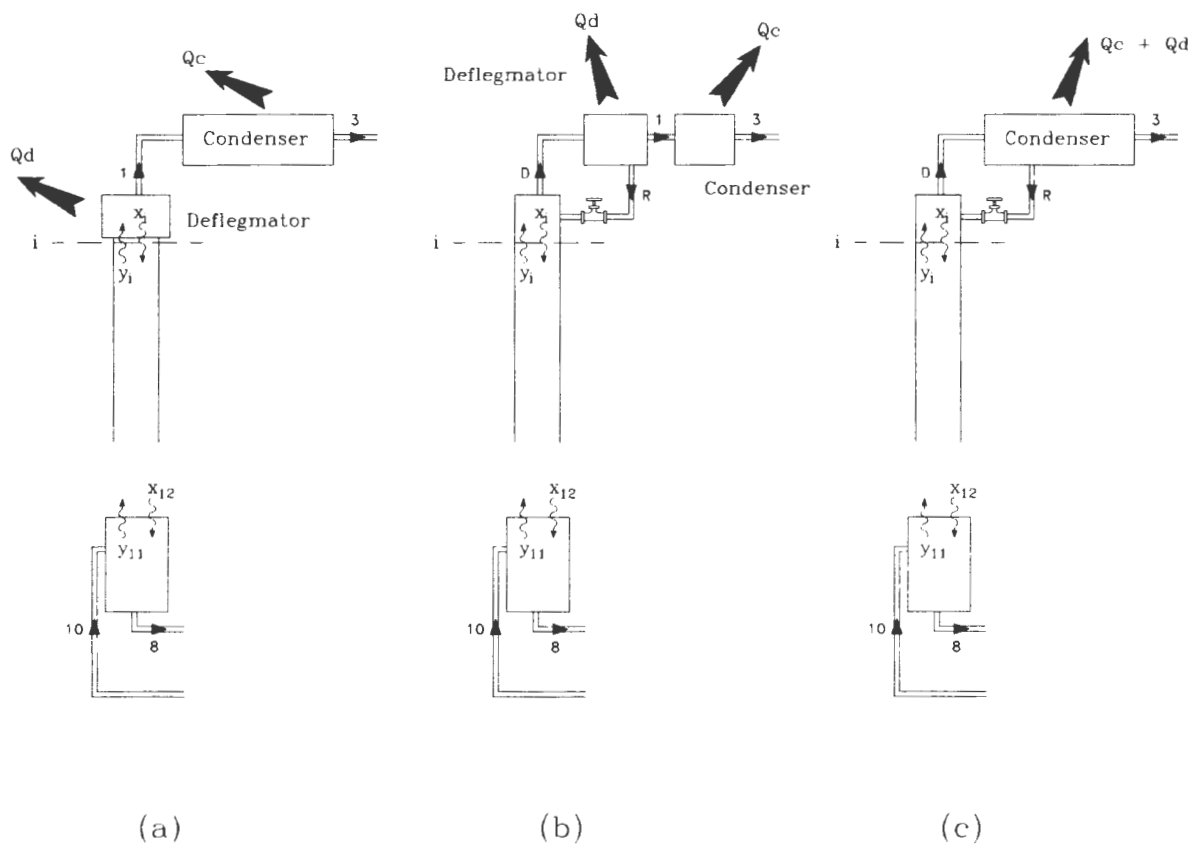


Figure 4.2 The steps taken for the condenser to include the heat of the deflegmator

Choice is made between the length and the number of water-tubes as well as tube diameters from standard size materials.

The detail design of the condenser is shown in Figure 4.3 and its characteristics in Table 4.1. The calculations are presented in APPENDIX E.

Table 4.1		
Condenser's load	W	1240
Saturation temperature	°C	25.0
Water inlet temperature	°C	18.0
Water outlet temperature	°C	19.8
Water velocity	m/s	1.2
Tube material		steel
Tube outside diameter	m	0.006
Tube wall thickness	m	0.0015
Shell tube inside diameter	m	0.050
Water-side fouling factor		0.0001
Refrigerant side fouling factor		0.00025
Water-side H.T.C.	W/m ² K	5284.7
Refrigerant-side H.T.C.	W/m ² K	14154.6
Overall H.T.C.	W/m ² K	1175
Cooling water quantity	lt/s	0.161
Number of tubes		19
Length of tubes	m	0.5

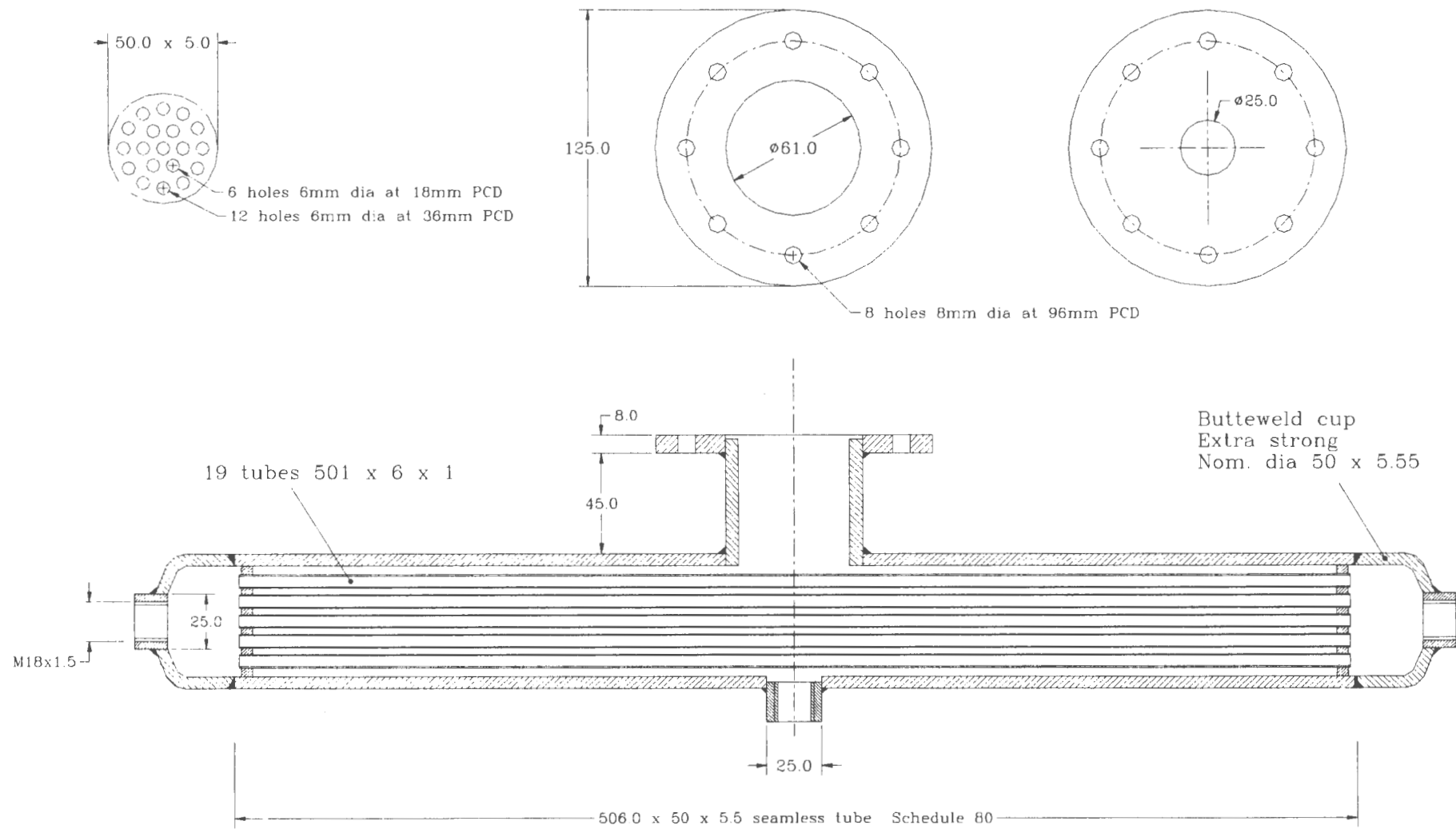


Figure 4.3 *Detail drawing of the condenser*

4.1.2 Solution heat exchanger

The heat exchanger is of a double-tube, counter-flow design. The inner tube contains the hot-weak solution and the annular gap contains the cold-strong solution.

The design characteristics of the heat exchanger are shown in Table 4.2 and the calculations are presented in APPENDIX F.

Table 4.2		
Heat exchanger's load	W	1016
Strong solution inlet temp.	°C	25
Strong solution outlet temp	°C	73.9
Weak solution inlet temp.	°C	100
Weak solution outlet temp	m/s	37.6
Tube materials		steel
Inner tube outside diameter	m	0.006
outer tube outside diameter	m	0.010
Inner tube wall thickness	m	0.001
outer tube wall thickness	m	0.001
Strong solution velocity	m/s	0.256
Weak solution velocity	m/s	0.347
Strong solution-side H.T.C.	W/m ² K	393.4
Weak solution-side H.T.C.	W/m ² K	2852.9
Overall H.T.C.	W/m ² K	406.3
Required external surface area of the inner tube	m ²	0.135
Mass rate of weak solution	kg/s	3.675×10 ⁻³
Length of tubes	m	7.16

4.1.3 Absorber

The absorber is of a vertical shell and multi-tubular core design. The absorption of gas is taking place in the tubular core, while cooling is achieved by passing water in the annular space.

According to Keizer [K5], either heat or mass transfer was controlling the absorption process when $U < 100$ or $U > 1000$ W/m²K respectively. These two limiting situations were observed when both the overall heat transfer and the volumetric mass transfer coefficients were varied between $100 < U < 3000$ W/m²K and $2.5 < K_L \alpha < 260$ kg/m³s through a computer model. Although similar results pointing to a simultaneous heat and mass transfer, were computed by the author

in APPENDIX G, it is felt that a fictitious variation of U and $K_L\alpha$ is neither practical nor feasible without seriously affecting the geometry of the absorber.

Present design calculations show that the height of tubes necessary for heat transfer is sufficient for mass transfer too. This is in agreement with the statements in paragraphs 2.7.2 and 2.8. The procedure for the design of the absorber is presented in APPENDIX G. The results of these computations are shown in Table 4.3. The detailed design of the tubular absorber is shown in Figure 4.3.

Table 4.3

Absorber's load	Q_{ABS} [W]	1835
Strong solution temperature	$^{\circ}\text{C}$	25
Mass rate of weak solution	m_8 [kg/s]	3.668×10^{-3}
Mass rate of refrigerant	m_6 [kg/s]	0.847×10^{-3}
Concentration of weak solution	x^{WE}	0.305
Concentration of strong solution	x^{ST}	0.433
Driving potential	$\Delta x = x^* - x^{WE}$	0.685
Inlet temperature of cooling water	$^{\circ}\text{C}$	18
Outlet temperature of cooling water	$^{\circ}\text{C}$	18.7
Cooling water quantity	lt/min	36.8
Inner diameter of tubes	m	0.170
Outer diameter of tubes	m	0.020
Number of tubes		7
Height of tubes	m	1.38
Inner diameter of shell tube	m	0.078
Wall thickness of shell tube	m	0.005
Solution-side H.T.P.	$\text{W/m}^2\text{K}$	296
Water-side H.T.P.	$\text{W/m}^2\text{K}$	1837
Overall H.T.P.	$\text{W/m}^2\text{K}$	258
Volumetric M.T.P.	$\text{kg/m}^3\text{s}$	8.92

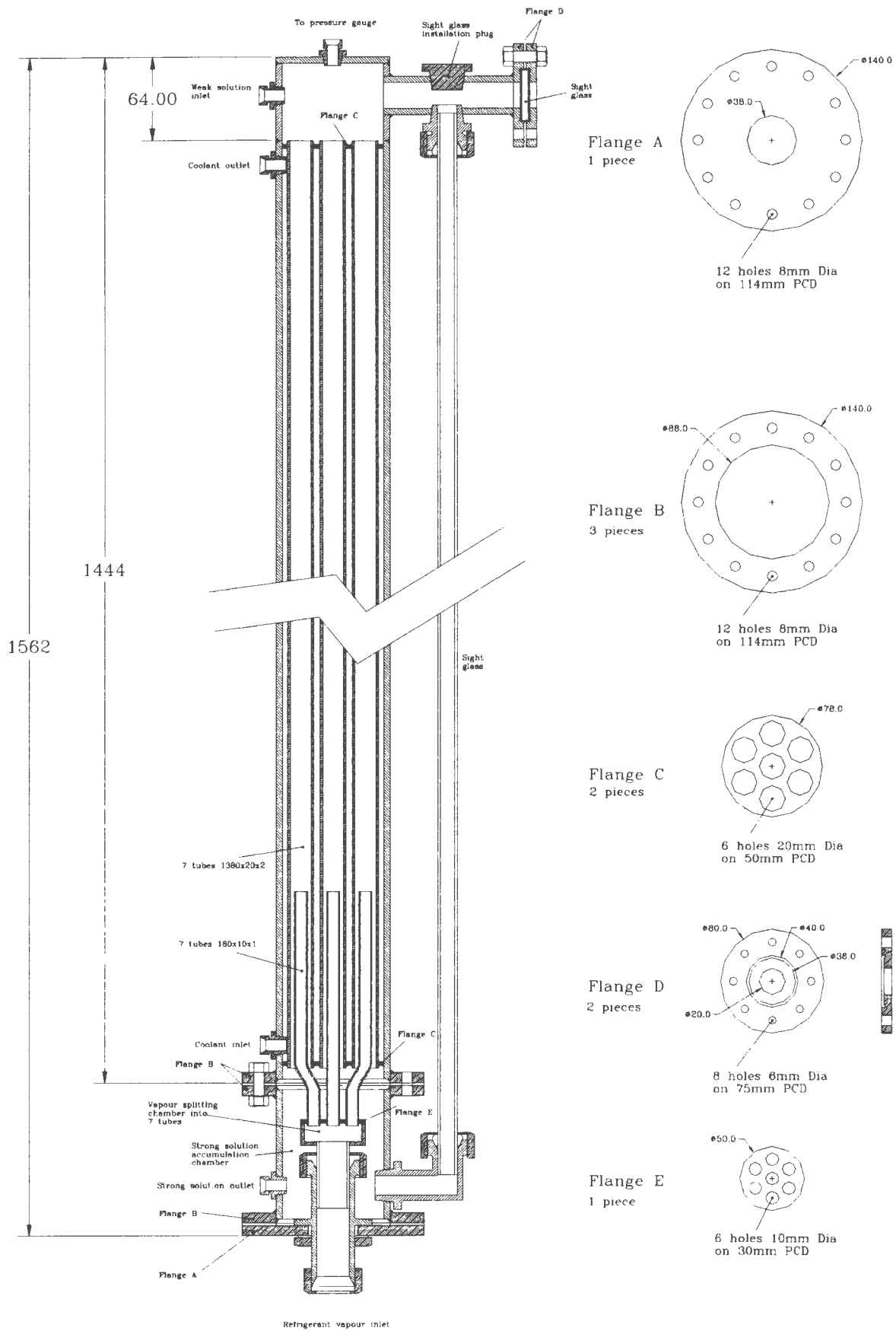


Figure 4. 4 *The vertical tubular absorber*

4.1.4 Evaporator and pre-cooler

4.1.4.1 Evaporator tubes

The evaporator is designed taking into account two-phase flow correlations. The design is for two concentric tubes in a horizontal arrangement. The refrigerant flows through the inner tube while water which serves as a load, flows through the annular space.

As the refrigerant boils in the tube, it produces vapour which initially travels at low speed relative to the liquid, but progressively increases, resulting in an annular two-phase flow. The geometry of the evaporator is such that annular flow is established when the vapour quality of the refrigerant is 60%. For a specified tube length, iteration procedures evaluate the diameter of the tube, so that the liquid and vapour momentum fluxes and friction pressure drops locate the flow in the annular region of the Taitel and Dukler [T1] two-phase flow map.

The frictional pressure drop is calculated based on a homogeneous model of two-phase flow that ignores details of the flow patterns as suggested by Mills [M6]. Frictional pressure drop correlations suggested by Martinelli et al [M5], for a viscous-turbulent liquid-vapour flow, give a pressure drop which is about half the value for the individual vapour pressure drop and about 180 times larger than the individual liquid pressure drop. However, due to the uncertainty of the accuracy of either method, the larger liquid-vapour combined frictional pressure drop is considered. The correlations by Mills [M6] are supported in these calculations.

Heat transfer correlations, suggested by Klimenko [M6], establish that the boiling is within the nucleate region and then calculate the refrigerant-side heat transfer coefficient. Correlations suggested by Gnielinski [M6] calculate the water-side heat transfer coefficient.

The procedure to design the evaporator is presented in APPENDIX H.

The results of the calculations are presented in Table 4.4 and the design of the evaporator is shown in Figure 4.5.

Table 4.4		
Evaporator's load	W	1000
Coil inlet temperature	°C	-15.0
Coil outlet temperature	°C	-12.0
Initial and average pressures	bars	2.2, 2.198
Total pressure drop in evaporator	Pa	271.75
Water inlet and outlet temperatures	°C	18.0 9.35
Water velocity	m/s	0.195
Water flow rate	lt/min	1.66
Refrigerant mass flow rate	g/s	0.847
Tube material		STEEL
Inner tube inside and outside diameters	m	0.009, 0.012
Outer tube inside and outside diameters	m	0.018, 0.020
Required surface area	m ²	0.0565
Length of tubes	m	2
Refrigerant side H.T.C.	W/m ² K	629532.4
Water side H.T.C.	W/m ² K	504.12
Overall H.T.C.	W/m ² K	657.16
Effectiveness		0.275

4.1.4.2 Pre-cooler

The pre-cooler is a counter-flow vapour-liquid heat exchanger.

Its effectiveness is predetermined by the simulation procedure and set to $\mathcal{E}=0.7$. This is necessary in order to establish the working conditions of the pre-cooler (paragraph 3.6.2), prior to the evaluation of the low pressure and the design concentration of the condensed refrigerant.

The pre-cooler is of a shell-and-tube design, and this has been established by a trial-and-error method between the length, diameter and number of tubes, and vapour frictional pressure drop. Through the tubes, vapour refrigerant flows in a counter-flow direction to the liquid refrigerant which is in the annular space.

The design of the pre-cooler is shown in Figure 4.5 linked to the evaporator tubes. The computations are presented in APPENDIX H and the results are presented in Table 4.5

Table 4.5		
Pre-cooler's load	W	102
Temperature of condensate	°C	25
Condensate temperature drop	°C	26.5
Vapour pressure at outlet	Bars	2.195
Pressure drop	Pa	188.98
Refrigerant tube material		Steel
Tube inside and outside diameters	m	0.003, 0.006
Shell-tube inside diameter	m	0.050
Required surface area	m ²	0.179
Number of tubes		19
Length of tubes	m	0.50
Vapour-side H.T.C.	W/m ² K	58.70
Liquid-side H.T.C.	W/m ² K	269.11
Overall H.T.C.	W/m ² K	25.79
Effectiveness		0.7
Dryness factor		0.95

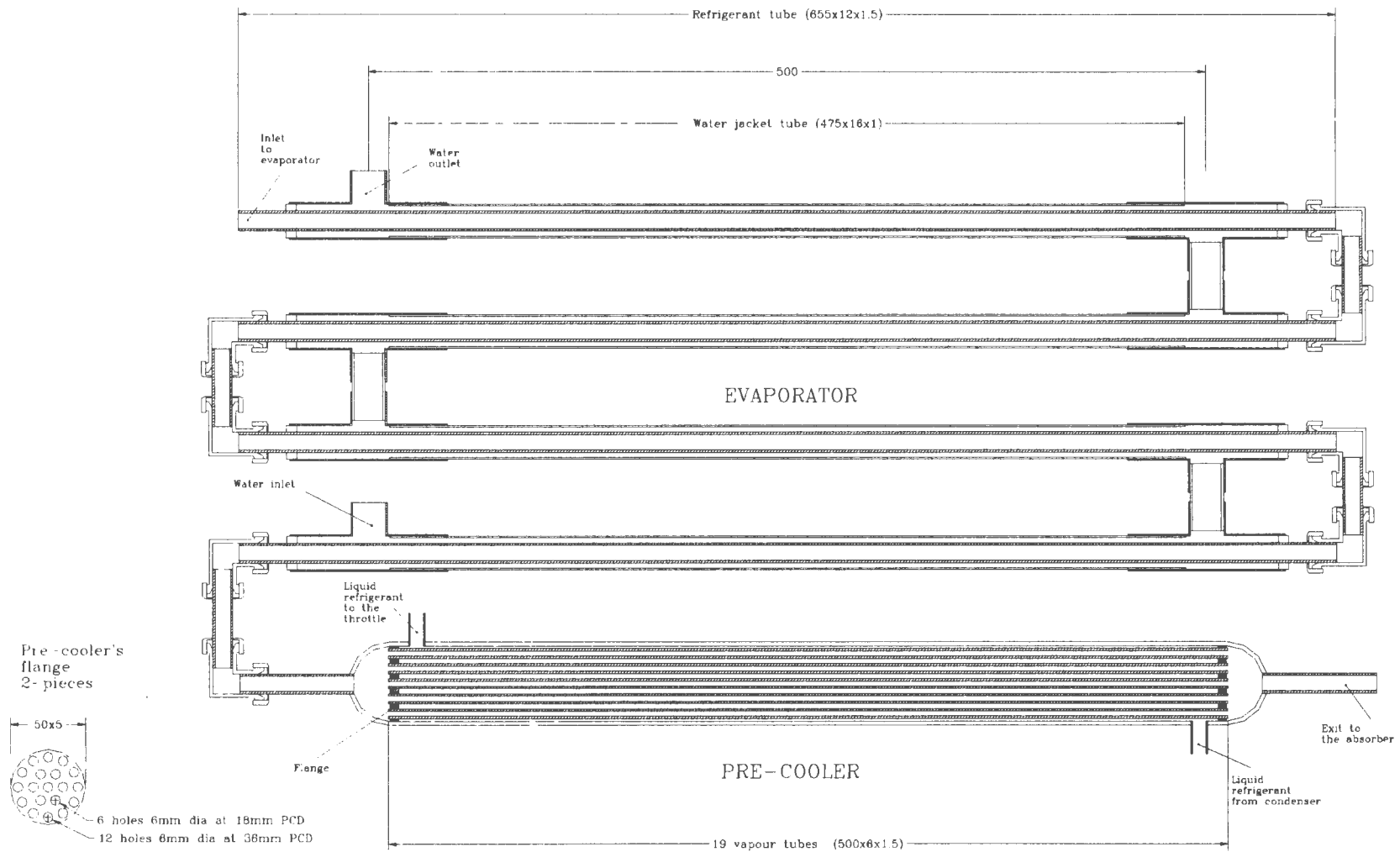


Figure 4. 5 *Evaporator and pre-cooler integrated design*

4.1.5 Generator

Boiling of binary mixtures is characterised by a close link between heat and mass transfer processes, with the evaporation rate being limited by the mass transfer process [C1]. Binary mixtures' pool boiling differs from pure single component boiling in the sense that their heat transfer rate is generally lower than that of an ideal pure liquid with the same physical properties as the real mixture. The boiling heat transfer coefficients for aqua mixtures are a strong function of the liquid concentration and heat flux and they drop rapidly with the addition of small amounts of the more volatile substance [Bib. 44]. For nucleate pool boiling the heat transfer coefficient is a function of the wall superheat, that is the difference between the wall temperature and the liquid boiling temperature [U1].

The most popular correlation for the prediction of the wall temperature is that of Stephan and Korner [U1], in which the actual mixture's superheat is related to the ideal mixture superheat as a function of the liquid and vapour compositions and overall pressure. With this correlation, Unal [U1] claims to have obtained an RMS error of 14.8% over 388 collected data of different binary mixtures.

The generator is designed for nucleate boiling of the ammonia-water binary mixture. It is of a shell and tube construction, in which the mixture is in the shell. The heat supplied to the mixture comes from hot-air passing through the tubes. Due to the large number of parameters involved in the design of the physical dimensions of the boiler, the number, length, diameter and thickness of the tubes are pre-determined.

The characteristics of the generator, Figures 4.6 and 4.7, are shown in the Table 4.6, supported by the equations and method demonstrated in APPENDIX I.

Table 4.6				
Generator's load	W	1000	2000	3000
Solution bulk temperature (sat)	°C	85.85		
Wall temperature	°C	146.26	193.51	241.02
Hot gases inlet temperature	°C	317	497	676
Hot gases outlet temperature	°C	97.36	106.35	115.40
Hot gases mass rate	kg/s	0.00448	0.00495	0.00506
Hot gases velocity	m/s	7.67	10.15	12.10
Number of tubes		7		
Outside tube-diameter and tube thickness	m	0.015, 0.0015		
Tube length	m	1.2		
Effectiveness		0.95		
Solution-side H.T.C	W/m ² K	32.95	22.38	20.18
Gas-side H.T.C.	W/m ² K	27.14	30.33	30.89
Overall H.T.C.	W/m ² K	43.07	48.42	50.75

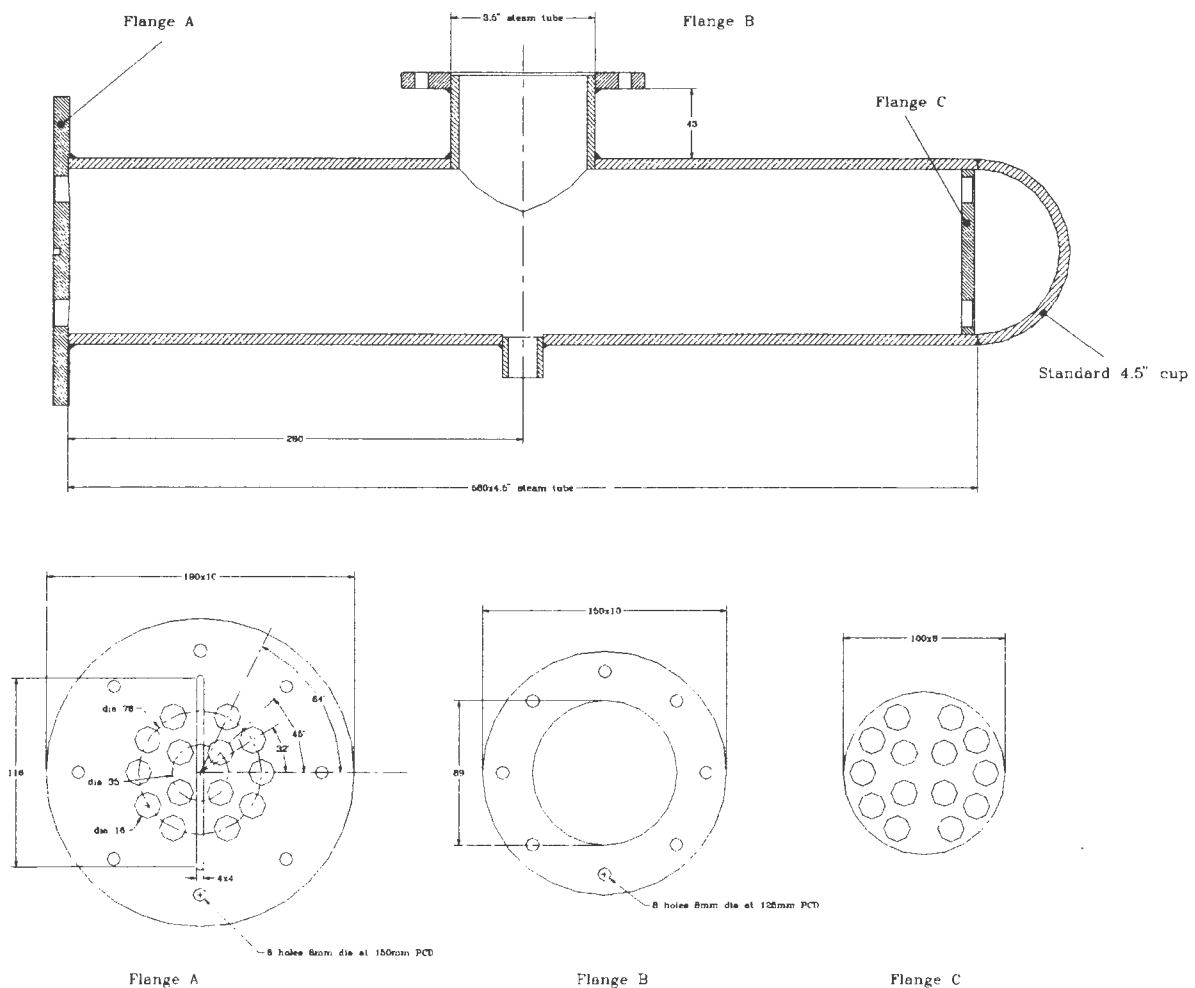


Figure 4. 6 Detail drawing of the generator

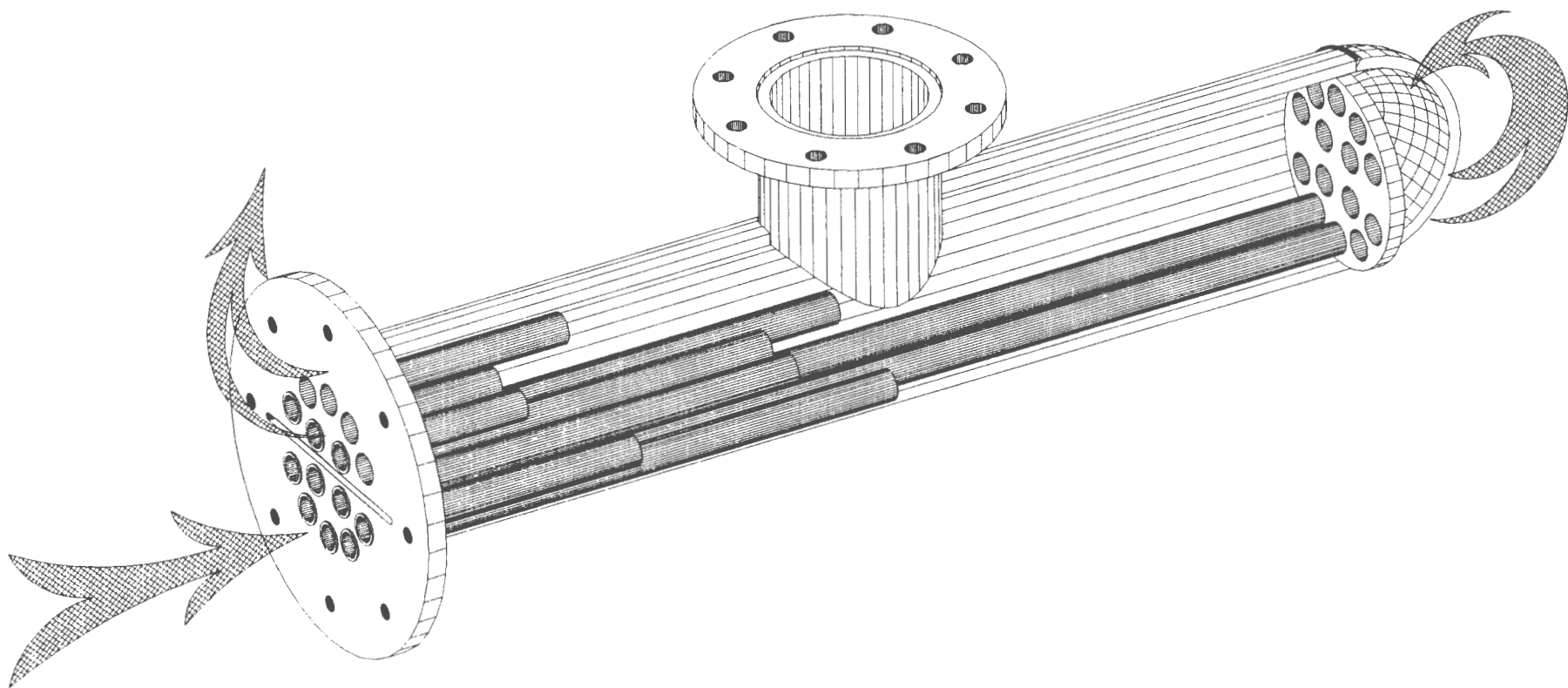


Figure 4. 7 *A 3-D sectional view of the Generator*

4.1.6 Distillation column - the M^CCabe-Thiele method

With reference to the M^CCabe-Thiele diagram of Figure 4.8, the intercept Φ is calculated from the relation:

$$\Phi = \frac{L}{V} \cdot \sqrt{\frac{\rho_{mv}}{\rho_{ml}}}$$

where L/V is the Liquid-Vapour ratio on a plate of the distillation column and ρ_{mv} and ρ_{ml} are the vapour and liquid molal densities respectively.

This ratio is given by:

$$\frac{L}{V} = \frac{R}{R+1}$$

where R is the reflux ratio defined by eq. (3.10).

The co-ordinates of the intercept Φ are (0.0, 0.724) calculated in APPENDIX J.

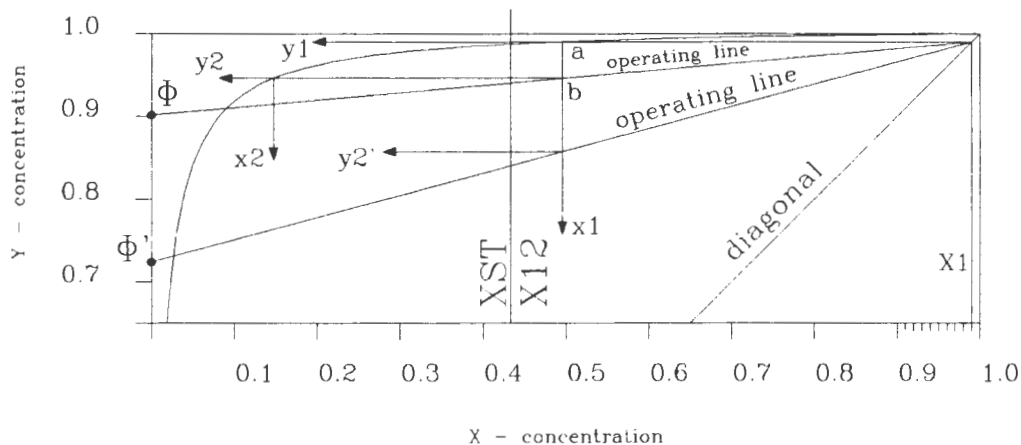


Figure 4.8 The M^CCabe-Thiele method for the construction of the operating line and the estimation of the number of theoretical plates.

A vertical line is drawn from the concentration of the ammonia refrigerant to the 45° diagonal line. From this intersection, a horizontal line (representing the vapour concentration leaving the top plate of the distillation column) is drawn to the liquid-vapour concentration (x-y) curve at point "a". From point "a" a vertical line (representing the liquid concentration falling from

the top plate) is drawn to the operating line at a point “b”. Since the concentration x_1 of this liquid is higher than the concentration of the feed, a second plate is needed. Thus following the same procedure a second vapour line is drawn from point “b” intersecting the x-y curve at a point “c”. This liquid concentration is less than that of the feed.

Figure 4.8 shows that two theoretical plates are sufficient to achieve a distillate of 0.990 concentration, from a feed of concentration 0.433. This is in agreement with the result obtained by the Ponchon-Savarit method, Figure 3.12, although the concentration of the liquid falling from the second plate does not agree to the result of Figure 4.8. This is because the operating line of Figure 4.8 is drawn for an average molal vapour density between the top plate and the vapour leaving the generator.

The operating line with an intercept at the co-ordinate $\Phi' = (0.0, 0.724)$ is obtained for a generator's temperature 140°C . This construction also shows that two plates are sufficient to achieve 0.990 concentration of the distillate. Table 4.7 compares two different design temperatures in the generator.

Table 4.7			
		$T_g=100$	$T_g=140$
reflux ratio		0.03406	0.3659
Sieve diameter	m	0.057	0.063
plate spacing	m	0.100	0.100
vapour velocity	m/s	0.52	0.52
theoretical number of plates		2	2
actual number of plates		3	3
concentration of feed		0.433	0.433
concentration of distillate		0.99	0.99

The comparison shows that an increase in the temperature of the bulk solution in the generator causes a large increase of the mass flow rates within the distillation column and also in the diameter of the plates. The number of plates remains unchanged for a constant concentration of the feed stream as shown in Figure 4.8.

The distillation column is shown in Figures 4.9, 4.10 and in assembly with the generator in Figure 4.11.

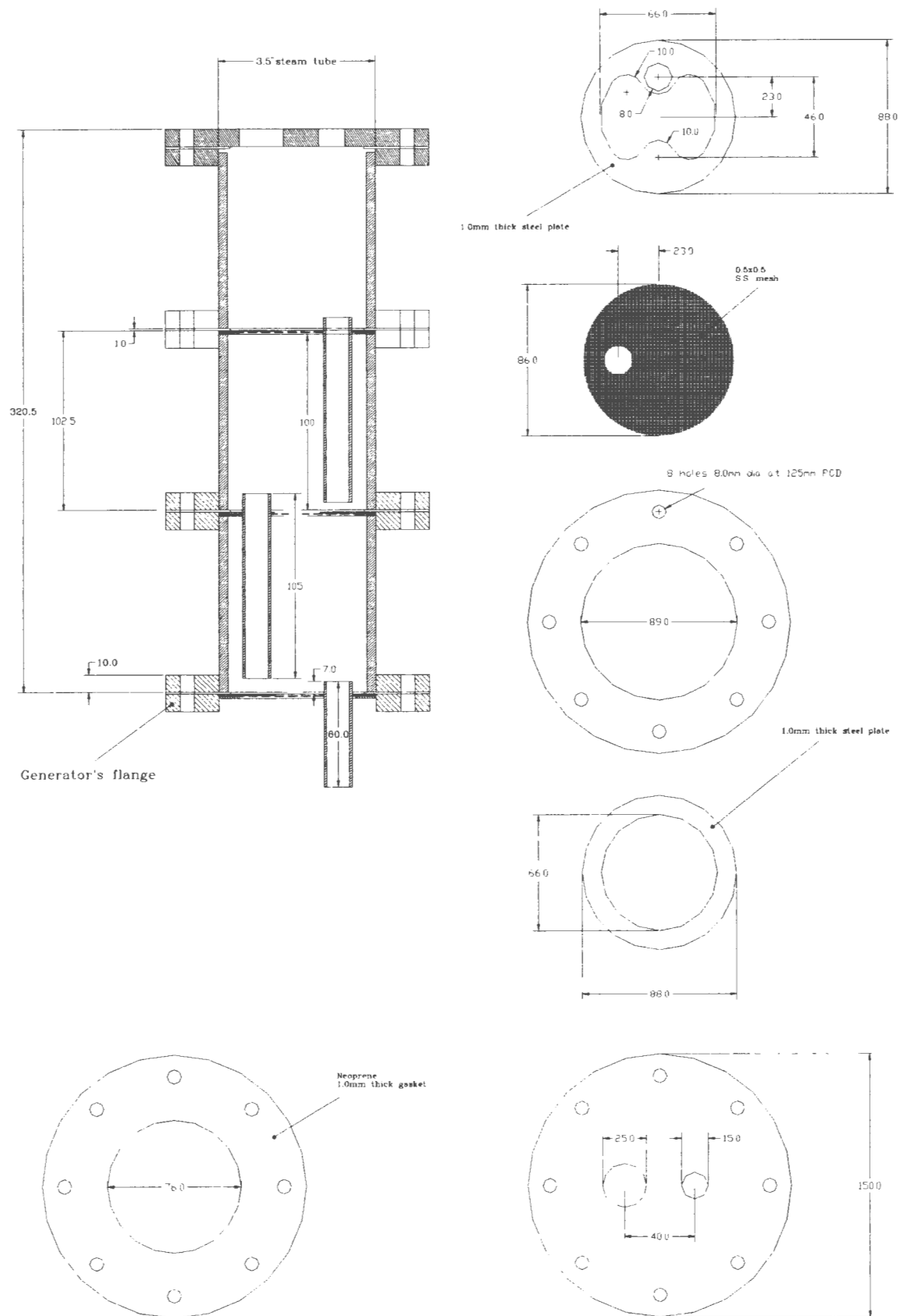


Figure 4.9 Component detail design of the distillation column

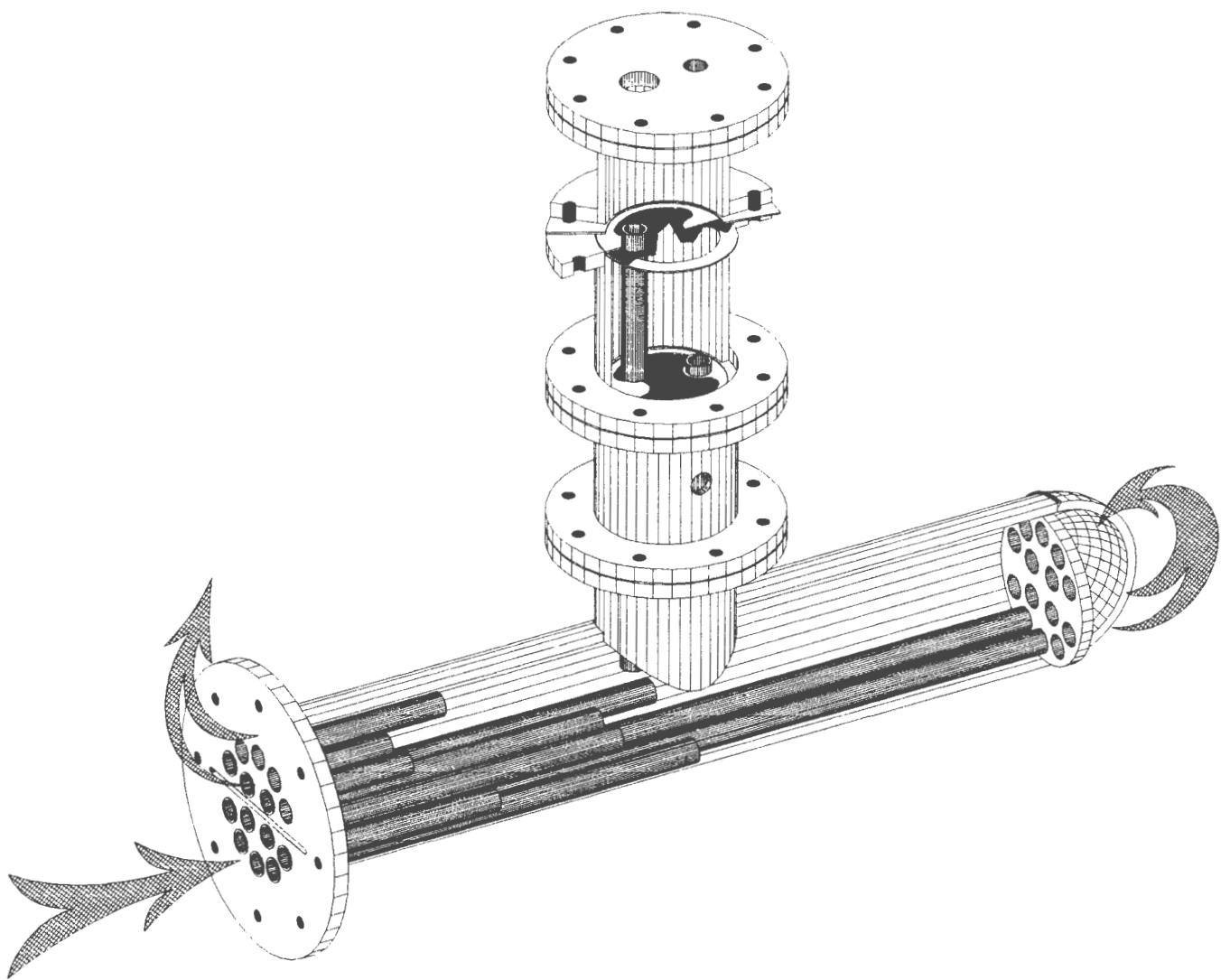


Figure 4. 11 *A 3-D view of the generator-distillation column integrated arrangement*

CHAPTER 5

RESULTS AND CONCLUSIONS

5.0 GENERAL

In this chapter the results from the simulated behaviour are compared to the performance of the absorption refrigeration experimental plant.

Because the experimental plant was designed and built to verify the simulation model, readings were taken only from points monitoring internal heat and mass fluxes. It is felt that, since its components were designed from simulated internal fluid parameters and external environmental conditions, heat transfer relationships can establish an overall energy balance on individual components. Thus except for the condenser (as it will be explained in paragraph 5.4.4), external mass and heat fluxes are not included in this comparative study.

Extensive data were collected in terms of temperatures, pressures and mass flow rates of the refrigerant. These are grouped under four evaporator temperatures: ($T_E = -5^\circ\text{C}$, -10°C , -12°C and -15°C).

For the above evaporator temperatures, data were collected at constant condensate temperature at about 22°C , (except for $T_E = -12^\circ\text{C}$ where data were collected at about 31°C) while varying the temperature of the generator.

The combination of conditions of evaporator $T_E = -12^\circ\text{C}$ and condensate $T_S = 31^\circ\text{C}$ temperatures respectively were purposefully included in the results in order to demonstrate the behaviour of the plant operating beyond its design characteristics. Reference is made to Chapter 3, paragraph 3.4, page 106, where the design condensate temperature had been selected as $T_S = 25^\circ\text{C}$. This temperature was selected on the basis of average environmental conditions of approximately 20°C prevailing in situ, during the winter and spring months. Later, during the hot summer period, prevailing ambient conditions were on average 26°C .

The comparison between the simulated to the experimental results was done as follows:

1. The experimental data were processed using the pertinent equations evaluating the mass flow rate and heat effects within the components of the absorption refrigeration machine culminating in the calculation of the COP.
2. The condensate, evaporator and hot-weak solution temperatures as measured from the experimental plant were processed using the simulation program. This yielded predicted mass flow rates and heat effects within the individual plant components and a value for the COP of the plant.

3. A third group of results was produced by the simulated model for a range of generator temperatures, giving emphasis to the trend of the components' behaviour. In order to achieve this the following experimental temperature data were used:

- (a) the average evaporator temperature $\overline{T_E} = \frac{\sum_{i=1}^n T_{Ei}}{n}$
- (b) the average condensate temperature $\overline{T_S} = \frac{\sum_{i=1}^n T_{Si}}{n}$ n=number of readings

5.1 MEASURING EQUIPMENT

- Temperatures were recorded at points where the solutions are expected to be at a saturation state and before and after each component. The readings were obtained in mV using "Chromel vs. Alumel" thermocouples and converted to degrees Celsius. The thermocouples were inserted in the pipelines at the inlet and exit points of each component through thermocouple-pockets and were in direct contact with the fluids.

The uncertainty of the temperature readings was $\pm 0.01\text{mV}$ which corresponds to $\pm 0.1^\circ\text{C}$.

- Pressures at the condenser, generator, as well as at the absorber and evaporator were measured using ammonia pressure gauges.

The uncertainty for both the high and low pressures readings was $\pm 5\text{kPa}$.

- The refrigerant flow rates were obtained just prior to the expansion valve. The measuring device was a calibrated, 6.0mm internal diameter glass-tube connected in parallel to the condenser. The condenser was isolated and the emptying time of 100mm of the condensate in the glass-tube was used to calculate the mass flow rate.

Experimentation with this type of measurement, based on numerous trials of ten consecutive readings, yielded an uncertainty of $\pm 0.02\text{g/s}$

- The cooling water flow rates for the condenser/deflegmator's load were measured using a calibrated rotameter.

5.1.1 Error analysis

A brief error analysis on experimental data selected at random determines the uncertainty involved in the performances of the generator.

Table A. Data No 3				
Stream	Temp	Pres	Heat	Uncertainty
	°C	bar	kJ/kg	
8	90.50	8.75	527.54	4.80
8	90.40	8.75	527.18	
8	90.50	8.65	522.81	
8	90.40	8.65	522.74	
10	56.26	8.75	55.36	3.54
10	56.16	8.75	54.23	
10	56.26	8.65	52.96	
10	56.16	8.65	51.82	
11	70.61	8.65	1375.24	1.47
11	70.51	8.65	1374.88	
11	70.56	8.70	1374.06	
11	70.61	8.75	1373.77	
12	69.36	8.65	0.18	0.03
12	69.36	8.75	0.17	
12	69.31	8.70	0.17	
12	69.26	8.65	0.15	

Table B. Data No 30				
Stream	Temp	Pres	Heat	Uncertainty
	°C	bar	kJ/kg	
8	101.18	8.75	540.78	3.02
8	101.08	8.65	537.76	
10	64.77	8.65	167.61	3.72
10	64.67	8.75	163.89	
11	80.44	8.65	1394.76	1.77
11	80.44	8.75	1393.00	
12	79.26	8.75	0.65	0.09
12	79.16	8.75	0.57	

Table C. Data No 71				
Stream	Temp	Pres	Heat	Uncertainty
	°C	bar	kJ/kg	
8	119.51	8.65	585.66	2.78
8	119.51	8.55	582.88	
10	65.79	8.55	153.33	2.89
10	65.69	8.65	150.43	
11	92.92	8.55	1423.30	1.87
11	92.92	8.65	1421.43	
12	91.04	8.55	2.04	0.15
12	91.14	8.65	1.89	

Table D. Data No 82				
Stream	Temp	Pres	Heat	Uncertainty
	°C	bar	kJ/kg	
8	94.98	11.65	674.64	6.24
8	94.98	11.55	668.40	
10	59.69	11.65	174.58	4.44
10	59.59	11.55	170.14	
11	90.66	11.55	1375.00	1.43
11	90.56	11.65	1373.57	
12	89.16	11.65	0.72	0.06
12	89.06	11.65	0.66	

Table E. Data No 97				
Stream	Temp	Pres	Heat	Uncertainty
	°C	bar	kJ/kg	
8	114.51	12.15	673.16	3.21
8	114.61	12.05	669.95	
10	73.06	12.15	281.20	3.29
10	72.96	12.05	277.91	
11	101.71	12.05	1390.80	1.52
11	101.61	12.15	1389.28	
12	100.05	12.05	1.43	0.04
12	99.95	12.15	1.39	

The uncertainty analysis is performed on the generator, as a demonstration of the magnitude of the maximum error that could be involved in the determination of the heat input.

In the tables provided, temperatures and pressures are used, taking into account their upper and lower limits, to give the range of the total heat content of each stream around the generator.

In Table A., all four temperature-pressure combinations indicated that the maximum and minimum heat of each stream is when the upper and the lower limits are used respectively, a fact not easily deduced, because of the mass flow rate variations of the various streams.

Therefore, in the subsequent tables only the upper and lower combinations are shown.

A heat balance around the generator yields that:

$$Q_{in} = [Q_8 + Q_{11} - (Q_{10} + Q_{12})] \cdot \dot{m}_{ref}$$

The uncertainty in this value is calculated by the Kline and McClintock¹ method.

¹ **J.P. Holman and W.J. Gajda, Jr**
Experimental Methods for Engineers
 McGraw-Hill Kogakusha, Ltd, Third edition, 1978

$$w_{Q_{in}} = \sqrt{\left[\left(\frac{\partial Q_{in}}{\partial Q_8} \cdot w_{Q_8}\right)^2 + \left(\frac{\partial Q_{in}}{\partial Q_{11}} \cdot w_{Q_{11}}\right)^2 + \left(\frac{\partial Q_{in}}{\partial Q_{10}} \cdot w_{Q_{10}}\right)^2 + \left(\frac{\partial Q_{in}}{\partial Q_{12}} \cdot w_{Q_{12}}\right)^2 + \left(\frac{\partial Q_{in}}{\partial \dot{m}_{ref}} \cdot w_{\dot{m}_{ref}}\right)^2\right]}$$

where

$$\frac{\partial Q_{in}}{\partial \dot{m}_{ref}} = [Q_8 + Q_{11} - (Q_{10} + Q_{12})]$$

$$\frac{\partial Q_{in}}{\partial Q_8} = \dot{m}_{ref}$$

$$\frac{\partial Q_{in}}{\partial Q_{11}} = \dot{m}_{ref}$$

$$\frac{\partial Q_{in}}{\partial Q_{10}} = -\dot{m}_{ref}$$

$$\frac{\partial Q_{in}}{\partial Q_{12}} = -\dot{m}_{ref}$$

$w_{\dot{m}_{ref}} = 0.02 \text{ g/s}$ (previously shown)

The above equations are used to construct the following table of uncertainties.

Table F.				
Data No	$\dot{m}_{ref}$	Q_{in} Nominal	Uncertainty	%
	g/s	W	W	
3	0.830	1914.46	5.1	0.27
30	0.850	2001.18	4.3	0.21
71	0.810	2364.20	3.6	0.15
82	0.880	4507.95	6.9	0.15
97	0.871	2671.64	4.2	0.16

The results in Table F. demonstrate that the uncertainty in the experimental data is very low.

This was to be expected because of the simple nature of measurements.

5.2 EXPERIMENTAL PROCEDURE AND CONDITIONS

The absorption refrigeration unit was firstly evacuated and subsequently charged with desalinated water and anhydrous ammonia through the water-cooled absorber. The amount of ammonia charged was varied, in order to produce the desired evaporator temperature. For example, to achieve an evaporator temperature of -10°C when the cooling in the absorber maintains the solution at 25°C , the concentration of the strong solution must be in the vicinity of 50% (see the h-x diagram of Figure C1 in APPENDIX C). For other evaporator temperatures the concentration of the solution varied accordingly. For each evaporator temperature (there were four in total), readings were taken at intervals of 30min, but only the ones recorded during equilibrium conditions were processed. Equilibrium conditions were established when:

- the liquid level in the condenser remained constant
- the liquid level in the absorber remained constant
- the temperature differences between consecutive readings at the evaporator and generator did not exceed 1°C .

5.3 CONTROLS OF THE EXPERIMENTAL PLANT

Data were collected from the designated points on the plant at various generator temperature settings. These settings were at random intervals due to a number of controlled conditions contributing to an equilibrium state of the plant and to a temperature change in the generator.

These were controlled as follows:

- Solution flow rates: The volumetric flow rates of the solutions were matched so that with the incoming refrigerant, the liquid level in the absorber remained constant. Because the speed of the pump could not be altered, an “adjustable toggle” valve was positioned in parallel to the pump, in such a way, that when activated by the piston motion it released the excessive fluid back to the absorber.
- External heat input to the generator: This was controlled by adjusting the amount of combustion gases² through the generator tubes. The combustion gases are the product of

² Combustion gases were used to simulate the conditions from an internal combustion engine exhaust-gases energy source.

burning Town-gas³ in air. An air-draft was provided to assist the flow of gases through the generator tubes.

- Condenser and absorber cooling loads: Constant temperature was maintained in the condenser and absorber by regulating the amount of cooling water.
- Refrigeration capacity and evaporator temperature: The aim of the experiment was to keep a constant 1kW refrigeration capacity at a constant evaporator temperature, thus monitoring the refrigeration requirement. This was achieved by the combined adjustment of the refrigerant expansion valve and the amount of water circulating through the annular space of the evaporator tubes.
- Reflux mass flow rate: This was controlled by adjusting the valve setting so that the vapour leaving the distillation column maintained a constant temperature. This temperature (about 60°C) was a predetermined value, known from the equilibrium condition of the vapour at high pressure (about 9bar) and with concentration in the vicinity of 0.990 (see the h-x diagram of Figure C1 in APPENDIX C).

5.4 COMPARISON OF RESULTS

The experimental readings are shown in Table 5.1 and the simulated results in Table 5.2.

5.4.1 Generator's heat and mass flow rates

The effect of the generator temperatures on the heat supply requirements per unit mass of refrigerant is shown in Figures 5.1 to 5.4.

The left side of each of the simulated trend curves indicates a minimum value. This corresponds to the optimum temperature T_8^{opt} . The trend also shows that the heat supply increases with the increase of the weak solution's temperature. The experimental results follow

³ Town-gas is a mixture of: 4.2% CO₂, 0.6% O₂ 19.0% CO 19.0% CH₄
 45.0% H₂ 3.2% Heavy hydrocarbons
 The calorific value is 16MJ/m³

this trend closely at the higher evaporator temperatures $T_E = -5^\circ\text{C}$ and $T_E = -10^\circ\text{C}$, but show considerable scatter at the lower evaporator temperatures $T_E = -12^\circ\text{C}$ and $T_E = -15^\circ\text{C}$.

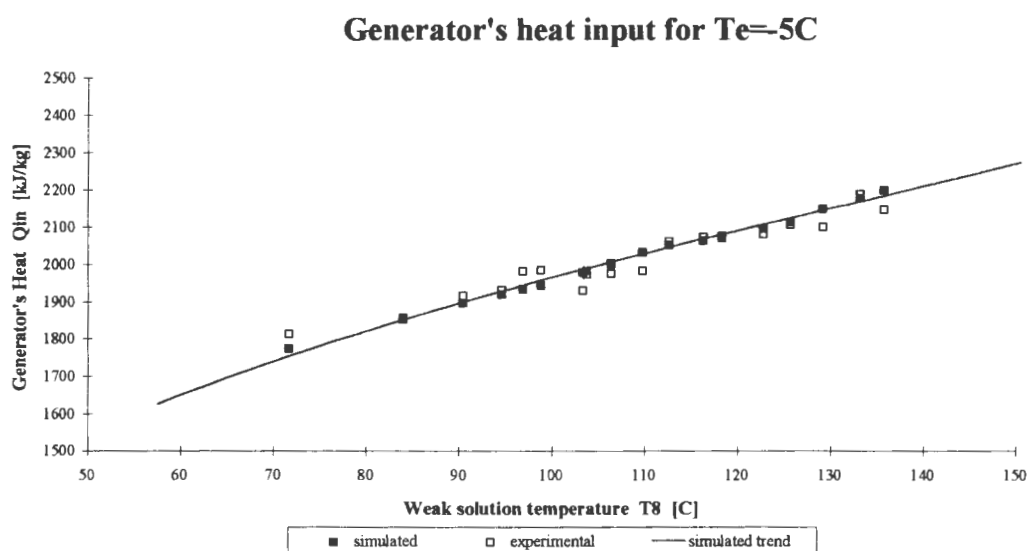


Figure 5.1 The trend of heat input at a condensate temperature $T_S = 22.14^\circ\text{C}$

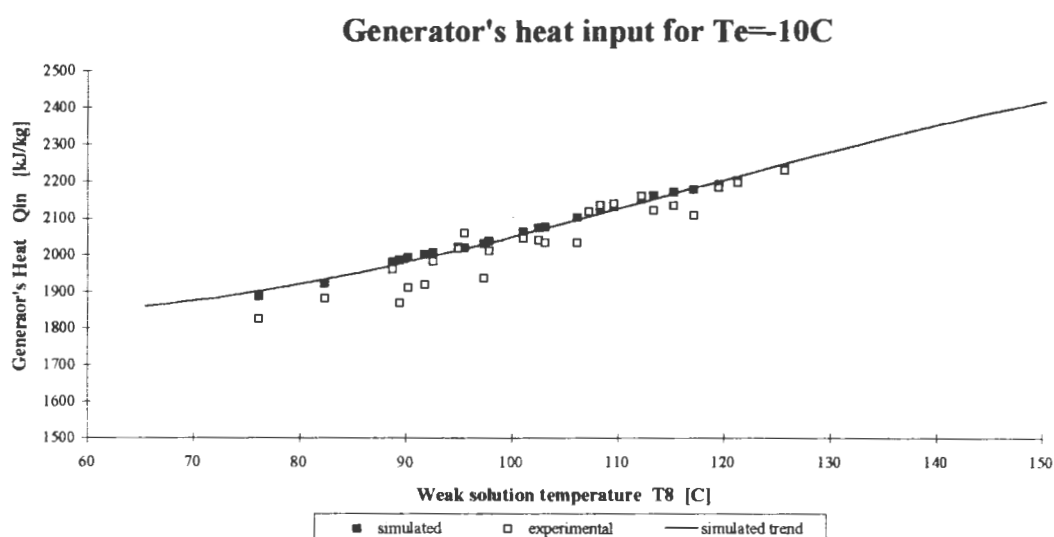


Figure 5.2 The trend of heat input at a condensate temperature $T_S = 22.43^\circ\text{C}$

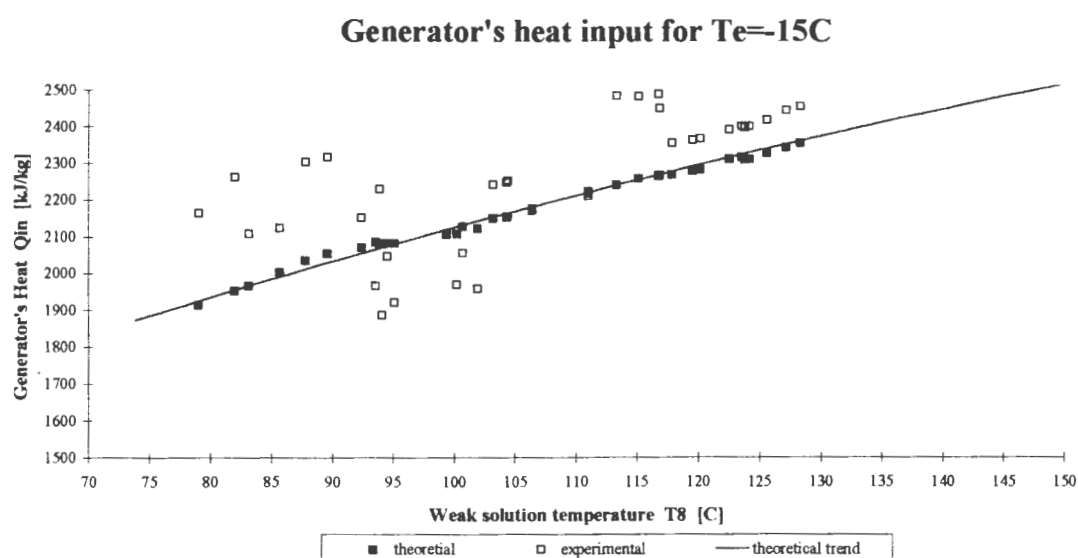


Figure 5.3 The trend of heat input at a condensate temperature $T_s = 22.78^\circ\text{C}$

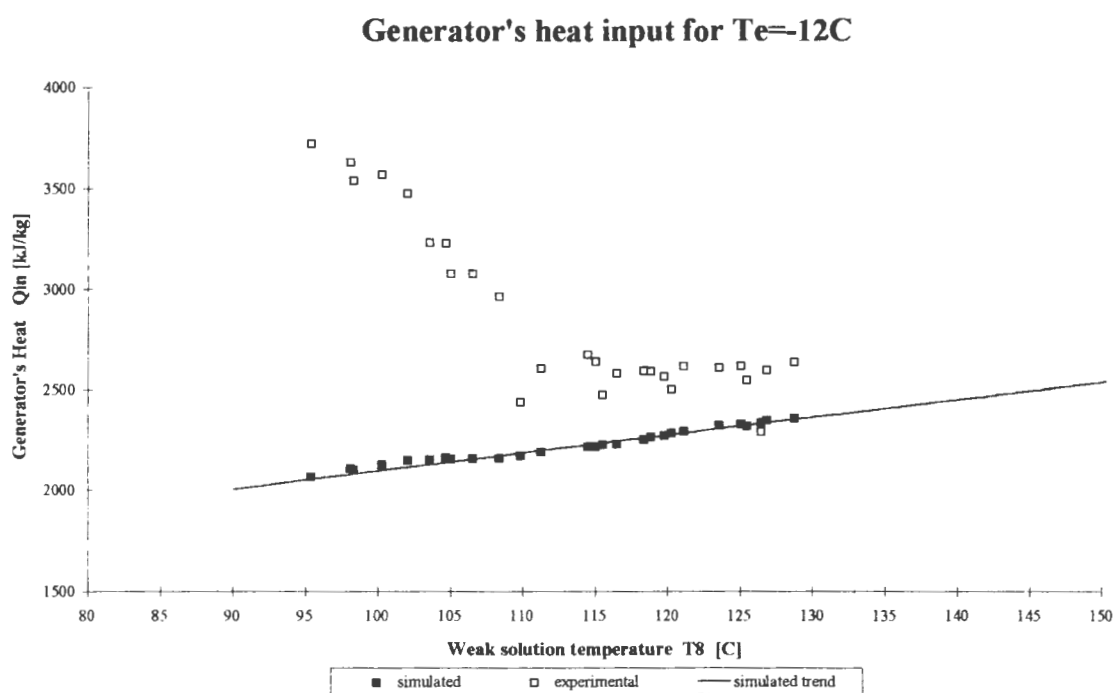


Figure 5.4 The trend of heat input at a condensate temperature $T_s = 31.67^\circ\text{C}$

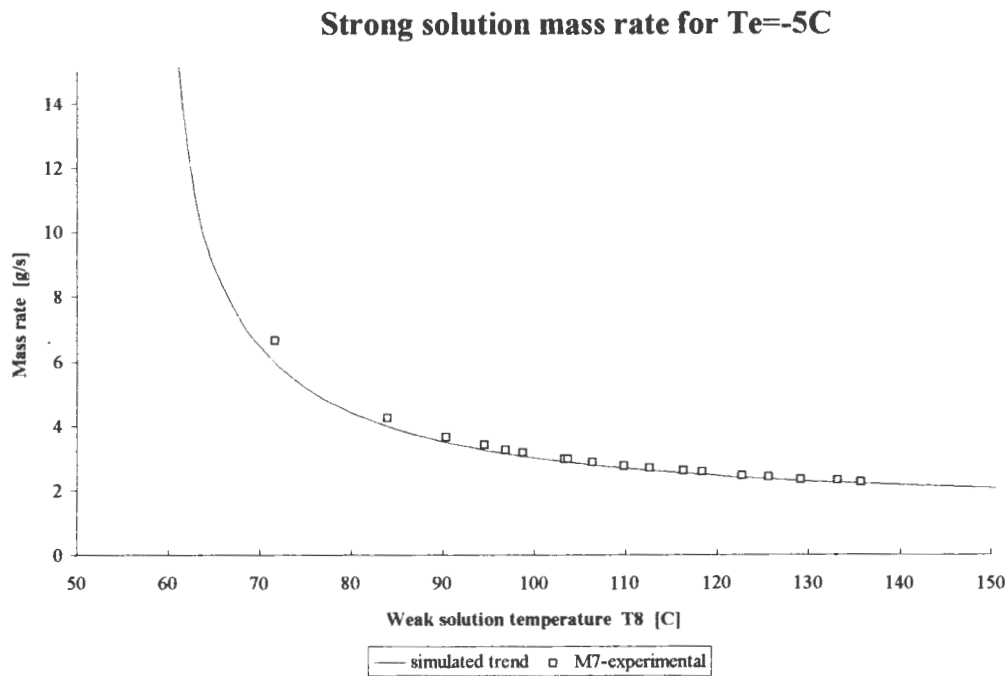


Figure 5. 5 *The trend of the solution mass flow rate at a condensate temperature $T_s = 22.14^\circ\text{C}$*

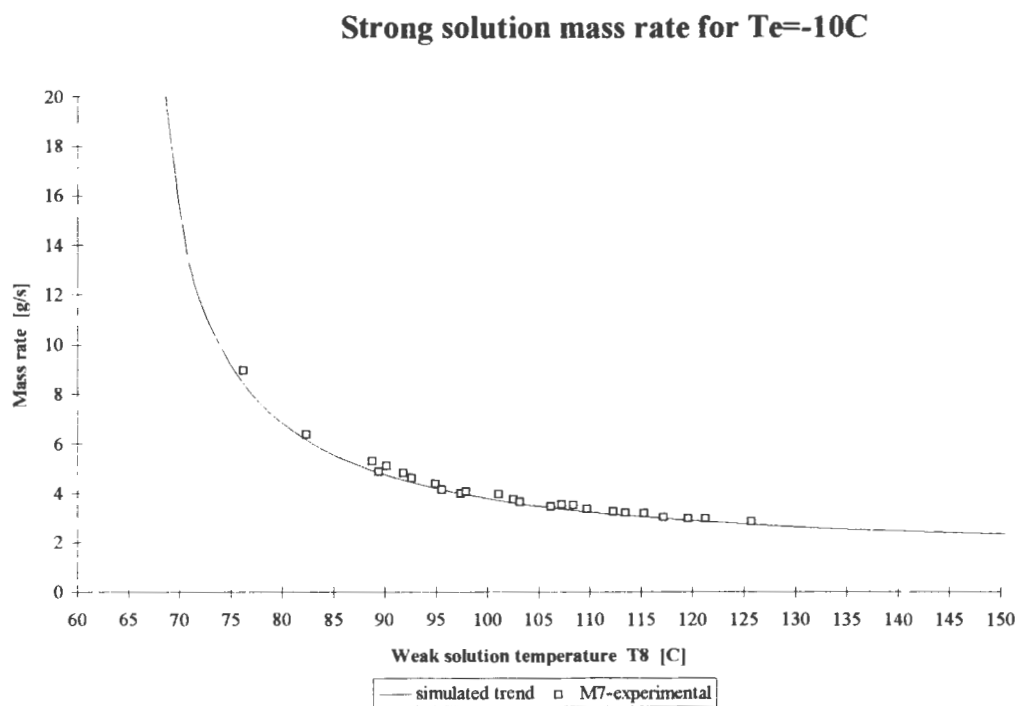


Figure 5. 6 *The trend of the solution mass flow rate at a condensate temperature $T_s = 22.43^\circ\text{C}$*

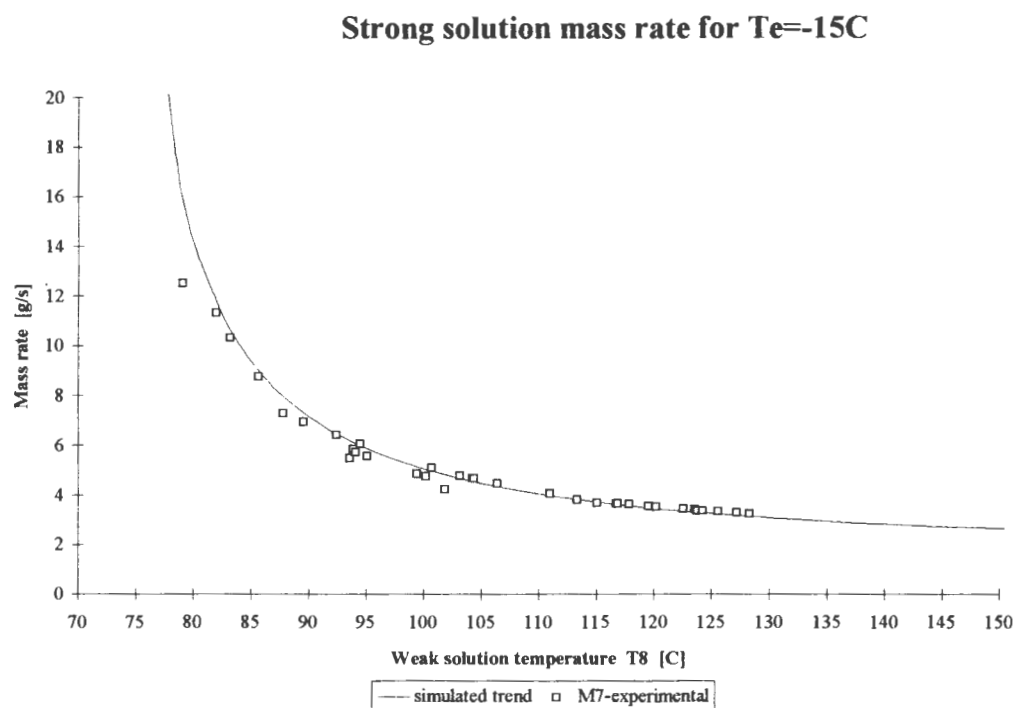


Figure 5. 7 The trend of the solution mass flow rate at a condensate temperature $T_s = 22.78^\circ\text{C}$

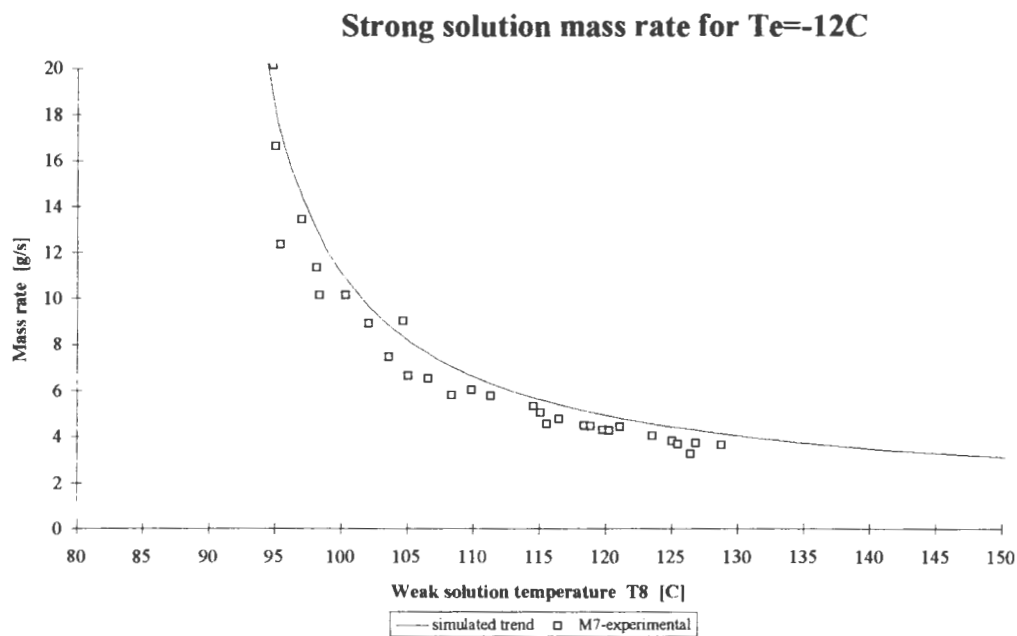


Figure 5. 8 The trend of the solution mass flow rate at a condensate temperature $T_s = 31.67^\circ\text{C}$

AMMONIA-WATER SOLUTIONS

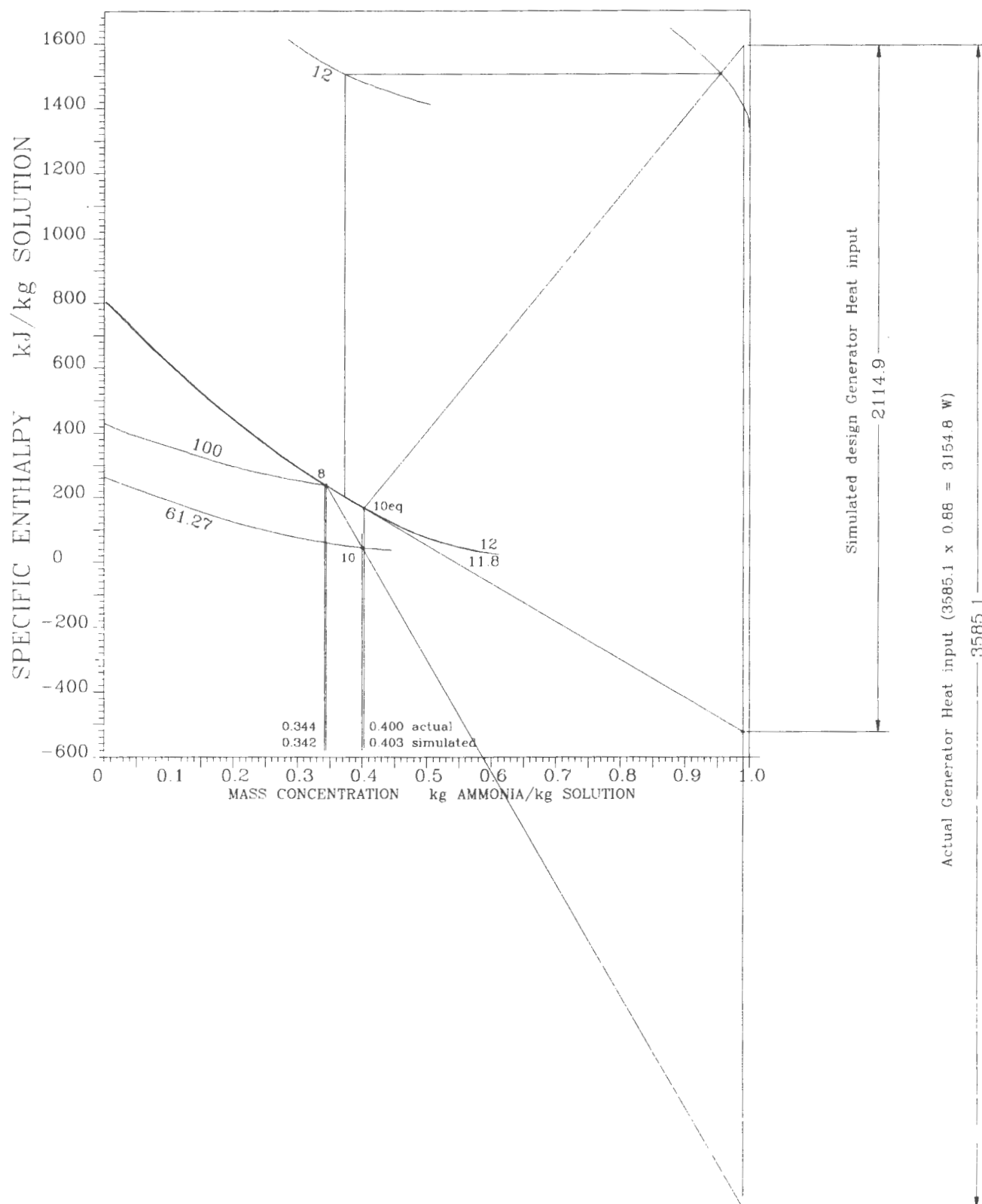


Figure 5.9 Graphical illustration of the disagreement between the experimental and the simulated generator's heat as noticed in Figure 5.4

The graph of Figure 5.4 (for $T_E = -12^\circ\text{C}$) shows considerable disagreement from the expected simulated results. This can be explained in view of the following observations:

- Figures 5.5 to 5.8 show the mass flow rate of the strong solution (and thus of the weak solution too) increasing rapidly as the minimum temperature T_{10eq} is approached.
- Figure 5.8 depicts strong solution mass flow rates (for $T_E = -12^\circ\text{C}$) much higher than in any other graph. These rates exceed by far the design mass flow rate ($m_{13} = m_7 \approx 4.5\text{g/s}$) through the heat exchanger (see Table 3.6 in page 125). For example at $T_8 = 100^\circ\text{C}$ the measured flow rate from the graph is approximately 10g/s .
- A comparison of the columns T_{10} and T_{10eq} in Table 5.1 of page 179, shows that the actual temperatures of the strong solution at the exit of the heat exchanger, are much lower than the optimised temperatures when saturation is reached.

From the last two observations it can be deduced that the heat exchanger is not optimised for such flow rates. Thus the increase in the heat input to the generator, as observed in the experimental results in Figure 5.4, is due to the cooler strong solution having been supplied. The increased heat requirement in the generator is illustrated by the h - x diagram of Figure 5.9, where the simulated design generator's heat input is compared to the experimental actual heat input.

5.4.2 Heat rejected from the absorber

Figures 5.10 to 5.13 exhibit similar trend as that observed for the heat input to the generator in Figures 5.1 to 5.4. The simulated results show a minimum value which corresponds to the optimum temperature T_8^{opt} . Near this temperature, experimental results in Figure 5.13 show a large thermal load in the absorber. This is for the same reason discussed in paragraph 5.4.1.

For further clarity of the heat and mass flow rates involved with the absorber, a detailed heat balance is shown in Figures 5.14 to 5.21.

Figures 5.17 and 5.21 indicate a large deviation in the heat rates involved with the streams 9 and 10 from the trend. These are the inlet streams to the absorber and generator respectively.

The considerable disagreement observed in Figures 5.13 and 5.21 is to be expected on the basis that it was the heat exchanger's inability to cope with the large mass flow rates that affected the performance of the generator, as shown in Figures 5.4 and 5.8 and in the subsequent discussion.

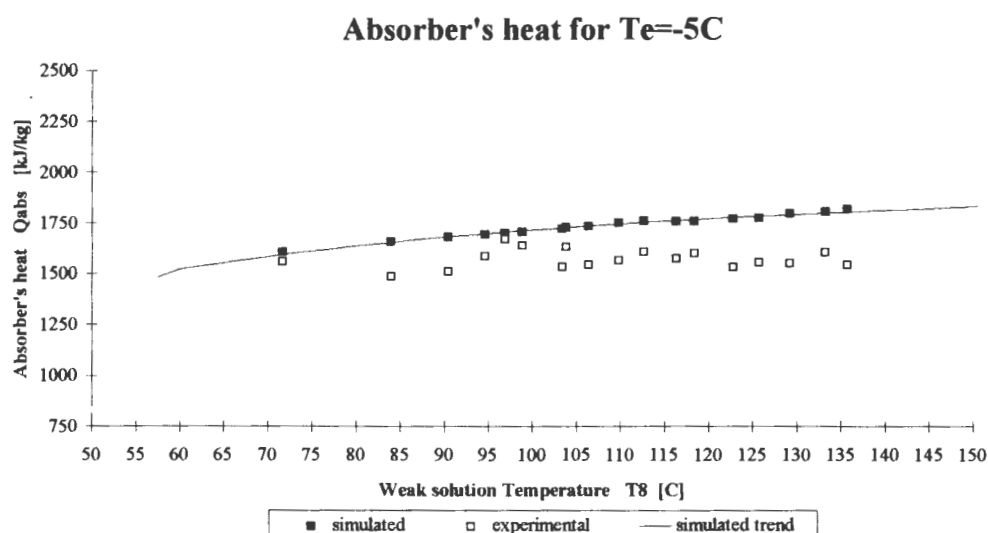


Figure 5. 10 The trend of the absorber's heat rate at a condensate temperature $T_s = 22.14^\circ\text{C}$

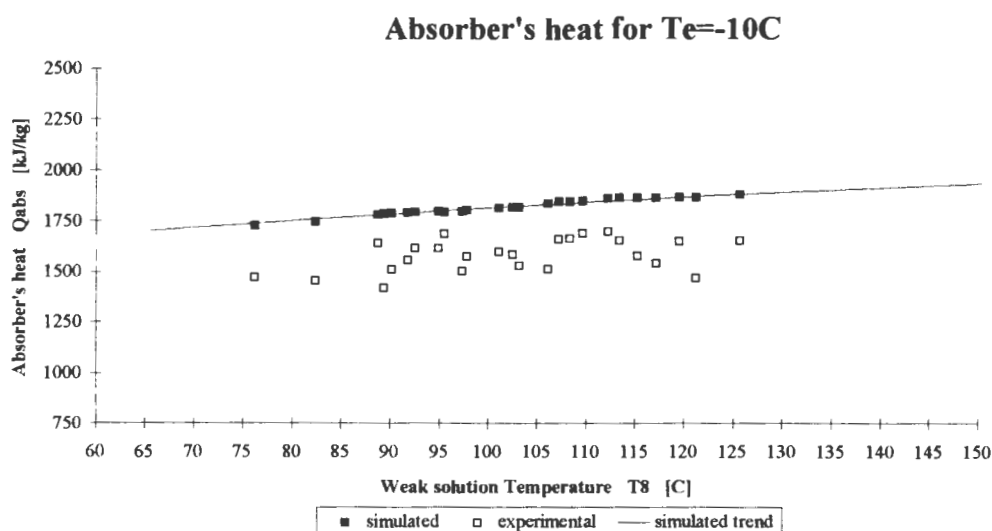


Figure 5. 11 The trend of the absorber's heat rate at a condensate temperature $T_s = 22.43^\circ\text{C}$

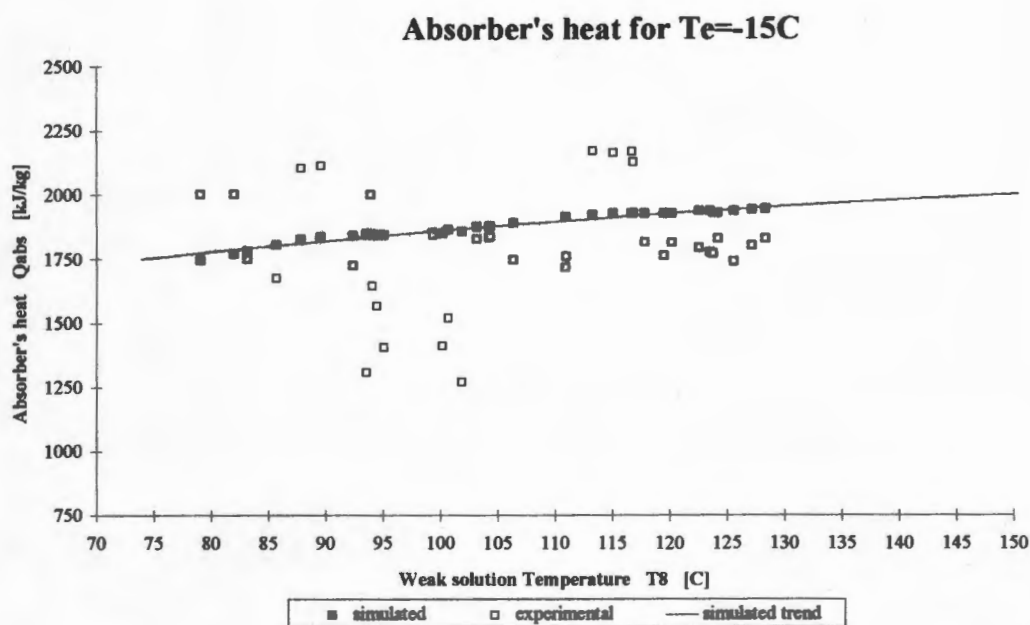


Figure 5. 12 The trend of the absorber's heat rate at a condensate temperature $T_s = 22.78^\circ\text{C}$

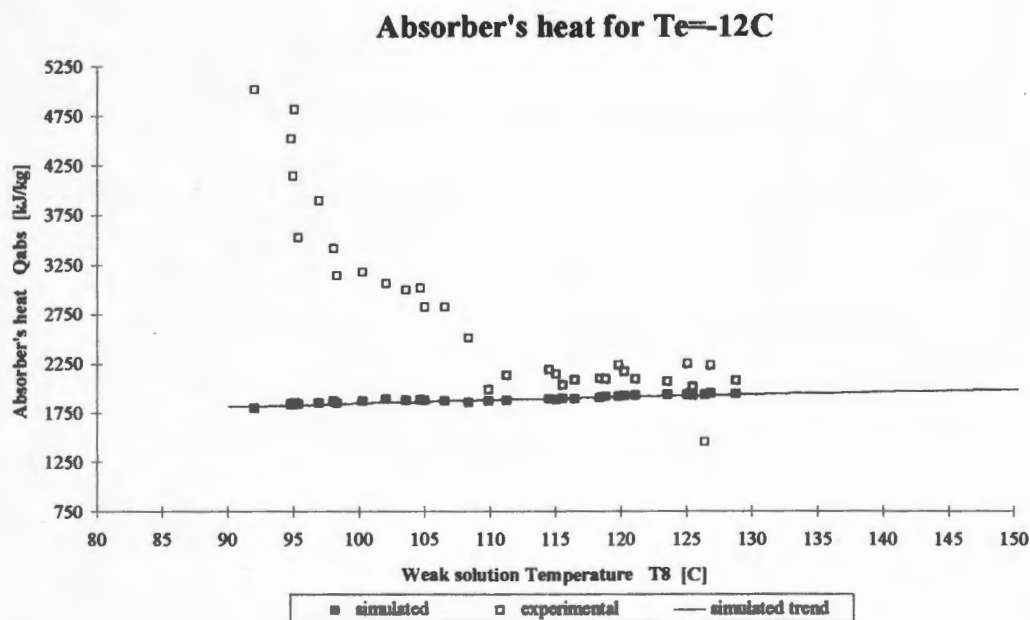


Figure 5. 13 The trend of the absorber's heat rate at a condensate temperature $T_s = 31.67^\circ\text{C}$

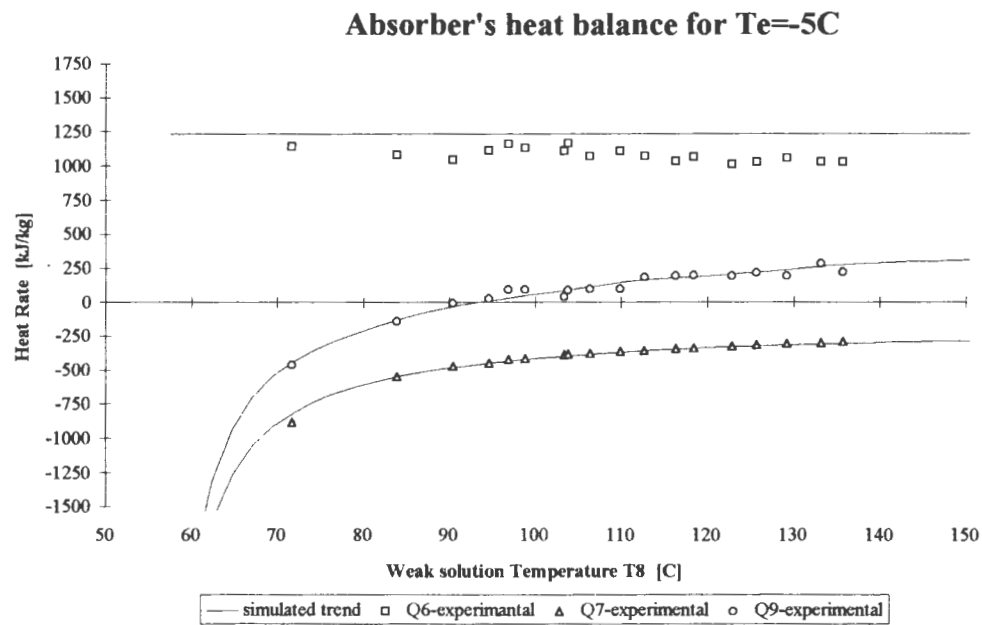


Figure 5. 14 The trend of the absorber's heat balance at a condensate temperature $T_s = 22.14^\circ\text{C}$

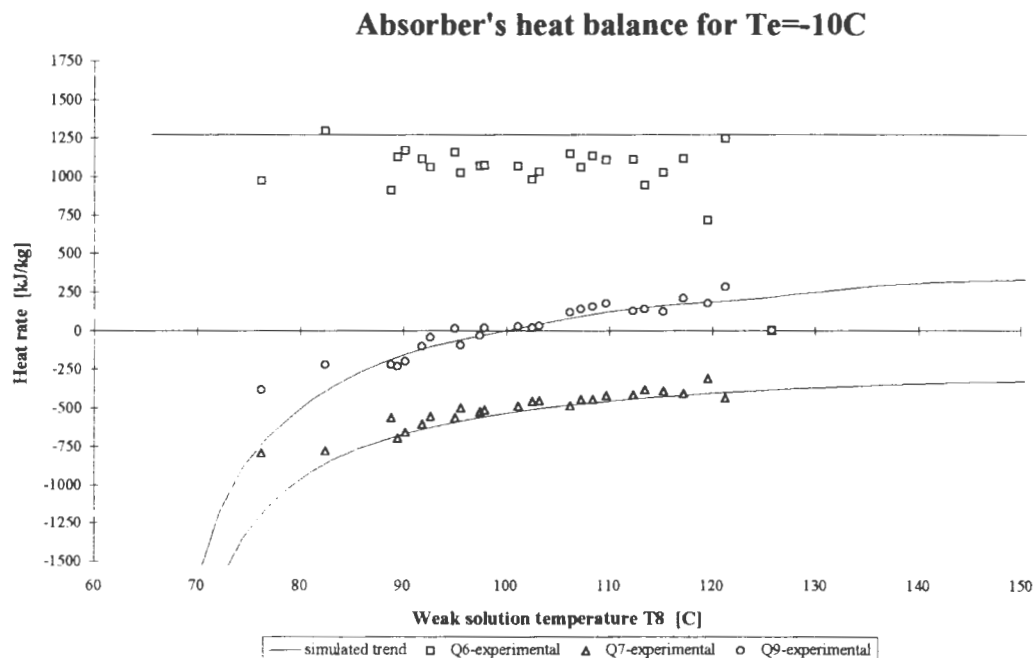


Figure 5. 15 The trend of the absorber's heat balance at a condensate temperature $T_s = 22.43^\circ\text{C}$

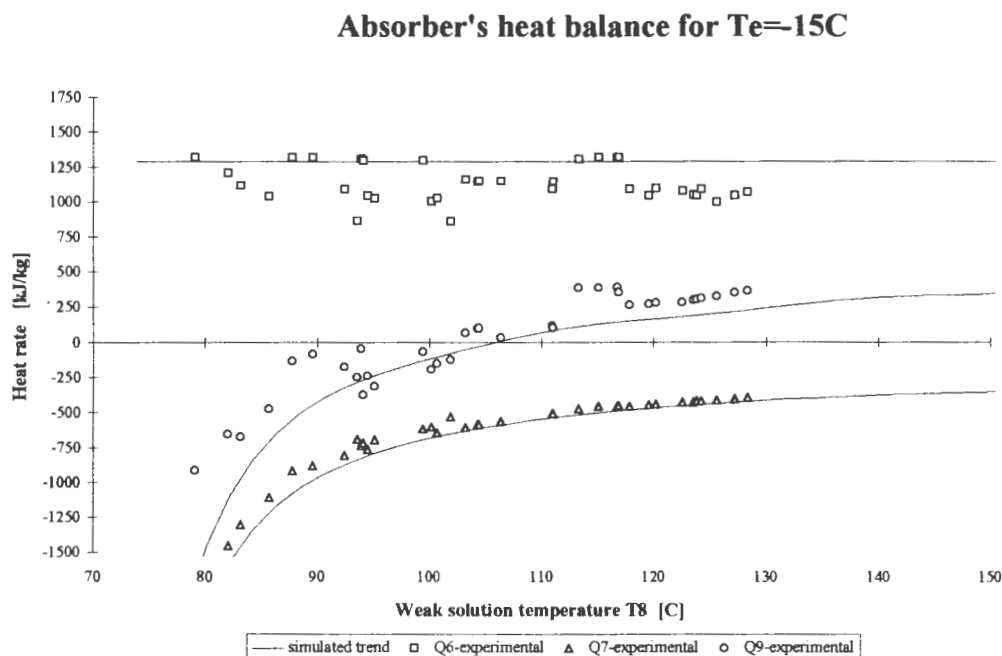


Figure 5. 16 The trend of the absorber's heat balance at a condensate temperature $T_s = 22.78^\circ\text{C}$

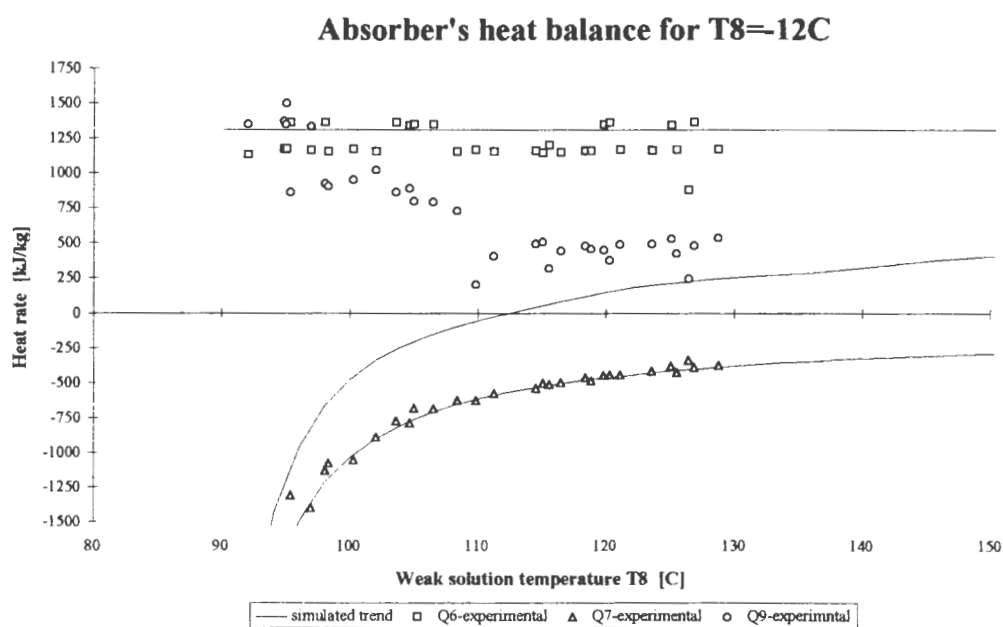


Figure 5. 17 The trend of the absorber's heat balance at a condensate temperature $T_s = 31.67^\circ\text{C}$

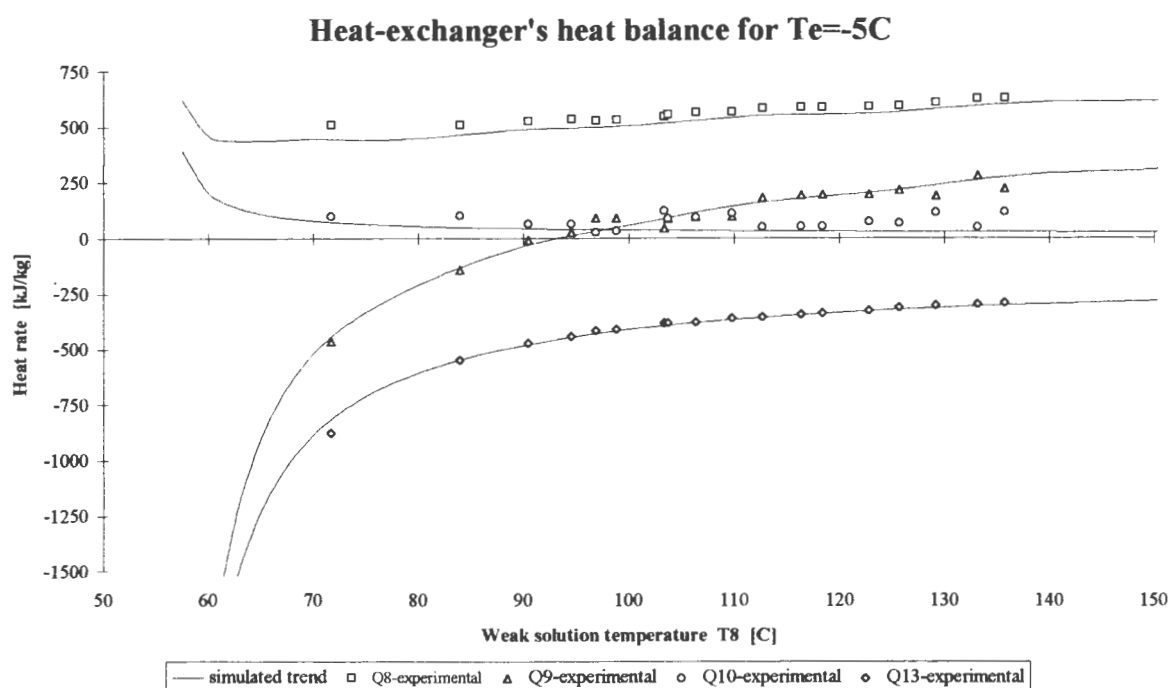


Figure 5. 18 The trend of the H. exchanger's heat balance at a condensate temperature $T_s = 22.14^\circ\text{C}$

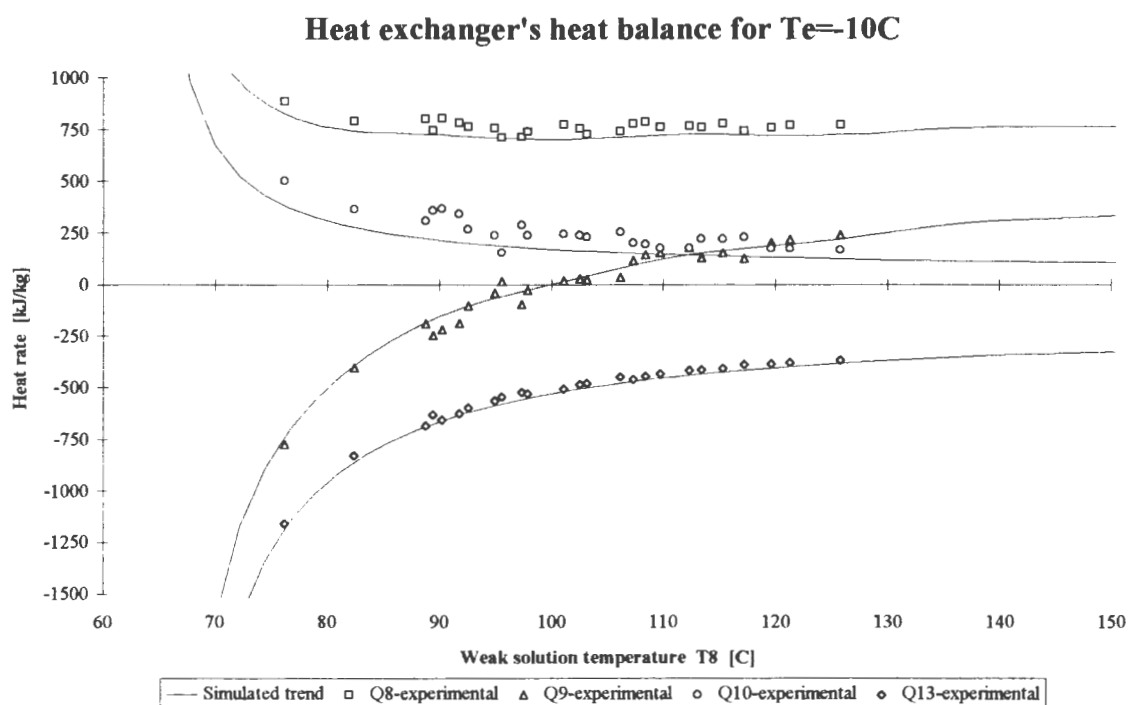


Figure 5. 19 The trend of the H. exchanger's heat balance at a condensate temperature $T_s = 22.43^\circ\text{C}$

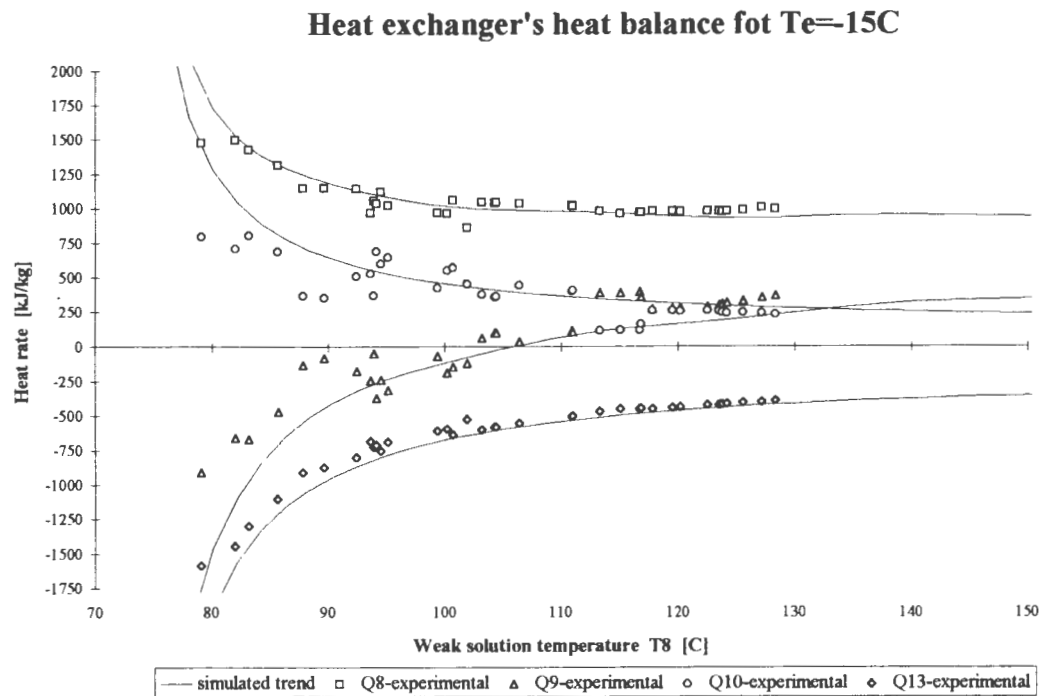


Figure 5. 20 The trend of the H. exchanger's heat balance at a condensate temperature $T_s = 22.78^\circ\text{C}$

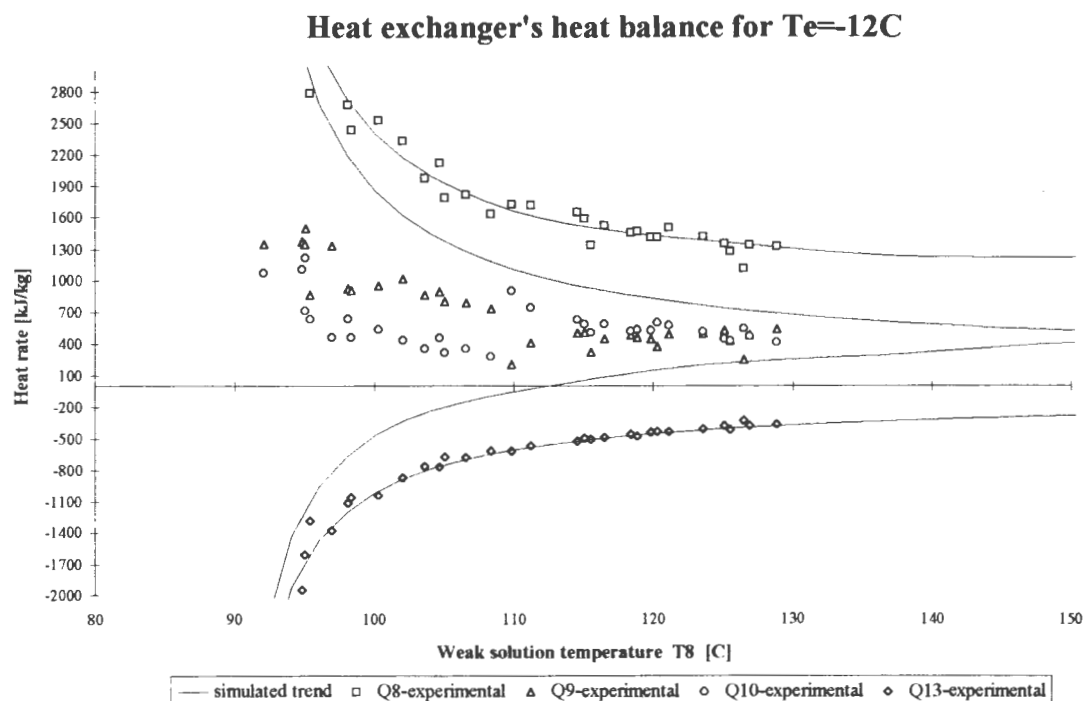


Figure 5. 21 The trend of the H. exchanger's heat balance at a condensate temperature $T_s = 31.67^\circ\text{C}$

5.4.3 Heat rejected from the condenser and deflegmator

The condenser is designed to include the deflegmator's heat load too. Experimental temperature readings were collected from the condenser's refrigerant and cooling-water inlet and outlet streams. The mass flow rate of the cooling-water was also recorded. The total condenser's load is calculated as the heat received by the water. The condenser's load alone is calculated from the heat rejected to liquefy the saturated vapour, at a rate corresponding to the mass flow rate of the refrigerant. The difference of the two is the deflegmator's heat.

The graphs in Figures 5.22 to 5.25 show a good agreement between the experimental and simulated results. The simulated condenser's alone is shown unaffected by the changes in the generator's conditions. This is because both the mass flow rate and the condensing temperature are constant.

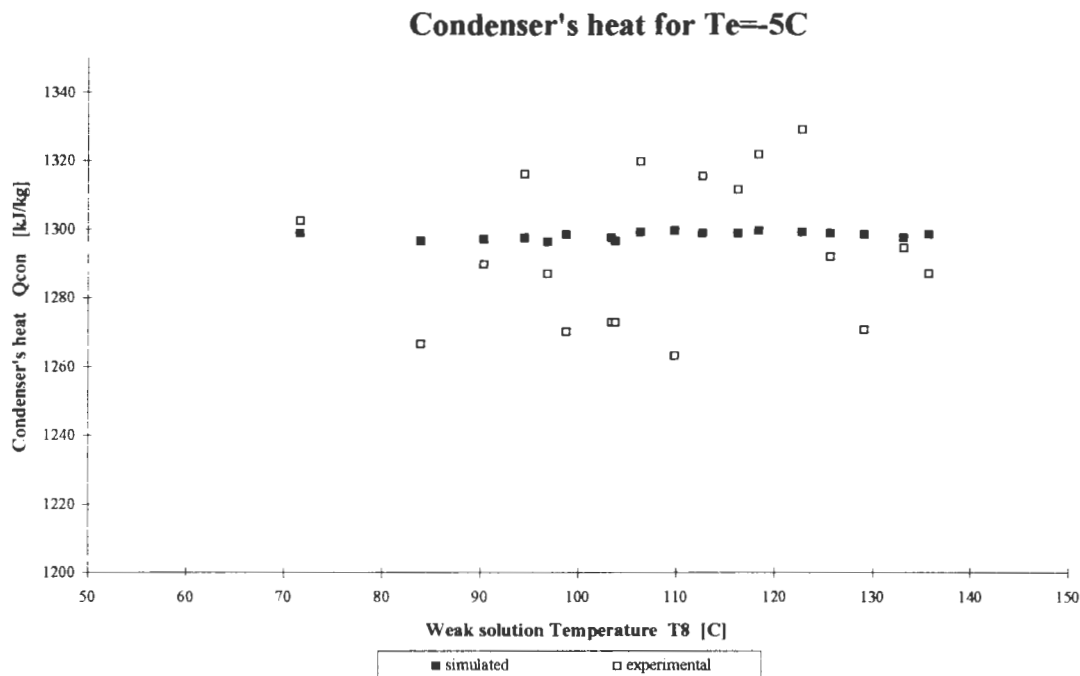


Figure 5.22 The trend of the condenser's load at a condensate temperature $T_s = 22.14^\circ\text{C}$

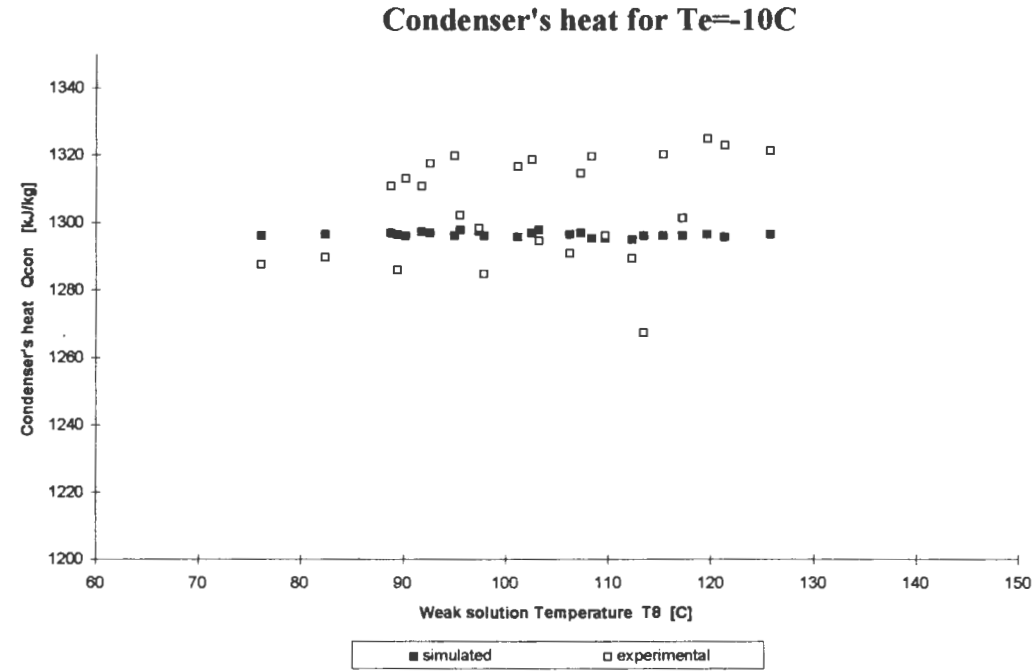


Figure 5. 23 The trend of the condenser's load at a condensate temperature $T_S = 22.43^\circ\text{C}$

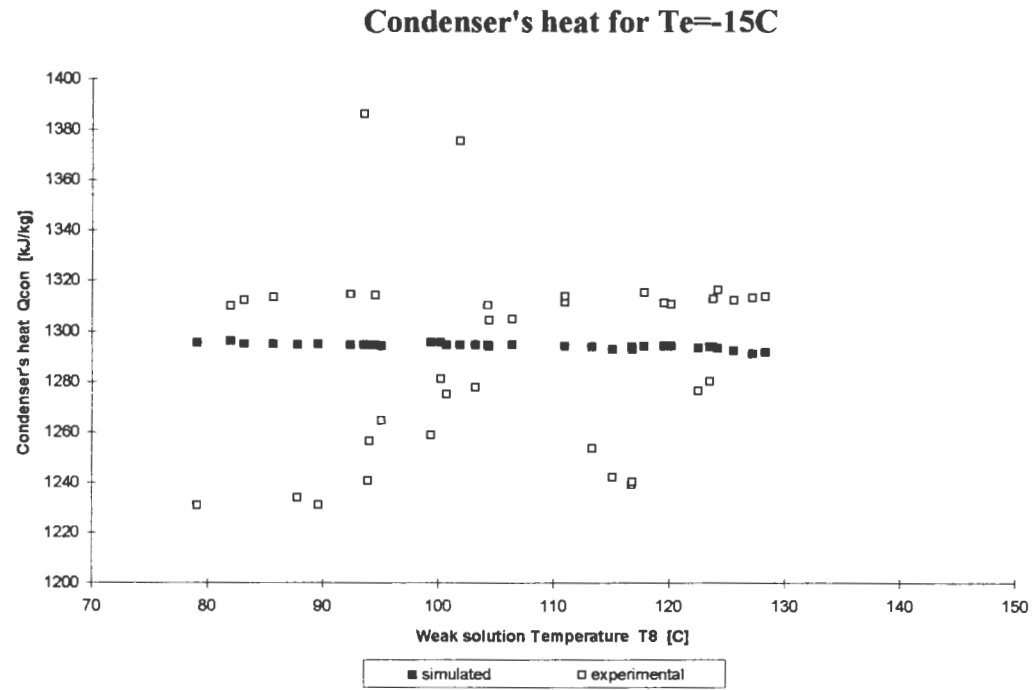


Figure 5. 24 The trend of the condenser's load at a condensate temperature $T_S = 22.78^\circ\text{C}$

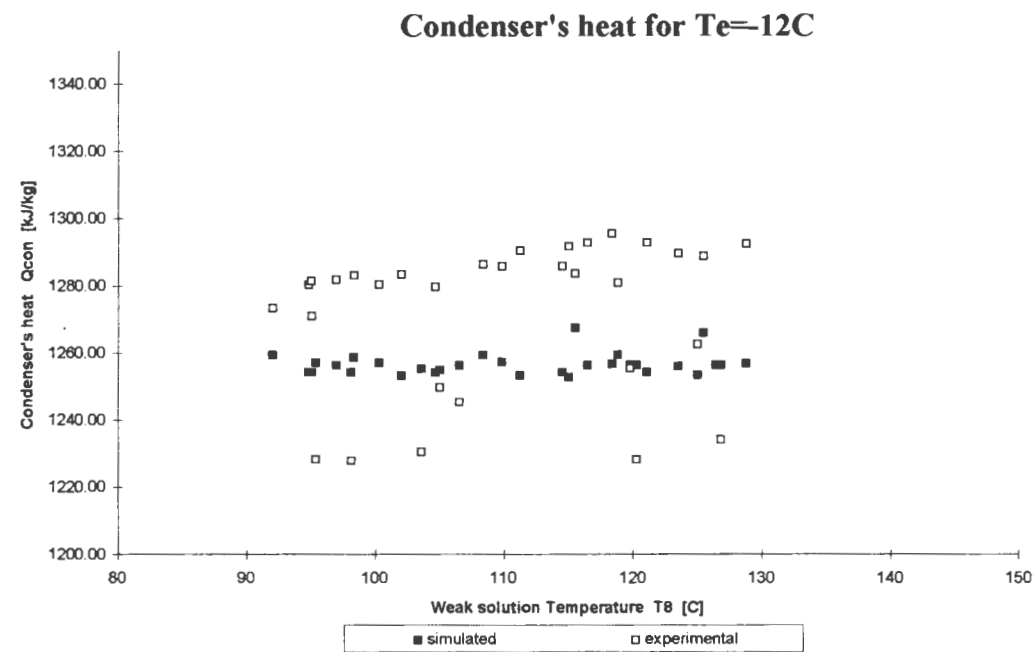


Figure 5. 25 The trend of the condenser's load at a condensate temperature $T_s = 31.67^\circ\text{C}$

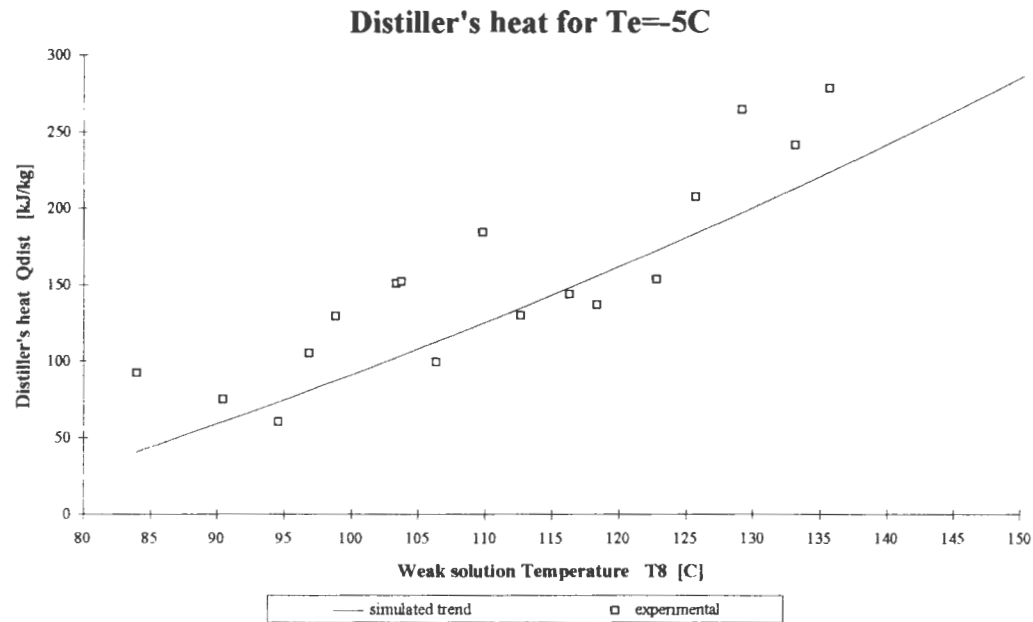


Figure 5. 26 The trend of the distiller's heat rate at a condensate temperature $T_s = 22.14^\circ\text{C}$

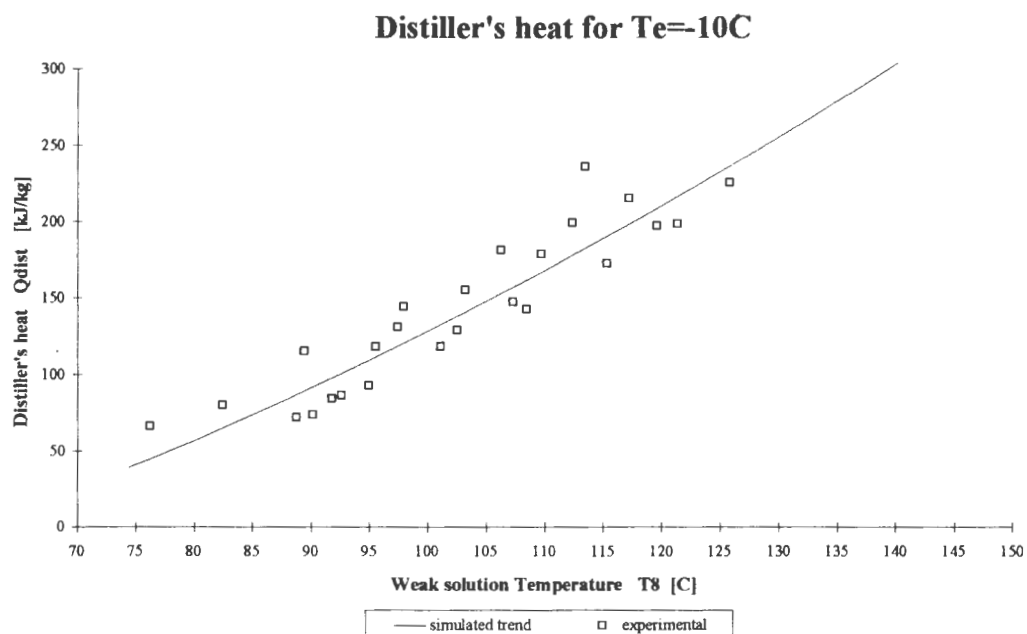


Figure 5. 27 The trend of the distiller's heat rate at a condensate temperature $T_s = 22.43^\circ\text{C}$

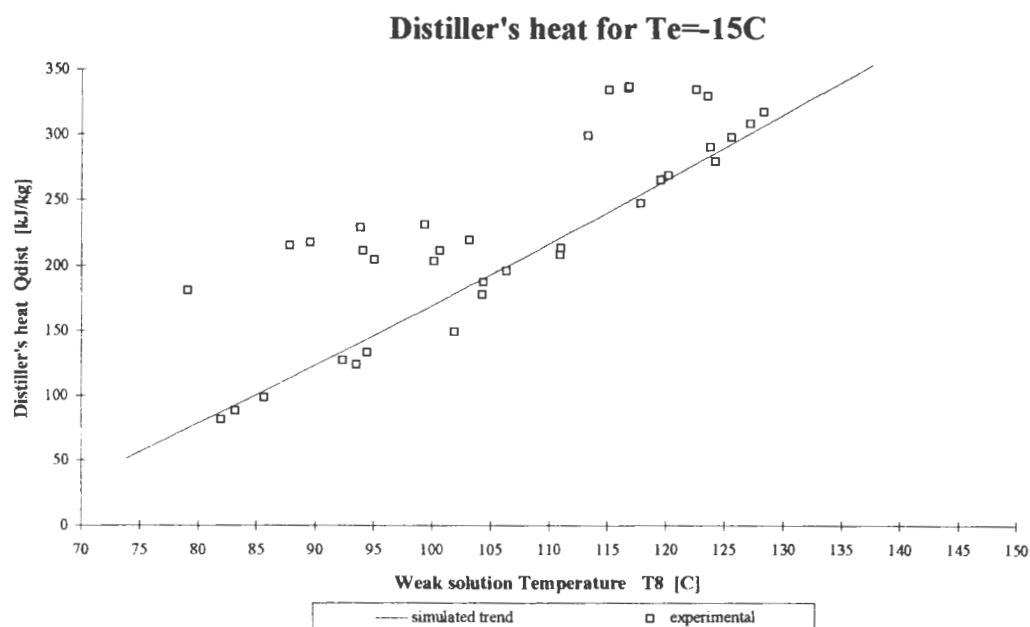


Figure 5. 28 The trend of the distiller's heat rate at a condensate temperature $T_s = 22.78^\circ\text{C}$

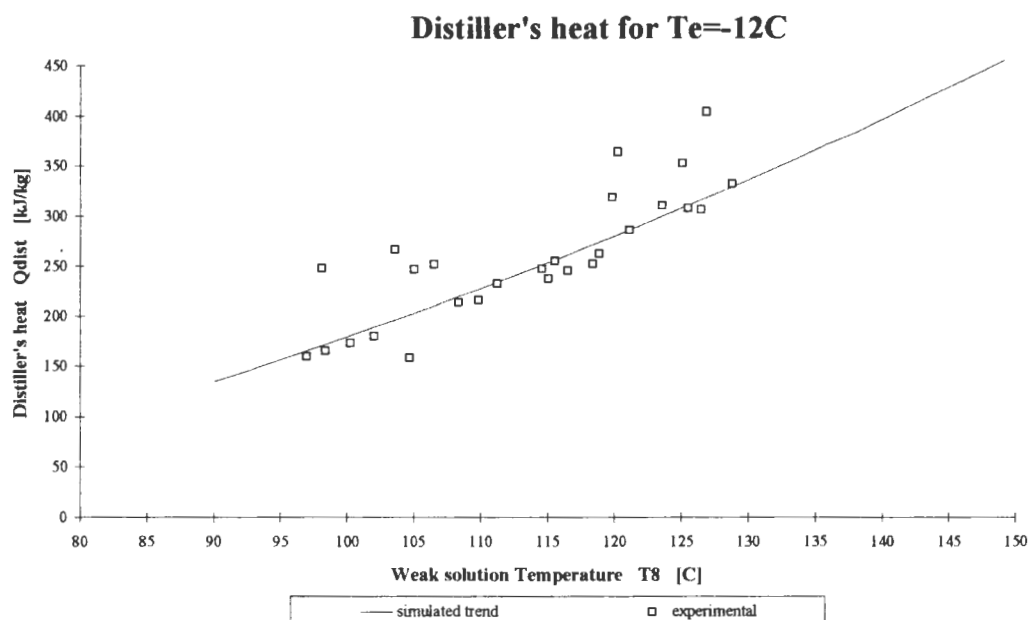


Figure 5. 29 The trend of the distiller's heat rate at a condensate temperature $T_s = 31.67^\circ\text{C}$

5.4.4 The work of the solution-processing pump

The strong solution was processed by an electrically driven, single acting piston pump. The measurement of the power consumption of the motor was not an indication of the power necessary to pump the solution. Thus the work demand of the pump was calculated as the enthalpy increase of the strong solution.

Temperature readings taken before at the inlet and outlet of the pump (Table 5.1 columns T_7 and T_{13}) show an unpredictable pattern. In some readings the temperature increase is small and in others it is negative. For this reason, temperature T_{13} readings after the pump were neglected. The specific volume of the mixture necessary to calculate the work of the pump, was estimated from the temperature T_7 prior to the pump.

5.4.5 Coefficient of performance

The results for the coefficient of performance are shown in Figures 5.31 to 5.34. All graphs indicate a good agreement between the experimental COP and the predicted values.

Figure 5.34 shows a rapid drop in the COP for generator temperatures approaching the minimum temperatures due to the combined effect of the rapid increase in the heat loads in the generator, observed in Figure 5.4 and the absorber, observed in Figure 5.13 respectively.

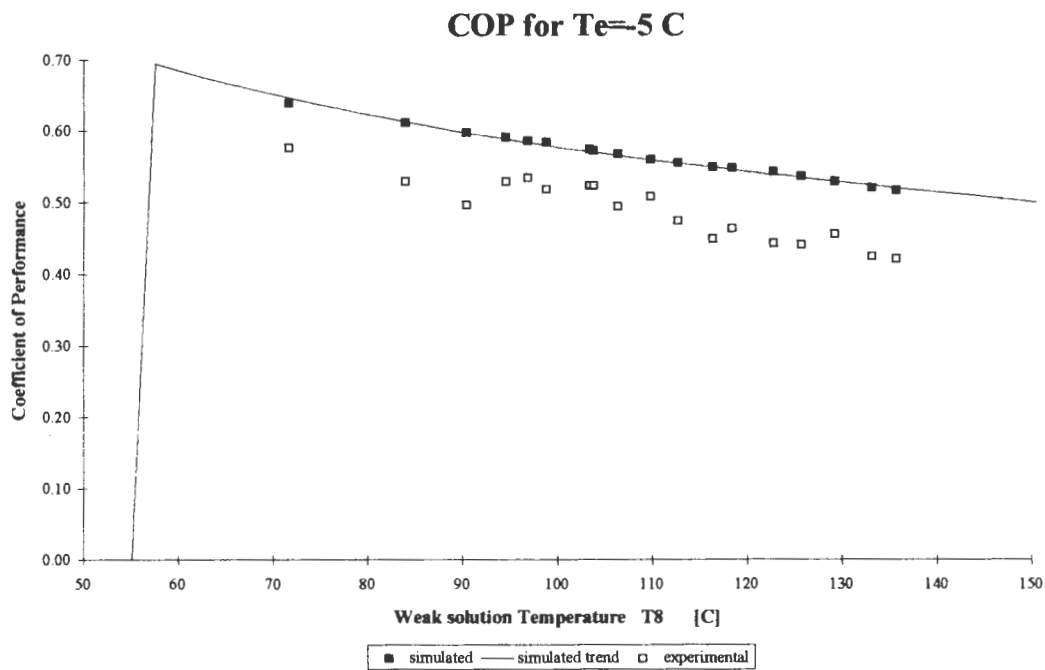


Figure 5. 30 The trend on the COP at a condensate temperature $T_s = 22.14^\circ\text{C}$

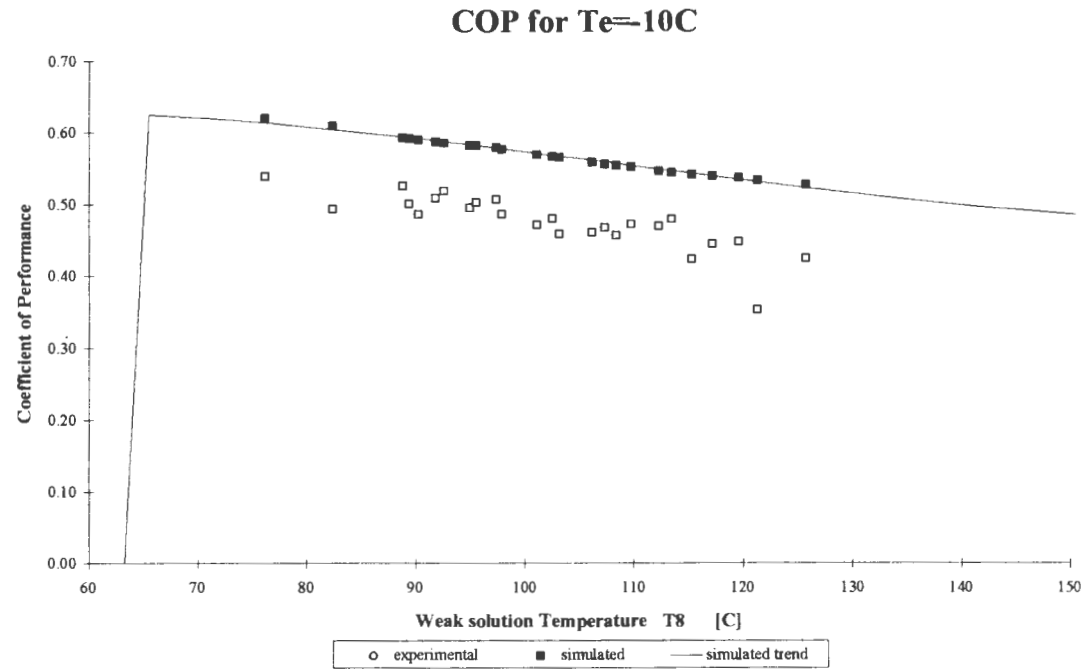


Figure 5. 31 The trend on the COP at a condensate temperature $T_S = 22.43^\circ\text{C}$

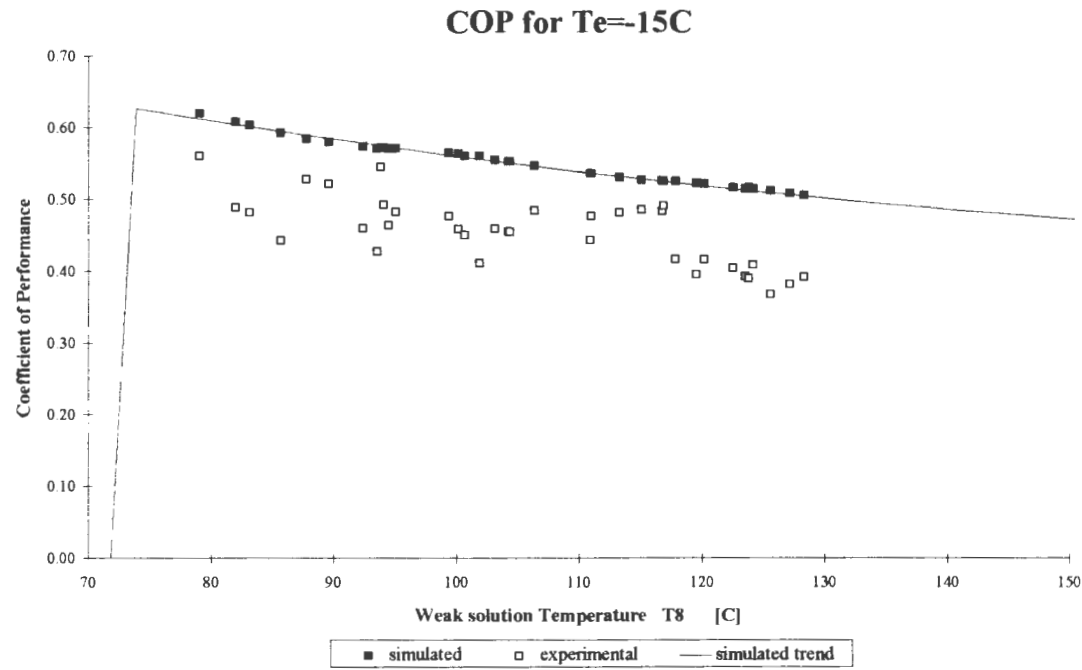


Figure 5. 32 The trend on the COP at a condensate temperature $T_S = 22.78^\circ\text{C}$

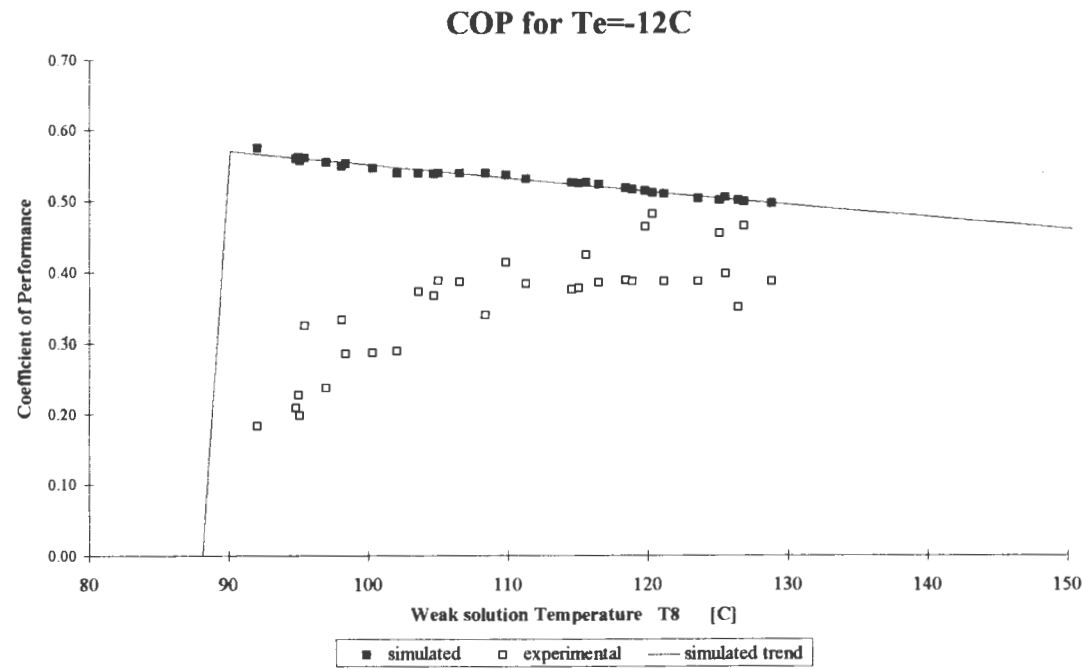


Figure 5. 33 The trend on the COP at a condensate temperature $T_s=31.67^{\circ}\text{C}$

Table 5.1
EXPERIMENTAL RESULTS
 $T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$ (average)

Mref	Phigh	Plow	T_D	T2-T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13	T15	T10eq	Xref	xst	xwe	R.E.	Qin	Wp	Qabs	Qcon	Qdis	T_w^{in}	T_w^{out}	M _w	COP
g/s	bar	bar	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$				W	W	W	W	W	W	$^\circ\text{C}$	$^\circ\text{C}$	lt/min	
0.917	8.7	3.2	60.9	22	8.1	-4.9	20.21	23.91	71.7	31.5	56.35	62.09	61.39	24.63	-5.71	56.61	0.987	0.507	0.422	960	1662	5	1426	1201	7.156	19.00	20.73	10.8	0.576
0.85	8.7	3.3	62.28	22.5	7.97	-5.1	20.04	24.41	83.98	39.2	56.65	67.86	66.34	24.62	-5.93	56.18	0.995	0.509	0.36	834	1574	3	1262	1155	78.15	19.20	20.99	10.9	0.529
0.83	8.7	3.3	62.01	22.4	8.18	-4.8	20.3	24.31	90.45	43.9	56.21	70.56	69.31	24.62	-5.59	56.07	0.992	0.51	0.328	789	1589	2	1252	1132	61.85	19.10	20.90	10.8	0.496
0.893	8.5	3.2	61.73	22.3	8.11	-4.9	20.21	24.21	94.59	46.5	56.36	72.57	71.29	24.62	-5.72	56.08	0.987	0.505	0.304	913	1724	2	1414	1228	53.31	19.20	21.00	10.9	0.529
0.917	8.5	3.3	62.56	22.6	8.11	-4.9	20.23	24.51	96.92	48.1	53.66	73.62	72.28	24.33	-5.7	55.44	0.993	0.509	0.293	978	1816	2	1527	1276	96.43	18.90	20.73	10.9	0.538
0.893	8.7	3.3	61.18	22.1	7.97	-5.1	20.04	24.01	98.85	49.2	53.95	74.68	73.27	24.62	-5.93	55.74	0.995	0.512	0.288	926	1772	2	1460	1249	115.1	18.80	20.64	10.9	0.522
0.871	8.5	3.3	61.7	22.3	7.97	-5.1	20.08	24.21	103.39	52.3	56.64	76.74	75.25	24.62	-5.94	55.11	0.995	0.511	0.263	882	1680	2	1333	1240	131.1	19.00	20.87	10.9	0.524
0.917	8.5	3.3	62.25	22.5	7.97	-5.1	20.08	24.41	103.8	52.6	53.94	76.92	75.25	24.62	-5.94	55.33	0.995	0.509	0.261	981	1808	2	1495	1306	139.4	19.10	20.96	10.9	0.542
0.854	8.5	3.2	63.48	21.9	7.97	-5.1	20.08	23.81	106.4	54.4	56.64	78.09	77.23	24.33	-5.94	55.64	0.989	0.507	0.249	835	1686	2	1317	1212	84.80	19.10	20.98	10.8	0.495
0.871	8.7	3.3	63.21	21.8	7.9	-5.2	19.96	23.71	109.87	56.6	56.8	79.76	78.22	24.33	-6.06	55.43	0.996	0.514	0.238	880	1727	2	1361	1261	160.5	19.10	21.01	10.8	0.509
0.854	8.5	3.2	60.9	22	8.06	-4.97	20.23	23.91	112.7	59.3	56.05	80.53	79.21	24.33	-5.98	55.76	0.989	0.507	0.221	836	1759	2	1370	1234	110.6	19.10	21.00	10.9	0.475
0.83	8.7	3.2	60.9	22	8.05	-4.98	20.2	23.91	116.35	61.5	56.33	82.35	81.19	24.62	-5.71	56.61	0.987	0.507	0.21	779	1720	2	1302	1207	119.1	19.20	21.10	10.9	0.452
0.854	8.7	3.2	63.22	21.8	8.1	-4.9	20.21	23.71	118.4	63	56.35	83.25	82.18	24.62	-5.71	56.39	0.987	0.508	0.201	828	1771	2	1363	1245	116.4	19.20	21.15	10.7	0.467
0.81	8.5	3.2	63.51	21.9	8.04	-5	20.24	23.81	122.82	65.9	56.51	85.29	84.16	24.62	-5.83	55.65	0.988	0.507	0.177	746	1683	1	1239	1200	124.0	19.20	21.20	10.6	0.443
0.81	8.7	3.3	60.9	22	8.11	-4.89	20.25	23.91	125.7	68	56.34	86.43	85.15	23.83	-5.71	55.64	0.993	0.513	0.169	753	1705	1	1256	1214	167.9	19.25	21.28	10.6	0.441
0.83	8.5	3.3	61.17	22.1	7.9	-5.2	19.96	24.01	129.19	70	56.8	88.22	87.13	24.33	-6.06	54.90	0.996	0.512	0.151	798	1744	1	1285	1274	219.4	19.25	21.31	10.6	0.457
0.81	8.7	3.3	61.73	22.3	8.1	-4.9	20.21	24.21	133.2	73.3	56.34	89.58	88.12	24.62	-5.71	55.97	0.993	0.511	0.139	752	1773	1	1297	1244	195.2	19.00	21.07	10.6	0.424
0.81	8.5	3.3	61.17	22.1	7.83	-4.9	19.93	24.01	135.76	74.5	56.97	91	90.1	24.62	-5.88	54.90	0.994	0.512	0.125	757	1739	1	1245	1267	225.2	19.00	21.13	10.5	0.435

Bold-type face indicates measured experimental readings.

Table 5.1 (continued)
EXPERIMENTAL RESULTS
 $T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$ (average)

Mref	Phigh	Plow	T_D	T_2-T_3	T_4	T_5	T_6	T_7	T_8	T_9	T_{10}	T_{11}	T_{12}	T_{13}	T_{15}	T_{10eq}	Xref	xst	xwe	R.E.	Q _{in}	W _p	Q _{abs}	Q _{con}	Q _{dis}	T_w^h	T_w^{out}	M _w	COP
g/s	bar	bar	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$				W	W	W	W	W	W	$^\circ\text{C}$	$^\circ\text{C}$	lit/min	
0.854	8.7	2.6	62.16	22.5	3.28	-9.8	19.14	24.41	76.21	29.1	64.07	68.28	67.33	24.62	-11.42	63.68	0.993	0.465	0.399	843	1559	7	1256	1156	56.473	19.00	21.00	9.7	0.538
0.81	8.7	2.6	61.89	22.4	3.28	-9.8	18.96	24.31	82.41	32.8	64.04	71.22	70.3	24.62	-11.41	63.57	0.992	0.466	0.368	754	1524	4	1178	1109	64.636	19.00	21.01	9.7	0.493
0.917	8.7	2.5	61.61	22.3	3.76	-9.85	18.88	24.21	88.81	36.7	64.14	74.3	73.27	24.62	-11.48	64.68	0.985	0.46	0.336	962	1797	4	1501	1267	65.587	19.00	21.01	9.8	0.534
0.81	8.5	2.6	61.89	22.4	4.4	-9.9	19.12	24.31	89.41	37	64.21	74.62	73.27	24.33	-11.54	62.70	0.993	0.466	0.329	759	1514	3	1148	1135	93.105	19.20	21.23	9.8	0.5
0.854	8.7	2.5	62.16	22.5	4.88	-9.95	19.28	24.41	90.21	37.5	64.33	75.04	74.26	24.62	-11.6	64.90	0.986	0.458	0.33	833	1631	4	1289	1184	62.625	18.90	21.18	8.7	0.509
0.893	8.7	2.5	61.33	22.2	4.4	-9.9	19.12	24.11	91.81	38.5	64.21	75.76	74.26	23.83	-11.54	64.57	0.986	0.46	0.322	914	1713	4	1390	1246	75.091	18.80	21.04	8.9	0.532
0.893	8.5	2.5	61.61	22.3	4.4	-9.9	19.12	24.21	92.61	39	64.21	76.13	75.25	24.33	-11.54	63.81	0.986	0.46	0.314	917	1769	3	1441	1253	77.056	19.00	21.30	8.7	0.517
0.871	8.5	2.5	62.16	22.5	4.4	-9.9	19.12	24.41	94.97	40.5	64.2	77.25	76.24	24.62	-11.54	64.03	0.986	0.458	0.302	870	1756	3	1404	1230	80.679	19.10	21.40	8.8	0.494
0.893	8.5	2.6	61.06	22.1	2.16	-9.7	19	24.01	95.57	41.2	60.87	77.35	76.24	24.62	-11.3	62.37	0.992	0.468	0.3	927	1838	3	1507	1268	105.341	19.20	21.50	8.8	0.503
0.85	8.5	2.6	61.33	22.2	2.64	-9.75	19.19	24.11	97.4	42.2	63.97	78.25	77.23	24.32	-11.36	62.48	0.992	0.467	0.291	836	1645	3	1277	1215	111.142	18.90	20.75	11.0	0.507
0.846	8.7	2.6	62.16	22.5	4.4	-9.9	19.12	24.41	97.9	42.4	64.21	78.62	77.23	24.62	-11.54	63.67	0.993	0.465	0.293	827	1701	3	1331	1208	121.696	18.80	20.64	11.1	0.486
0.85	8.7	2.5	62.42	22.6	7.76	-9.8	19.44	24.51	101.13	44	64.72	80.39	79.21	24.62	-11.76	65.01	0.988	0.458	0.278	828	1737	3	1356	1219	100.109	19.00	21.57	8.0	0.476
0.85	8.5	2.5	61.61	22.3	6.64	-10.1	19.82	24.21	102.53	45	64.54	80.94	79.21	24.62	-11.77	63.81	0.988	0.46	0.267	832	1733	3	1345	1230	109.227	19.10	21.46	8.8	0.479
0.81	8.7	2.6	61.06	22.1	3.28	-9.8	19.14	24.01	103.2	45.9	64.07	80.99	79.21	24.62	-11.42	63.23	0.993	0.468	0.268	755	1646	3	1238	1174	125.444	19.10	21.07	10.5	0.458
0.81	8.5	2.6	61.89	22.4	4.4	-9.9	19.12	24.31	106.2	47.7	64.22	82.46	81.19	24.33	-11.54	62.70	0.993	0.466	0.25	759	1646	2	1226	1192	146.623	19.10	21.36	9.3	0.46
0.858	8.7	2.5	61.59	22.3	8.88	-10.3	19.43	24.21	107.29	47.8	64.86	83.31	82.18	24.62	-12	64.68	0.99	0.46	0.25	849	1816	3	1422	1254	126.264	19.10	21.33	9.4	0.467
0.85	8.7	2.5	62.71	22.7	6.64	-10.1	19.82	24.61	108.4	48.8	64.54	83.63	82.18	24.33	-11.77	65.13	0.988	0.457	0.245	829	1813	3	1411	1242	120.945	19.20	21.57	8.8	0.457
0.871	8.7	2.6	62.71	22.7	3.28	-9.8	19.14	24.61	109.7	50.1	64.07	83.95	82.18	24.62	-11.42	63.90	0.993	0.464	0.239	878	1862	2	1470	1284	155.347	19.20	21.45	9.4	0.471
0.871	8.7	2.6	62.99	22.8	4.4	-9.9	19.12	24.71	112.3	51.6	64.21	85.22	84.16	24.62	-11.54	64.01	0.993	0.463	0.227	880	1879	2	1478	1296	172.953	19.20	21.28	10.2	0.468
0.871	8.7	2.6	62.14	22.5	7.76	-9.9	19.44	24.41	113.45	51.9	64.72	86.01	85.15	24.33	-11.89	63.68	0.996	0.465	0.222	885	1846	2	1440	1309	204.996	19.00	21.45	8.8	0.479
0.81	8.7	2.5	62.16	22.5	5.68	-10.01	19.32	24.41	115.3	53.4	64.41	86.67	85.15	24.62	-11.67	64.90	0.987	0.458	0.214	747	1726	2	1277	1209	139.33	19.00	21.26	9.4	0.432
0.81	8.5	2.6	62.16	22.5	3.28	-9.8	19.14	24.41	117.2	55	64.41	87.31	86.14	24.62	-11.42	62.81	0.993	0.465	0.202	757	1705	2	1249	1228	173.906	19.00	20.95	11.1	0.444
0.85	8.5	2.5	61.89	22.4	4.88	-9.95	19.28	24.31	119.6	56.3	64.31	88.54	87.13	24.33	-11.6	63.92	0.986	0.459	0.191	827	1854	2	1401	1293	166.992	19.00	21.35	9.2	0.446
0.697	8.7	2.5	62.44	22.6	4.4	-9.9	19.12	24.51	121.3	57.5	64.21	89.23	88.12	24.33	-11.54	65.01	0.986	0.458	0.188	542	1529	2	1023	1060	137.882	19.00	20.93	11.3	0.354
0.83	8.7	2.5	61.89	22.4	6.64	-10.1	19.82	24.31	125.76	60.2	64.54	91.4	90.1	24.62	-11.77	64.79	0.988	0.459	0.169	788	1849	2	1370	1283	186.558	19.00	20.99	11.1	0.426

Table 5.1 (continued)
EXPERIMENTAL RESULTS
 $T_E = -15^\circ\text{C}$ and $T_S = 22.78^\circ\text{C}$ (average)

Mref	Phigh	Plow	T_D	T2-T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13	T15	T10eq	Xref	xst	xwe	R.E.	Qin	Wp	Qabs	Qcon	Qdis	T_w^h	T_w^{out}	M_w	COP
g/s	bar	bar	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$				W	W	W	W	W	W	$^\circ\text{C}$	$^\circ\text{C}$	lt/min	
0.917	8.5	2.1	62.09	22.5	1.89	-14.9	20.09	24.41	79.1	24.9	65.75	73.92	72.28	24.62	-17.36	69.44	0.998	0.429	0.38	1123	1983	10	1835	1294	165	19.00	21.07	9.7	0.563
0.942	8.6	2	61.54	22.3	1.74	-14.7	20.21	24.21	82.02	26.8	65.45	75.17	74.26	24.62	-17.13	71.17	0.99	0.422	0.367	1049	2131	10	1884	1310	76	19.20	21.22	9.8	0.49
0.871	8.5	2	62.37	22.6	1.74	-14.7	20.21	24.51	83.2	27.4	68.72	75.74	74.26	24.62	-17.13	71.07	0.99	0.421	0.359	893	1835	8	1522	1219	76	18.90	20.96	9.7	0.484
0.81	8.5	2	62.37	22.6	1.74	-14.7	20.21	24.51	85.7	28.8	68.72	76.96	75.25	22.1	-17.13	71.07	0.99	0.421	0.347	768	1719	6	1355	1143	79	18.80	20.85	9.8	0.445
0.893	8.5	2.1	62.64	22.7	1.89	-14.9	20.09	24.61	87.82	29.8	62.62	78.15	77.23	24.62	-17.36	69.67	0.998	0.428	0.337	1094	2056	6	1876	1294	192	19.00	21.10	9.8	0.531
0.85	8.7	2.1	62.37	22.6	1.89	-14.9	20.09	24.51	89.62	30.8	62.62	79.02	78.22	23.83	-17.36	70.44	0.998	0.429	0.332	1038	1969	5	1795	1232	185	19.20	21.58	8.7	0.526
0.85	8.5	2	62.64	22.7	1.74	-14.7	20.21	24.61	92.42	32.7	68.72	80.19	79.21	24.33	-17.13	71.18	0.99	0.42	0.315	848	1828	5	1463	1225	108	18.90	21.22	8.9	0.463
0.889	8.5	1.9	62.64	22.7	2.02	-15.1	20.05	24.61	93.6	32.9	72.84	81.1	80.2	24.33	-17.6	72.75	0.87	0.412	0.309	751	1747	4	1161	1342	110	19.20	21.42	9.7	0.429
0.81	8.6	2.1	62.64	22.7	1.81	-14.8	20.18	24.61	93.9	33.4	65.6	80.99	79.21	24.62	-17.25	70.11	0.998	0.428	0.31	986	1805	4	1619	1191	186	18.90	21.03	9.8	0.545
0.792	8.5	2.1	62.64	22.7	1.67	-14.6	20.23	24.61	94.1	33.8	72	80.93	79.21	24.62	-17.02	69.67	0.996	0.428	0.306	953	1494	4	1301	1162	167	18.80	20.93	9.8	0.636
0.814	8.6	2	62.64	22.7	1.74	-14.7	20.21	24.61	94.5	33.9	72.16	81.19	80.2	24.63	-17.13	71.62	0.99	0.42	0.307	774	1665	5	1273	1178	108	19.20	21.58	8.7	0.463
0.792	8.5	2.1	62.92	22.8	1.6	-14.5	20.21	24.71	95.1	34.5	71.83	81.31	80.2	24.33	-16.9	69.78	0.996	0.427	0.302	738	1522	4	1112	1163	162	19.20	21.36	9.7	0.484
0.893	8.5	2.1	61.81	22.4	1.67	-14.6	20.23	24.31	99.4	36.9	65.31	83.44	82.18	24.33	-17.02	69.33	0.996	0.43	0.282	1075	1883	4	1646	1330	206	18.90	21.06	9.8	0.57
0.774	8.5	2.1	61.81	22.4	1.46	-14.3	20.2	24.31	100.2	37.8	71.5	83.56	82.18	24.33	-16.67	69.33	0.994	0.43	0.278	702	1524	3	1091	1149	157	19.20	21.35	9.8	0.46
0.792	8.7	2	62.31	22.6	2.17	-15.3	20.11	24.51	100.7	36.8	73.12	84.63	83.17	23.83	-17.83	71.95	0.995	0.421	0.28	735	1627	4	1202	1177	167	18.90	21.34	8.7	0.451
0.85	8.7	2	62.71	22.7	1.43	-14.3	20.2	24.61	101.9	38.79	71.72	84.47	83.17	24.62	-16.67	72.06	0.887	0.42	0.274	703	1664	3	1078	1296	127	18.80	21.25	8.9	0.422
0.893	8.7	2	62.31	22.6	2.1	-14.85	20.14	24.51	103.2	38.4	66.16	85.72	84.16	24.62	-17.81	71.95	0.995	0.421	0.268	947	2001	4	1629	1337	196	19.20	21.40	9.7	0.472
0.893	8.7	2	62.64	22.7	1.81	-14.8	20.18	24.61	104.3	39.6	65.6	85.92	84.16	24.62	-17.25	72.06	0.991	0.42	0.263	940	2008	4	1634	1329	159	18.90	21.06	9.8	0.467
0.893	8.7	2	62.92	22.8	1.89	-14.9	20.09	24.71	104.4	39.5	65.75	86.04	85.15	24.62	-17.36	72.17	0.992	0.42	0.263	941	2012	4	1637	1332	167	18.80	20.96	9.8	0.467
0.893	8.7	2	62.64	22.7	1.89	-14.9	20.09	24.61	106.4	40.7	72.49	86.97	85.15	24.33	-17.36	72.06	0.992	0.42	0.254	941	1939	4	1556	1340	175	19.00	21.46	8.7	0.484
0.85	8.7	2	62.82	22.8	1.85	-14.8	20.15	24.71	110.97	43.6	72.07	88.86	87.13	24.62	-17.25	72.17	0.991	0.42	0.233	846	1886	3	1457	1292	177	19.20	21.64	8.9	0.448
0.893	8.6	2	62.92	22.8	1.81	-14.8	20.18	24.71	111.01	43.6	72.32	89.02	88.12	24.62	-17.25	71.73	0.991	0.42	0.231	941	1974	3	1568	1364	190	18.90	21.41	8.7	0.476
0.871	8.7	2.1	63.19	22.9	1.74	-14.7	20.21	24.81	113.31	45.1	56.54	89.97	88.12	24.62	-17.13	70.78	0.997	0.427	0.223	1052	2163	3	1886	1352	260	19.20	21.73	8.8	0.485
0.792	8.5	2.1	60.97	23.1	1.89	-14.9	20.09	25.01	115.11	46	56.8	90.96	89.11	23.83	-17.36	70.12	0.998	0.426	0.211	970	1966	3	1710	1248	265	19.20	21.75	8.8	0.493
0.83	8.8	2.1	63.37	22.9	1.82	-14.9	20.34	24.81	116.76	46.8	57.19	91.98	90.1	23.83	-17.36	71.22	0.998	0.427	0.21	1013	2066	3	1797	1307	278	18.90	20.94	11.0	0.49
0.85	8.7	2.1	60.97	23.1	1.89	-14.9	20.09	25.01	116.86	47	59.64	91.75	90.1	24.62	-17.36	71.00	0.998	0.426	0.207	1038	2082	3	1805	1340	286	18.80	20.83	11.1	0.498
0.85	8.6	2	62.92	22.8	1.81	-14.8	20.18	24.71	117.86	47.8	65.6	92.13	91.09	24.33	-17.25	71.73	0.991	0.42	0.201	848	2003	3	1540	1328	210	19.20	22.00	8.0	0.423
0.81	8.6	2	62.92	22.8	1.89	-14.9	20.09	24.71	119.56	48.7	65.74	92.97	91.09	24.33	-17.36	71.73	0.992	0.42	0.194	768	1915	3	1425	1276	214	18.90	21.47	8.8	0.401
0.85	8.6	2	62.92	22.8	1.89	-14.9	20.09	24.71	120.21	49.1	65.76	93.27	92.08	24.62	-17.36	71.73	0.992	0.42	0.191	849	2012	3	1538	1342	228	18.80	20.95	10.5	0.421
0.83	8.6	2	63.12	22.9	2.24	-15.4	20.12	24.81	122.55	49.8	66.44	94.73	93.07	24.62	-17.95	71.85	0.995	0.419	0.181	813	1986	3	1486	1337	277	19.00	21.47	9.3	0.409
0.81	8.7	2	62.87	22.8	2.1	-14.91	20.14	24.71	123.55	50.7	66.16	95.01	94.06	24.62	-17.89	72.17	0.995	0.42	0.178	772	1944	3	1435	1304	267	19.20	21.66	9.4	0.397
0.81	8.6	2	63.19	22.9	1.89	-14.9	20.09	24.81	123.8	51.3	65.75	94.88	93.07	24.62	-17.36	71.85	0.992	0.419	0.175	768	1941	3	1430	1298	235	18.90	21.50	8.8	0.395
0.85	8.7	2	60.71	23	1.81	-14.8	20.18	24.91	124.2	51.7	65.6	94.97	93.07	24.33	-17.25	72.40	0.991	0.418	0.176	846	2040	3	1550	1356	237	18.80	21.24	9.4	0.414
0.774	8.7	2	61.24	23.2	1.89	-14.9	20.09	25.11	125.6	52.4	65.75	95.69	94.06	24.62	-17.36	72.62	0.992	0.417	0.17	697	1872	2	1343	1246	230	19.00	20.96	11.7	0.372
0.81	8.7	2	62.03	23.5	1.89	-14.9	20.09	25.41	127.2	53.4	65.75	96.41	95.05	24.33	-17.36	72.96	0.992	0.415	0.163	767	1981	2	1455	1313	249	19.10	21.10	11.6	0.387
0.83	8.6	2	61.67	23.4	1.92	-14.9	20.09	25.31	128.34	54.1	65.52	96.77	95.05	24.33	-17.36	72.41	0.992	0.416	0.156	807	2038	2	1513	1353	263	19.10	21.09	11.7	0.396

Table 5.1 (continued)
EXPERIMENTAL RESULTS
 $T_E = -12^\circ\text{C}$ and $T_S = 31.67^\circ\text{C}$ (average)

Mref	Phigh	Plow	T_D	T2-T3	T4	T5	T6	T7	T8	T9	T10	T11	T12	T13	T15	T10eq	Xref	xst	xwe	R.E.	Qin	Wp	Qabs	Qcon	Qdis	$T_{in,w}$	$T_{out,w}$	M_w	COP
g/s	bar	bar	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$				W	W	W	W	W	W	$^\circ\text{C}$	$^\circ\text{C}$	lit/min	
0.85	11.8	2.3	67.9	31	3.72	-11.93	29.40	31.48	92.08	32.0	60.18	89.31	88.12	28.81	-13.9	87.55	0.99	0.403	0.38	840	4584	31	4264	1205	122	24.60	26.53	10.5	0.182
0.875	11.9	2.3	69.22	32.2	4.11	-11.93	30.00	32.68	94.85	33.1	62.09	92.13	91.09	29.74	-13.9	89.26	0.99	0.396	0.369	897	4273	27	3955	1258	138	23.90	26.01	9.7	0.209
0.88	11.6	2.3	68.37	32.2	3.89	-11.69	29.35	32.68	95.03	33.8	59.64	90.61	89.11	28.73	-13.63	88.21	0.987	0.396	0.363	903	3967	21	3644	1261	133	24.60	26.68	9.8	0.226
0.875	12.1	2.3	69.22	32.2	3.99	-12.09	30.09	32.68	95.11	33.1	62.28	92.39	91.09	29.28	-14.09	89.95	0.991	0.396	0.371	898	4524	29	4210	1259	146	23.90	25.99	9.8	0.197
0.875	11.7	2.4	69.00	31.5	3.72	-11.93	29.40	31.98	95.41	33.8	61.06	90.95	89.11	28.81	-13.9	86.39	0.996	0.407	0.362	1064	3254	16	3083	1274	199	24.60	26.74	9.7	0.325
0.871	11.9	2.3	69.95	31.7	3.95	-11.93	29.78	32.18	97.00	34.4	57.79	92.59	91.09	29.63	-13.9	88.69	0.99	0.399	0.358	887	3723	18	3389	1255	138	23.90	25.99	9.8	0.237
0.88	11.9	2.4	69.22	32.2	4.11	-11.93	30.00	32.68	98.13	34.9	62.10	93.74	92.08	28.82	-13.9	87.88	0.996	0.403	0.353	1071	3192	15	3005	1298	217	24.50	26.64	9.8	0.334
0.871	11.6	2.3	68.39	31.2	3.89	-11.69	29.35	31.68	98.37	35.6	59.89	92.24	91.09	28.79	-13.63	87.07	0.987	0.402	0.347	884	3079	13	2731	1261	143	24.60	26.73	9.7	0.286
0.88	12	2.3	69.51	31.5	3.95	-11.93	29.78	31.98	100.3	36.2	61.27	94.2	93.07	29.63	-13.9	88.81	0.990	0.400	0.344	905	3136	14	2793	1279	152	23.90	26.01	9.8	0.287
0.867	11.7	2.3	68.75	32.4	3.72	-11.93	29.40	32.88	102.06	37.4	60.18	94.2	93.07	28.79	-13.9	88.78	0.99	0.395	0.331	877	3012	12	2651	1268	155	24.60	26.72	9.8	0.29
0.871	11.9	2.4	70.40	31.9	3.95	-11.93	29.78	32.38	103.6	38.0	59.91	95.8	94.06	29.54	-13.9	87.54	0.996	0.405	0.327	1059	2815	10	2608	1303	231	23.90	26.36	8.7	0.375
1.001	11.9	2.3	69.22	32.2	4.11	-11.93	30.00	32.68	104.7	33.1	62.09	92.13	91.09	29.74	-13.9	89.26	0.99	0.396	0.322	1194	3231	12	3016	1439	158	24.60	26.91	8.9	0.368
0.871	11.6	2.4	67.95	32	3.89	-11.69	29.35	32.48	105.05	39.3	59.89	95.47	94.06	28.7	-13.63	86.61	0.995	0.404	0.315	1049	2678	9	2457	1303	214	23.90	26.10	9.7	0.39
0.867	12.1	2.4	69.95	31.7	4.11	-11.69	29.88	32.18	106.57	39.9	61.0	97.04	96.04	29.54	-13.63	88.00	0.995	0.406	0.316	1042	2667	9	2447	1297	217	23.90	26.07	9.8	0.389
0.871	11.6	2.3	67.95	31	3.89	-11.69	29.35	31.48	108.39	41.1	59.89	97.08	96.04	28.7	-13.63	86.84	0.987	0.403	0.300	884	2579	8	2184	1306	185	24.60	26.78	9.8	0.342
0.88	12	2.3	69.51	31.5	4.11	-11.69	29.19	31.98	109.88	41.8	79.61	98.62	97.03	29.54	-13.63	88.81	0.987	0.400	0.300	897	2145	8	1751	1321	189	23.90	26.37	8.7	0.417
0.867	11.8	2.3	69.65	32.4	4.11	-11.93	29.48	32.88	111.27	42.1	75.19	100.1	99.01	28.82	-13.9	89.13	0.990	0.395	0.290	877	2261	8	1848	1320	201	24.50	26.74	9.7	0.386
0.871	12.1	2.3	69.22	32.2	4.11	-11.93	29.48	32.68	114.56	44.0	73.01	101.66	100	28.82	-13.9	89.95	0.990	0.396	0.280	882	2327	7	1907	1335	214	24.60	26.82	9.8	0.378
0.867	11.7	2.3	69.01	32.5	3.89	-11.69	28.85	32.98	115.07	45.0	72.52	100.23	99.01	28.7	-13.63	88.90	0.987	0.394	0.271	871	2288	7	1862	1325	205	23.90	26.12	9.8	0.379
0.903	11.7	2.3	68.34	29.2	3.61	-12.09	28.90	29.68	115.56	44.8	72.76	100.77	99.01	28.81	-14.09	85.15	0.991	0.413	0.269	957	2234	6	1834	1389	229	23.90	26.43	8.7	0.427
0.867	11.9	2.3	69.95	31.7	4.11	-11.69	29.37	32.18	116.51	45.5	73.87	101.75	100	29.74	-13.63	88.69	0.987	0.399	0.268	871	2239	6	1807	1333	212	24.60	26.86	9.7	0.388
0.875	11.7	2.3	69.27	31.6	3.89	-11.69	28.85	32.08	118.41	46.9	72.53	101.79	100	28.7	-13.63	87.88	0.987	0.399	0.257	889	2270	6	1835	1354	220	23.90	26.15	9.8	0.391
0.871	12.1	2.3	67.87	31	3.61	-12.09	28.90	31.48	118.87	46.8	73.01	102.32	100.99	28.82	-14.09	88.58	0.991	0.403	0.261	882	2255	6	1826	1344	228	24.50	26.74	9.8	0.39
0.875	12	2.4	69.95	31.7	4.11	-11.69	29.37	32.18	119.83	47.5	73.87	103.31	101.98	29.74	-13.63	87.66	0.995	0.406	0.255	1049	2244	6	1956	1377	278	24.60	27.18	8.7	0.466
0.88	12	2.4	69.95	31.7	3.83	-12.09	29.31	32.18	120.3	47.3	77.48	103.87	101.98	28.75	-14.09	87.66	0.997	0.406	0.253	1070	2199	6	1914	1401	319	23.90	26.24	9.7	0.485
0.875	12	2.3	69.22	32.2	4.11	-11.93	29.48	32.68	121.13	47.7	75.17	104.75	102.97	28.82	-13.9	89.60	0.990	0.396	0.250	893	2287	6	1834	1381	249	24.60	26.89	9.8	0.389
0.871	11.8	2.3	70.18	31.8	3.83	-12.09	29.31	32.28	123.59	49.2	74.41	105.42	103.96	28.7	-14.09	88.45	0.991	0.398	0.236	888	2272	5	1802	1393	269	23.90	26.22	9.8	0.39
0.875	11.6	2.4	68.8	32.4	3.89	-11.69	28.85	32.88	125.09	50.8	72.52	104.92	102.97	28.7	-13.63	87.06	0.995	0.402	0.226	1050	2288	5	1965	1412	308	24.60	27.25	8.7	0.458
0.875	11.9	2.3	69.04	29.5	3.61	-12.09	28.90	29.98	125.51	50.6	72.76	105.44	103.96	28.81	-14.09	86.19	0.991	0.411	0.229	894	2227	5	1760	1396	268	23.90	26.47	8.9	0.4
0.871	11.8	2.2	69.95	31.7	4.11	-11.69	29.37	32.18	126.46	51.3	81.43	106.42	104.95	29.74	-13.63	89.77	0.855	0.392	0.223	705	1994	4	1266	1450	266	24.50	27.23	8.7	0.353
0.875	12	2.4	69.95	31.7	3.83	-12.09	29.31	32.18	126.88	51.1	74.41	106.97	104.95	29.63	-14.09	87.66	0.997	0.406	0.225	1065	2271	5	1951	1433	353	24.60	27.27	8.8	0.468
0.88	11.7	2.3	69.18	31.6	3.61	-12.09	28.90	32.08	128.82	52.5	72.12	107	105.94	28.81	-14.09	87.88	0.991	0.399	0.211	906	2317	5	1831	1429	292	25.00	27.63	8.8	0.39

Table 5.2															
SIMULATED PERFORMANCE															
$T_E = -5^\circ\text{C}$ and $T_S = 22.14^\circ\text{C}$ (average)															
No	Phigh	Plow	T2-T3	T8	T15	Xref	xst	xwe	R.E.	Qin	Qp	Qabs	Qcon	Qdis	COP
	bar	bar	°C	°C	°C				kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	
1	8.84	3.25	22.00	71.70	-5.71	0.990	0.523	0.425	1133.84	1772.74	4.06	1602.55	1298.67	9.42	0.638
2	8.98	3.22	22.50	83.98	-5.93	0.990	0.518	0.366	1136.02	1855.85	2.98	1654.15	1296.48	44.22	0.611
3	8.95	3.26	22.40	90.45	-5.59	0.990	0.521	0.334	1134.10	1895.41	2.50	1677.80	1297.05	57.16	0.598
4	8.93	3.25	22.30	94.59	-5.72	0.990	0.521	0.313	1134.88	1918.08	2.33	1689.97	1297.46	67.87	0.591
5	9.01	3.25	22.60	96.92	-5.70	0.990	0.519	0.304	1132.82	1931.90	2.31	1695.16	1296.24	75.63	0.586
6	8.87	3.22	22.10	98.85	-5.93	0.990	0.52	0.292	1136.43	1943.07	2.17	1702.10	1298.27	81.30	0.584
7	8.93	3.22	22.30	103.39	-5.94	0.990	0.519	0.272	1135.99	1978.07	2.09	1719.53	1297.46	99.16	0.574
8	8.98	3.22	22.50	103.80	-5.94	0.990	0.518	0.272	1135.88	1982.86	2.12	1722.39	1296.48	101.98	0.572
9	8.81	3.22	21.90	106.40	-5.94	0.990	0.522	0.256	1136.57	2000.81	1.94	1731.01	1299.08	109.23	0.568
10	8.78	3.20	21.80	109.87	-6.06	0.990	0.521	0.24	1137.60	2030.27	1.87	1746.67	1299.48	123.59	0.56
11	8.84	3.21	22.00	112.70	-5.98	0.990	0.521	0.228	1138.41	2049.03	1.84	1755.73	1298.67	134.88	0.555
12	8.84	3.25	22.00	116.35	-5.71	0.990	0.523	0.212	1133.84	2061.90	1.76	1752.74	1298.67	146.09	0.549
13	8.78	3.25	21.80	118.40	-5.71	0.990	0.524	0.202	1134.13	2068.53	1.70	1752.71	1299.48	152.17	0.548
14	8.81	3.23	21.90	122.82	-5.83	0.990	0.523	0.184	1137.08	2094.04	1.67	1764.45	1299.08	169.26	0.543
15	8.84	3.25	22.00	125.70	-5.71	0.990	0.523	0.172	1133.84	2111.81	1.64	1768.01	1298.67	180.61	0.536
16	8.87	3.20	22.10	129.19	-6.06	0.990	0.519	0.158	1137.17	2149.42	1.64	1789.62	1298.27	200.34	0.529
17	8.93	3.25	22.30	133.20	-5.71	0.990	0.521	0.143	1133.41	2177.92	1.60	1798.31	1297.46	217.17	0.52
18	8.87	3.23	22.10	135.76	-5.88	0.990	0.521	0.132	1138.07	2199.22	1.56	1812.15	1298.27	228.44	0.517

Table 5.2 (continued)															
SIMULATED PERFORMANCE															
$T_E = -10^\circ\text{C}$ and $T_S = 22.43^\circ\text{C}$ (average)															
No	Phigh	Plow	T2-T3	T8	T15	Xref	xst	xwe	R.E.	Qin	Qp	Qabs	Qcon	Qdis	COP
	bar	bar	°C	°C	°C				kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	
19	8.99	2.56	22.50	76.21	-11.42	0.990	0.475	0.405	1171.74	1886.84	6.83	1722.75	1295.91	46.76	0.619
20	8.96	2.57	22.40	82.41	-11.41	0.990	0.475	0.373	1171.87	1921.95	4.89	1741.26	1296.31	61.14	0.608
21	8.93	2.56	22.30	88.81	-11.48	0.990	0.476	0.341	1172.38	1979.08	3.90	1778.88	1296.72	79.77	0.591
22	8.96	2.55	22.40	89.41	-11.54	0.990	0.475	0.339	1172.31	1984.25	3.90	1781.63	1296.31	82.52	0.59
23	8.99	2.54	22.50	90.21	-11.60	0.990	0.473	0.336	1172.30	1990.76	3.88	1784.91	1295.91	86.12	0.588
24	8.90	2.55	22.20	91.81	-11.54	0.990	0.476	0.326	1172.60	2000.04	3.57	1788.45	1297.13	90.64	0.585
25	8.93	2.55	22.30	92.61	-11.54	0.990	0.475	0.323	1172.46	2005.09	3.54	1790.17	1296.72	94.20	0.584
26	8.99	2.55	22.50	94.97	-11.54	0.990	0.474	0.313	1172.17	2019.27	3.43	1794.11	1295.91	104.86	0.58
27	8.87	2.58	22.10	95.57	-11.30	0.990	0.478	0.308	1171.64	2017.88	3.19	1791.24	1297.69	103.77	0.58
28	8.90	2.57	22.20	97.40	-11.36	0.990	0.477	0.3	1172.13	2029.92	3.12	1795.50	1297.29	112.37	0.577
29	8.99	2.55	22.50	97.90	-11.54	0.990	0.474	0.299	1172.17	2036.37	3.22	1798.54	1295.91	117.32	0.575
30	9.01	2.53	22.60	101.13	-11.76	0.990	0.472	0.285	1173.08	2061.35	3.10	1809.59	1295.50	132.45	0.568
31	8.93	2.53	22.30	102.53	-11.77	0.990	0.473	0.276	1173.07	2072.57	2.94	1815.24	1296.72	136.61	0.565
32	8.87	2.56	22.10	103.20	-11.42	0.990	0.477	0.272	1172.20	2073.83	2.79	1815.19	1297.69	135.94	0.564
33	8.96	2.55	22.40	106.20	-11.54	0.990	0.475	0.26	1172.31	2101.02	2.76	1830.72	1296.31	149.07	0.557
34	8.93	2.50	22.30	107.29	-12.00	0.990	0.472	0.255	1173.94	2116.22	2.76	1840.92	1296.72	155.28	0.554
35	9.04	2.53	22.70	108.40	-11.77	0.990	0.471	0.252	1172.48	2122.37	2.79	1843.02	1295.09	159.52	0.552
36	9.04	2.56	22.70	109.70	-11.42	0.990	0.473	0.246	1171.45	2128.47	2.69	1845.39	1295.09	162.13	0.55
37	9.07	2.55	22.80	112.30	-11.54	0.990	0.472	0.235	1171.73	2149.57	2.64	1855.78	1294.69	173.47	0.544
38	8.99	2.51	22.50	113.45	-11.89	0.990	0.471	0.228	1173.53	2161.07	2.57	1861.85	1295.91	179.42	0.542
39	8.99	2.54	22.50	115.30	-11.67	0.990	0.473	0.22	1172.91	2168.92	2.49	1862.34	1295.91	186.07	0.54
40	8.99	2.56	22.50	117.20	-11.42	0.990	0.475	0.212	1171.74	2175.64	2.41	1860.75	1295.91	193.12	0.538
41	8.96	2.54	22.40	119.60	-11.60	0.990	0.474	0.201	1172.44	2190.67	2.35	1863.22	1296.31	205.93	0.535
42	9.01	2.55	22.60	121.30	-11.54	0.990	0.473	0.195	1172.02	2200.74	2.34	1864.57	1295.50	215.03	0.532
43	8.96	2.53	22.40	125.76	-11.77	0.990	0.473	0.174	1172.92	2233.50	2.23	1875.36	1296.31	236.98	0.525

Table 5.2 (continued)															
SIMULATED PERFORMANCE															
$T_E = -15^\circ\text{C}$ and $T_S = 22.67^\circ\text{C}$ (average)															
No	Phigh	Plow	T2-T3	T8	T15	Xref	xst	xwc	R.E.	Qin	Qp	Qabs	Qcon	Qdis	COP
	bar	bar	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$				kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	
44	9.00	1.98	22.50	79.10	-17.36	0.990	0.431	0.391	1191.30	1912.71	13.33	1744.21	1295.20	77.94	0.619
45	8.94	2.00	22.30	82.02	-17.13	0.990	0.434	0.375	1190.85	1951.35	9.24	1768.90	1296.01	86.53	0.607
46	9.02	2.00	22.60	83.20	-17.13	0.990	0.432	0.371	1190.41	1965.77	9.09	1778.24	1294.79	92.24	0.603
47	9.02	2.00	22.60	85.70	-17.13	0.990	0.432	0.359	1190.41	2002.29	7.71	1802.89	1294.79	102.73	0.592
48	9.05	1.98	22.70	87.82	-17.36	0.990	0.43	0.349	1191.01	2033.09	7.16	1822.91	1294.38	113.96	0.584
49	9.02	1.98	22.60	89.62	-17.36	0.990	0.43	0.339	1191.15	2052.09	6.44	1833.53	1294.79	121.36	0.579
50	9.05	2.00	22.70	92.42	-17.13	0.990	0.432	0.327	1190.27	2070.35	5.70	1839.83	1294.38	132.11	0.573
51	9.05	1.96	22.70	93.60	-17.60	0.990	0.428	0.321	1191.47	2084.33	5.67	1847.48	1294.38	139.61	0.57
52	9.05	1.99	22.70	93.90	-17.25	0.990	0.431	0.319	1190.63	2080.59	5.45	1843.54	1294.38	138.74	0.571
53	9.05	2.01	22.70	94.10	-17.02	0.990	0.432	0.318	1190.08	2078.27	5.32	1841.13	1294.38	138.16	0.571
54	9.05	2.00	22.70	94.50	-17.13	0.990	0.432	0.317	1190.27	2081.85	5.29	1842.68	1294.38	140.34	0.57
55	9.08	2.02	22.80	95.10	-16.90	0.990	0.433	0.314	1189.45	2082.15	5.16	1841.07	1293.98	141.72	0.57
56	8.97	2.01	22.40	99.40	-17.02	0.990	0.434	0.292	1190.52	2105.92	4.36	1848.27	1295.60	156.92	0.564
57	8.97	2.04	22.40	100.20	-16.67	0.990	0.437	0.288	1189.42	2107.11	4.18	1847.24	1295.60	157.85	0.563
58	9.03	1.94	22.60	100.70	-17.83	0.990	0.427	0.287	1192.46	2127.03	4.56	1861.87	1294.57	167.61	0.559
59	9.05	2.04	22.70	101.90	-16.67	0.990	0.435	0.282	1188.98	2120.61	4.15	1853.93	1294.38	165.43	0.56
60	9.03	1.94	22.60	103.20	-17.81	0.990	0.427	0.275	1192.21	2149.18	4.27	1873.06	1294.57	178.03	0.554
61	9.05	1.99	22.70	104.30	-17.25	0.990	0.431	0.271	1190.63	2151.88	4.07	1872.83	1294.38	179.36	0.552
62	9.08	1.98	22.80	104.40	-17.36	0.990	0.429	0.271	1190.86	2154.69	4.13	1874.62	1293.98	181.08	0.552
63	9.05	1.98	22.70	106.40	-17.36	0.990	0.43	0.261	1191.01	2176.48	3.91	1886.89	1294.38	190.12	0.546
64	9.08	1.99	22.80	110.97	-17.25	0.990	0.43	0.241	1190.48	2222.71	3.60	1909.77	1293.98	213.03	0.535
65	9.08	1.99	22.80	111.01	-17.25	0.990	0.43	0.241	1190.48	2223.09	3.60	1909.94	1293.98	213.25	0.535
66	9.11	2.00	22.90	113.31	-17.13	0.990	0.43	0.231	1189.98	2240.46	3.47	1916.06	1293.57	224.28	0.53
67	9.17	1.98	23.10	115.11	-17.36	0.990	0.427	0.225	1190.43	2257.69	3.47	1923.17	1292.75	235.67	0.526
68	9.11	1.98	22.90	116.76	-17.36	0.990	0.429	0.216	1190.72	2265.38	3.32	1923.24	1293.57	242.61	0.525
69	9.17	1.98	23.10	116.86	-17.36	0.990	0.427	0.217	1190.43	2267.50	3.38	1924.53	1292.75	244.02	0.524
70	9.08	1.99	22.80	117.86	-17.25	0.990	0.43	0.211	1190.48	2268.23	3.23	1921.35	1293.98	246.61	0.524
71	9.08	1.98	22.80	119.56	-17.36	0.990	0.429	0.203	1190.86	2279.05	3.17	1922.68	1293.98	256.43	0.522
72	9.08	1.98	22.80	120.21	-17.36	0.990	0.429	0.201	1190.86	2282.75	3.14	1922.71	1293.98	260.06	0.521
73	9.12	1.93	22.90	122.55	-17.95	0.990	0.424	0.191	1192.14	2310.53	3.15	1931.33	1293.35	281.15	0.515
74	9.09	1.93	22.80	123.55	-17.89	0.990	0.425	0.186	1192.32	2316.48	3.08	1931.71	1293.75	286.42	0.514
75	9.11	1.98	22.90	123.80	-17.36	0.990	0.429	0.186	1190.72	2308.89	3.02	1926.30	1293.57	282.76	0.515
76	9.14	1.99	23.00	124.20	-17.25	0.990	0.429	0.185	1190.19	2310.82	3.02	1926.06	1293.16	284.80	0.514
77	9.21	1.98	23.20	125.60	-17.36	0.990	0.427	0.18	1190.24	2326.59	3.03	1931.26	1292.12	296.48	0.511
78	9.29	1.98	23.50	127.20	-17.36	0.990	0.425	0.175	1189.81	2343.38	3.05	1936.96	1290.90	308.38	0.507
79	9.26	1.98	23.40	128.34	-17.36	0.990	0.426	0.17	1189.95	2354.23	2.99	1941.37	1291.31	314.50	0.505

Table 5.2 (continued)															
SIMULATED PERFORMANCE															
$T_E = -12^\circ\text{C}$ and $T_S = 31.78^\circ\text{C}$ (average)															
No	Phigh	Plow	T2-T3	T8	T15	Xref	xst	xwe	R.E.	Qin	Qp	Qabs	Qcon	Qdis	COP
	bar	bar	$^\circ\text{C}$	$^\circ\text{C}$	$^\circ\text{C}$				kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	kJ/kg	
80	11.71	2.31	31.00	92.08	-13.90	0.990	0.406	0.379	1168.66	2009.13	27.01	1801.15	1259.26	144.38	0.574
81	12.13	2.31	32.20	94.85	-13.90	0.990	0.399	0.372	1167.08	2057.32	29.18	1839.19	1253.88	160.50	0.559
82	12.13	2.33	32.20	95.03	-13.63	0.990	0.401	0.372	1166.29	2053.61	26.42	1832.68	1253.88	159.76	0.561
83	12.13	2.29	32.20	95.11	-14.09	0.990	0.398	0.371	1167.77	2066.92	29.38	1847.59	1253.88	162.60	0.557
84	11.88	2.31	31.50	95.41	-13.90	0.990	0.403	0.366	1168.14	2066.84	20.39	1838.44	1256.96	159.96	0.56
85	11.95	2.31	31.70	97.00	-13.90	0.990	0.402	0.359	1167.84	2091.53	18.18	1854.42	1256.08	167.03	0.554
86	12.13	2.31	32.20	98.13	-13.90	0.990	0.399	0.357	1167.08	2109.01	18.77	1867.41	1253.88	173.57	0.548
87	11.78	2.33	31.20	98.37	-13.63	0.990	0.407	0.35	1167.57	2099.85	13.55	1853.88	1258.39	168.70	0.552
88	11.88	2.31	31.50	100.30	-13.90	0.990	0.403	0.342	1168.14	2127.59	13.03	1872.11	1256.96	179.68	0.546
89	12.20	2.31	32.40	102.06	-13.90	0.990	0.398	0.339	1166.78	2150.25	14.04	1886.87	1253.00	191.19	0.539
90	12.02	2.31	31.90	103.60	-13.90	0.990	0.401	0.329	1167.53	2153.29	11.45	1880.98	1255.20	196.09	0.539
91	12.13	2.31	32.20	104.70	-13.90	0.990	0.399	0.326	1167.08	2161.92	11.38	1883.91	1253.88	202.59	0.537
92	12.06	2.33	32.00	105.05	-13.63	0.990	0.402	0.323	1166.59	2155.00	10.48	1875.85	1254.76	201.46	0.539
93	11.95	2.33	31.70	106.57	-13.63	0.990	0.404	0.314	1167.05	2157.14	9.27	1870.07	1256.08	207.30	0.539
94	11.71	2.33	31.00	108.39	-13.63	0.990	0.408	0.302	1167.87	2158.12	7.77	1861.79	1259.26	212.71	0.539
95	11.88	2.33	31.50	109.88	-13.63	0.990	0.405	0.298	1167.12	2170.80	7.89	1866.62	1257.08	222.11	0.536
96	12.20	2.31	32.40	111.27	-13.90	0.990	0.398	0.297	1166.78	2190.80	8.68	1878.22	1253.00	235.04	0.53
97	12.13	2.31	32.20	114.56	-13.90	0.990	0.399	0.281	1167.08	2215.02	7.54	1886.27	1253.88	249.49	0.525
98	12.24	2.33	32.50	115.07	-13.63	0.990	0.399	0.28	1165.84	2215.94	7.56	1885.32	1252.56	251.47	0.524
99	11.08	2.29	29.20	115.56	-14.09	0.990	0.415	0.259	1171.97	2225.05	5.26	1894.05	1267.13	241.10	0.525
100	11.95	2.33	31.70	116.51	-13.63	0.990	0.404	0.269	1167.05	2229.97	6.59	1892.96	1256.08	254.56	0.522
101	11.92	2.33	31.60	118.41	-13.63	0.990	0.404	0.26	1167.20	2252.14	6.20	1904.98	1256.52	264.04	0.517
102	11.70	2.29	31.00	118.87	-14.09	0.990	0.405	0.255	1169.58	2265.92	5.91	1914.81	1259.15	267.45	0.515
103	11.95	2.33	31.70	119.83	-13.63	0.990	0.404	0.255	1167.05	2269.42	6.06	1913.73	1256.08	272.72	0.513
104	11.95	2.29	31.70	120.30	-14.09	0.990	0.401	0.253	1168.52	2284.63	6.16	1923.25	1256.08	279.98	0.51
105	12.13	2.31	32.20	121.13	-13.90	0.990	0.399	0.252	1167.08	2291.82	6.30	1925.20	1253.88	286.12	0.508
106	11.99	2.29	31.80	123.59	-14.09	0.990	0.4	0.239	1168.37	2321.60	5.78	1938.99	1255.64	301.12	0.502
107	12.20	2.33	32.40	125.09	-13.63	0.990	0.4	0.236	1165.99	2329.12	5.81	1939.15	1253.00	308.77	0.499
108	11.19	2.29	29.50	125.51	-14.09	0.990	0.413	0.217	1171.59	2317.81	4.48	1927.94	1265.77	300.17	0.504
109	11.95	2.33	31.70	126.46	-13.63	0.990	0.404	0.226	1167.05	2335.04	5.28	1938.00	1256.08	313.28	0.499
110	11.95	2.29	31.70	126.88	-14.09	0.990	0.401	0.224	1168.52	2346.87	5.36	1944.93	1256.08	319.74	0.497
111	11.92	2.29	31.60	128.82	-14.09	0.990	0.401	0.215	1168.67	2356.95	5.13	1944.06	1256.52	330.16	0.495

5.5 CONCLUSIONS

5.5.1 The simulation model

This study has developed a methodology which could be adopted in designing an absorption refrigeration plant, given a refrigeration requirement. The graphical approach discussed in chapter 3 enables one to design the complete absorption refrigeration machine per kg mass flow rate of refrigerant. With the usage of the proposed optimisation curves, Figure 3.5, the cycle achieves its maximum performance.

Existing methods analysing the cycle, use data collected from existing plants. Simulation models do exist to design an absorption refrigeration machine, but no model known to the author predicts optimum design conditions. The development of a computerised model was the obvious step towards the design of an optimised absorption refrigeration machine.

The computerised simulation model requires information consisting of the following information:

1. the cooling medium temperature
2. the evaporator temperature
3. the purity of the refrigerant
4. the optimised generator temperature
5. the refrigeration capacity which will eventually size the plant

Then each component is designed using the pertinent equations evaluating the concentrations of the solutions and the mass and heat fluxes. These equations are developed from the curves of the published aqua-ammonia enthalpy-concentration (h-x) chart, fitted with Fourier equations. The re-constructed h-x chart, Figure C1 of APPENDIX C, depicts accurate property values for the mixture at all concentrations.

Consequently, the simulation model predicts accurate properties of the working fluid and culminates in designing an absorption refrigeration plant.

5.5.2 Comparison between simulated and experimental results

The experimental plant was constructed using the mass and heat fluxes predicted by the simulation model. The close agreement between the experimental and predicted results verifies the accuracy of the simulation model when the plant was operating within the design parameters.

A relatively large deviation exists between the experimental and predicted heat flow in the generator, absorber and heat exchanger for the -12°C evaporator and 31°C condensate temperatures. It is the result of the absorption refrigeration plant being operated at an environmental temperature higher than the design characteristics of the condenser and the absorber, thus increasing the high pressure and the concentration of the strong solution. Consequently, the solution circulation rates increase, leading to an effectively under-designed heat exchanger delivering sub-cooled strong solution and hot weak solution to the generator and absorber respectively. Due to this, the overall effect is an increase in the heat input to the generator and heat rejected from the absorber, which results in a large COP drop.

5.5.3 Design considerations of an absorption refrigeration plant

The generator was powered from combustion gases, at temperatures much higher than those encountered with low grade heat. It was thus feasible to design a small size generator and vary with ease the heat supply and the temperature of its solutions at frequent intervals.

Heat transfer calculations in the design of the condenser and absorber, show that the water-side heat transfer coefficient is in the region of $5000\text{W/m}^2\text{K}$ to $6000\text{W/m}^2\text{K}$. In air-cooled heat exchangers, the heat transfer coefficient encountered in forced convection is relative small (about $500\text{W/m}^2\text{K}$ for an air velocity of 5m/s). In refrigeration plants larger than 10kW capacity this may render the absorption refrigeration unit far too big to fit in restricted space applications, as for example, car air-conditioning.

In order to achieve high heat transfer rates in the evaporator, the vapour must flow at high speed to ensure annular flow of the vapour/liquid refrigerant in the tube. This may lead to a notable pressure drop (due to friction and change in momentum) if compared to the evaporator pressure. This pressure drop must be taken into account when designing the absorber.

There are different types of pumps that can be used in an absorption refrigeration machine. The choice depends on the conditions under which the pumps will operate. For instance, centrifugal pumps require large volume flow rates, although for small applications a by-pass may be applied. Thermal pumps, such as the one invented by L. Szucs [S14] are extremely intermittent. Diaphragm pumps activated by electromagnetic action, may send a strong

pressure pulsation through the system. Measuring diaphragm pumps activated by a secondary fluid may not operate satisfactorily in deep-freezing temperature applications where the absorber pressure must be kept well in the sub-atmospheric region. A piston pump, although slightly intermittent, can operate under any condition. Its intermittence can be reduced by a double action piston. Gear pumps and all pumps using mechanical seals (except for the centrifugal pump) may have tremendous problems due to impurities in the solutions. For small refrigeration applications, for example the size of the test-plant, a single acting piston pump can be used with adequate filtration on the suction side.

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APPENDICES

SUBROUTINES

The routines used in the procedure to analyse the properties of the saturated fluid and its vapour, are polynomials modelled from the experimental data of Scatchard et al. [S1] and from the lines of the enthalpy-concentration diagram [K10]. The experimental data has provided the polynomials for the pressure-temperature-concentration (P-T-x) relation and the polynomials for the temperature-liquid-vapour concentrations (T-x-y) relation. The enthalpy-concentration diagram has provided the polynomials for the enthalpies of the liquid and vapour concentrations.

These routines are classified in three main categories:

1. those using the P-T-x relation for the saturated solution to evaluate the pressure, or the temperature, or the concentration of the solution.
 - Routine A.1: to evaluate the pressure, if the temperature and concentration are known
 - Routine A.2: to evaluate the temperature, if the pressure and concentration are known
 - Routine A.3: to evaluate the concentration, if the temperature and pressure are known
2. those evaluating the specific enthalpies of the liquid and vapour mixtures.
 - Routine B.1: to evaluate the enthalpy of the liquid for the whole range of concentrations and pressures
 - Routine B.2: to evaluate the enthalpy of the vapour for the whole range of pressures and for vapour concentrations from $0 < y < 0.9$ and from $0.9 < y < 1.0$
3. those using the temperature-concentration-activity (T-x- α) relation to evaluate the concentration of the vapour mixture
 - Routine C.1

APPENDIX A

A.1 THE P-T-x RELATION

The (P-T-x) relation is the key to solving the equilibrium properties of any binary saturated mixture. Attempts have been made in the past to demonstrate graphically this relation for the water-ammonia solution [B13, R1], by plotting the natural logarithm of the pressure over the solution, against the inverse of its temperature. These graphs without exception, were approximated as straight lines for all concentrations of the mixture, including the pure substances. This is far from an accurate representation of the P-T-x relation, as it is well known that the saturation pressures of both water and ammonia plotted in this way are not straight lines. According to the data published by Scatchard et al. [S1] the graphical representation of the P-T-x relation is shown in Figure A.1

This data is used to form an empirical equation eq. (a1) relating the pressures, temperatures and concentrations (P-T-x relation), for solutions in equilibrium with their vapours.

$$\text{eq. (a1)} \quad \ln(P) = Fa_{(x)} + Fb_{(x)}(T) + Fc_{(x)}\left(\frac{1}{T}\right) + Fd_{(x)}(T)^2 + Fe_{(x)}\ln(T)$$

where $Fa, \dots, Fe$ are polynomial functions of the concentration x , of the form

$$F = \sum_{i=0}^9 \alpha_i x^i$$

The coefficients α_i of the polynomials $Fa \dots Fe$ are given in the Table T.1

Table T 1					
	Fa	Fb	Fc	Fd	Fe
α_0	48.100160	-0.005189	-6885.475932	0.000006	-4.814120
α_1	53179.763749	38.110430	-861301.911072	-0.021900	-10474.506938
α_2	-1655338.768000	-1097.906211	29722087.270181	0.605770	321003.127449
α_3	15296692.392000	10034.452508	-278492513.247748	-5.505568	-2959804.714569
α_4	-69774070.123000	-45752.723670	1272321746.027310	25.109001	13498865.929997
α_5	181603740.212000	119235.065031	-3308565601.737380	-65.488969	-35141742.982097
α_6	-282176852.284000	-185421.712344	5135458882.362640	101.854640	54613136.660030
α_7	258712375.576756	169993.425689	-4705304433.275860	-93.308260	-50074231.069536
α_8	-129050174.762172	-84718.226634	2346590047.899490	46.426952	24975893.612227
α_9	26990456.413470	17689.522089	-490866310.033833	-9.671665	-5222648.460826

The P-T-x relationship

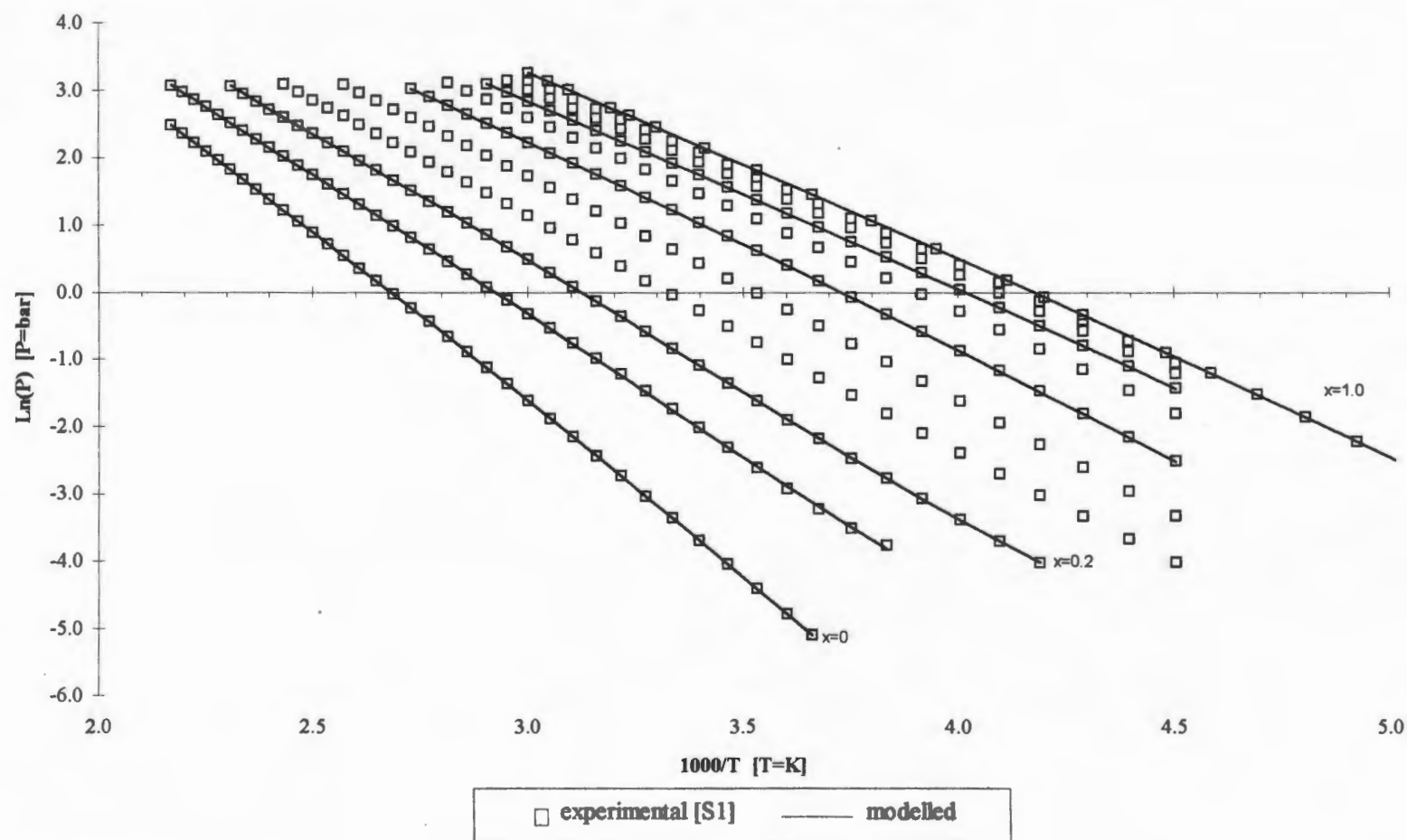


Figure A. 1

A.2 ROUTINE P-T-x (x-P ARE KNOWN)

Using Simpson's rule, the temperature of the saturated solution is iteratively calculated from the P-T-x relation within an accuracy of 0.01°C.

$$T_1 = T_0 - \frac{g_{(P)}}{g'_{(P)}}$$

where
$$g_{(P)} = \ln(P) - \left[Fa_{(x)} + Fb_{(x)}(T) + Fc_{(x)}\left(\frac{1}{T}\right) + Fd_{(x)}(T)^2 + Fe_{(x)}\ln(T) \right]$$

and
$$g'_{(P)} = - \left[Fb_{(x)} - Fc_{(x)}\left(\frac{1}{T^2}\right) + 2Fd_{(x)}(T) + \frac{Fe_{(x)}}{T} \right]$$

A.3 ROUTINE P-T-x (T-P ARE KNOWN)

Using Simpson's rule, the concentration of the saturated solution is iteratively calculated from the P-T-x relation within an accuracy of 10^{-5} .

$$x_1 = x_0 - \frac{g_{(P)}}{g'_{(P)}}$$

where $g_{(P)}$ is defined in paragraph A.2

and
$$g'_{(P)} = - \left[Fa'_{(x)} + Fb'_{(x)}(T) + Fc'_{(x)}\left(\frac{1}{T}\right) + Fd'_{(x)}(T)^2 + Fe'_{(x)}\ln(T) \right]$$

where the Fa' Fe' functions are the first derivatives of the Fa Fe functions described in paragraph A.1:

$$F' = \sum_{i=1}^9 ia_i x^{i-1}$$

APPENDIX B

B.1 ENTHALPY OF THE SATURATED LIQUID

The enthalpy of the saturated liquid is evaluated for all concentrations (including the pure substances) and for pressures within the range of 0.05bars and 30.0bars. The Fourier equation eq. (b1) is developed according to Morrison [M10], by taking 20 steps between $x=0.0$ and $x=1.0$. The independent parameters are the concentration "x" and the pressure "P" of which the pressure is a temperature dependant function itself.

$$\text{eq. (b1)} \quad h_{(x,P)} = \frac{A_{0(P)} + 2 \sum_{n=1}^{k-1} \left[A_{n(P)} \cdot \cos\left(2n\pi \frac{x_k}{2k}\right) \right]}{2k}$$

where the coefficients $A_{0(P)}$, ..., $A_{n(P)}$ are functions of pressure of the form:

$$\text{eq. (b2)} \quad A_{n(P)} = a_n + b_n \cdot P + \frac{c_n}{P} + \frac{d_n}{P^2} + e_n \cdot \ln(P) \quad \text{for } n=0 \text{ to } k-1$$

where

k is the number of steps within the concentration range. In this case $k=20$ and $x_k=20x$

The coefficients a_n , ..., e_n of the pressure dependent functions $A_{n(P)}$ are given in Table T2.

Table T 2					
n	a _n	b _n	c _n	d _n	e _n
0	-5737.657198266	243.68807855514	545.67554430165	-26.40236375388	4708.62951438
1	4425.6929503301	6.2454831179681	24.210056763353	-.859142433379	361.24982665941
2	3325.2732759945	1.8991927786602	55.127443677277	-2.942628898649	-5.621975716089
3	555.86814156735	2.1825595529516	-15.62620522503	1.7462441540074	-31.66839789966
4	681.58877163004	2.1486003896118	6.0686768503477	-.514335373036	-6.950331156627
5	224.09732320404	.2381795255529	4.8216079406042	-.270170823095	1.9600574712901
6	318.34580523841	-4.285582893783	29.373618550484	-1.607050201003	25.999023902088
7	132.29823123694	-1.636406802011	5.222442960175	-.061002692624	8.0988224223677
8	178.95424044565	-3.639002987497	25.302542141648	-1.38782496928	19.267665179519
9	70.087441780315	-4.224740497654	19.463356073905	-1.094672065729	23.889914668942
10	117.66629622102	-3.586765103566	24.120641316803	-1.689014517397	19.728213721716
11	59.134420696241	-1.358450531977	6.3751163719177	-.408506483347	4.3467794886473
12	101.50706503673	-.859158542319	9.4090711528524	-.746998768822	-1.764234187112
13	52.334606404168	-1.370085627092	2.3602252852585	-.136234354886	-.265875889225
14	78.439000181654	-1.457404884897	12.38420300889	-.757953970692	8.5139413710026
15	39.176204521858	-.745057829887	5.6262007335615	-.283735087397	2.8646703178369
16	71.238334946617	.0636764645767	6.598435604799	-.462225455998	.2947010265815
17	33.221458478344	-.061027959008	4.3642692911368	-.269406571991	.885707665349
18	54.895720632288	-.577246750851	13.048839552884	-.870631648953	6.9046892259742
19	29.412325354942	.0987156576093	4.5323718587918	-.231569237302	2.1901064487288

B.2 ENTHALPY OF THE SATURATED VAPOUR

The enthalpy of the saturated vapour is evaluated for the same pressure as that for its liquid.

The vapour concentrations are subdivided into two ranges:

- $0.000 < y < 0.900$
- $0.900 < y < 1.000$

In the first range of concentrations there are 10 k-steps.

$$\text{eq. (b3)} \quad h_{(x,P)} = \frac{A_{0(P)} + 2 \sum_{n=1}^{k-1} \left[A_{n(P)} \cdot \cos\left(2n\pi \frac{y_k}{2k}\right) \right]}{2k} 0.874972 \quad y_k = 10x$$

the coefficients $A_{0(P)}$, ..., $A_{n(P)}$ of eq. (b3) are functions of pressure of the form:

$$\text{eq. (b4)} \quad A_{n(P)} = a_n + b_n \cdot P + \frac{c_n}{P} + \frac{d_n}{P^2} + e_n \cdot \ln(P) \quad \text{for } n=0 \text{ to } k-1$$

The coefficients a_n , ..., e_n for the pressure dependent functions $A_{n(P)}$ of eq. (b4) are given in Table T3.

In the second range of concentrations there are 20 k-steps.

$$\text{eq. (b5)} \quad h_{(x,P)} = \frac{A_{0(P)} + 2 \sum_{n=1}^{k-1} \left[A_{n(P)} \cdot \cos\left(2n\pi \frac{x_k}{2k}\right) \right]}{2k}$$

where the coefficients $A_{0(P)}$, ..., $A_{n(P)}$ of eq. (b5) are functions of pressure of the form:

$$\text{eq. (b6)} \quad A_{n(P)} = a_n + b_n \cdot P + \frac{c_n}{P} + \frac{d_n}{P^2} + e_n \cdot \ln(P) \quad \text{for } n=0 \text{ to } k-1$$

The coefficients a_n , ..., e_n for the pressure dependent functions $A_{n(P)}$ of eq. (b6) are given in Table T4.

Table T 3					
n	a_n	b_n	c_n	d_n	e_n
0	2395.9380766992	.2207397856128	10.110900709721	-.63220219181	52.174724508466
1	268.9428587995	-.083929443448	-.075300010286	.0090837716743	.9218681568829
2	-4.364001161508	-.096846820382	-.016609769087	-.002593341161	-.355332897006
3	32.765240458178	-.049184617215	.5216004518891	-.03837575927	.7650227510391
4	-1.097837349326	-.019605555738	-.42748327101	.0289096918668	-.505882445001
5	14.947879318723	.0010590338311	.1297190736557	-.011297404609	.2780123148518
6	-1.111637088729	-.089675534551	.2315279203627	-.018441321974	.2296515130217
7	10.394639509169	.0486296415224	-.330393344016	.0177337305306	-.287904804994
8	-.903232339651	-.070059935636	.1244622249144	-.010061079085	.1251112237749
9	9.4102243227898	.0768497950187	-.46720229061	.0291216231561	-.392790896486

Table T 4					
n	a_n	b_n	c_n	d_n	e_n
0	48735.680612053	-25.65304102607	75.057175557151	-.332675912257	1297.5010797443
1	5738.7992405658	12.478102093287	128.60406595429	-11.20247304972	144.92387800206
2	.0014458221038	.0003243549827	-.002005760956	.0001460175271	-.00230285111
3	-640.3165099548	-4.319241599285	9.5811106620378	.5493491893461	56.108791900913
4	-.000518064663	.00009518161671	.0003152123809	-.0000457730552	-.000255175518
5	594.96602870419	-.740829178812	-14.10058433731	.5097967914078	-25.91004403612
6	-.006954370862	-.000467222129	.0049496284989	-.000279737372	.0053752319104
7	-263.3312181919	.0451667044672	18.800902435663	-1.013844398097	29.139270215121
8	-.001316506475	-.000173593344	.0012028331692	-.0000625877148	.0016683523621
9	235.44056644455	-.718974922421	-1.226642996996	.1026949920154	-4.818702646805
10	.004646604003	.0001405475868	-.003181400451	.0002163560806	-.002461563094
11	-90.48978809912	3.2566100117222	-16.9401001154	.9623833707872	-19.08618476714
12	-.001509573421	.0001237902327	.001587432882	-.000151403168	-.000287678432
13	88.582308380096	-3.021123395313	20.592061635386	-1.11569287403	25.063504766468
14	.0004315740892	.00006799847064	-.001168276113	.00007323215072	-.000468419125
15	-21.74651275785	2.8547542521876	-17.63143828745	.9367846631283	-22.36251358325
16	.0015400628731	.0000632472013	-.001228770085	.00008514764678	-.000758691843
17	33.519501407977	-1.901090510106	13.590539000027	-.725655342751	17.337855700429
18	-.000385497033	-.000127820331	.0002615042261	.00000189961176	.0009043764222
19	11.551473186426	.6644834297235	-5.366242654788	.3134604248938	-5.661767130318

APPENDIX C

C.1 VAPOUR MIXTURE CONCENTRATION

The concentration of the vapour mixture is related to the concentration of the saturated liquid and to the activity coefficient α by eq. (c1)

$$\text{eq. (c1)} \quad y = \frac{x \cdot \alpha}{1 + x(\alpha - 1)}$$

The activity coefficients are obtained from the data published by Scatchard et al [S1] and have been related to the concentration of the liquid x and its saturation temperature “T” by a double Fourier equation eq. (c2) and eq. (c3).

$$\text{eq. (c2)} \quad \ln(\alpha) = \frac{B_{(0)} + 2 \sum_{m=1}^{l-1} B_{(m)} \cos\left(2\pi \cdot m \frac{T_l}{2l}\right)}{2l} \quad l = 0, \dots, 43$$

where

$$\text{eq. (c3)} \quad B_{(m)} = \frac{A_{(0)} + 2 \sum_{n=1}^{k-1} A_{(n)} \cos\left(2\pi \cdot n \frac{x_k}{2k}\right)}{2k} \quad k = 0, \dots, 10$$

$$\text{and} \quad T_l = \frac{t + 60}{10} \quad (t = ^\circ\text{F}) \quad x_k = 10 \cdot x$$

The $A_{(n)}$ and $B_{(m)}$ coefficients are given in Table T5.

Table T 5

	A ₀	A ₁	A ₂	A ₃	A ₄	A ₅	A ₆	A ₇	A ₈	A ₉
B ₀	7989.960986	-986.0064482	-321.6404083	27.6660879	-47.77462586	-24.3078539	-10.9127197	-12.0718069	12.31716763	-26.05855936
B ₁	2325.994525	-651.9657462	-22.34392701	-28.22753903	10.4299405	-21.08667473	-4.051672026	4.135530305	-11.22999807	15.92882814
B ₂	429.3758593	-134.4459106	-10.38876131	-4.714046516	-6.101303371	-13.75152283	9.322730429	-4.277601434	8.375331685	-2.078042716
B ₃	339.6107959	-92.02505692	-1.500376842	-4.558333197	-1.283842937	-8.401584397	-0.91522316	-2.165859064	0.159071578	1.385612325
B ₄	124.4795325	-36.70526026	-1.637029942	-3.214338998	-3.684239207	-8.137514374	1.033898543	-2.478274187	3.273537493	1.211347134
B ₅	125.8165287	-37.59462956	-1.42396339	-2.007231029	0.102977526	-2.483135809	1.136615097	0.028949222	-0.036489755	1.878021115
B ₆	54.46899175	-17.48700903	-1.607983697	-0.170419086	-2.047086743	-1.765030762	-0.291294628	-0.32440094	0.217146601	2.47230426
B ₇	59.48043522	-19.43899019	-2.842489135	1.199664685	-0.305222507	2.469135923	-0.071549396	1.657831569	-0.643025056	1.517557503
B ₈	26.6165558	-8.838816165	-2.661297686	1.942595176	-1.228807995	1.852231512	-0.165779889	0.232418296	0.161210118	0.181443846
B ₉	36.11312928	-11.43876677	-1.955061577	1.644030251	0.277794106	1.691309861	0.409766664	-0.161537998	-0.030941486	-0.4892182
B ₁₀	19.53123027	-6.749727584	-0.490636273	1.152767107	0.386330785	0.053296346	0.462991781	-0.526808884	0.287667326	-0.911017922
B ₁₁	27.44016685	-8.249469873	0.256326473	0.502305319	0.833447484	-0.231667929	-0.507374662	-0.137027794	-0.318406408	-0.113668599
B ₁₂	14.62786793	-4.511302515	0.101035627	0.923837804	0.491022301	-0.066465313	-0.497500304	-0.295907942	-0.548105677	-0.667902718
B ₁₃	20.35415292	-5.097401114	-0.192887119	0.700984354	-0.248974979	-0.297634735	-0.645071191	-0.248630274	-0.289528328	-0.290334382
B ₁₄	11.13163647	-3.366294521	-0.480310832	0.586172839	-0.023377029	0.306570402	-0.008232924	-0.394334681	-0.09938209	-0.349819043
B ₁₅	15.62661697	-4.439883938	-0.308376458	-0.156275755	-0.102110934	0.192914488	-0.672727749	-0.093457776	0.021735693	0.75446656
B ₁₆	7.600491664	-3.224369146	-0.730407544	0.01532201	0.091310479	0.734335137	-0.144504494	0.277986339	0.258772054	0.338163294
B ₁₇	11.44113197	-3.624923724	-0.979019143	-0.118667583	-0.297411849	0.392768459	-0.296058895	-0.072260296	0.569362279	0.770189871
B ₁₈	6.196583077	-2.53896418	-0.938275544	-0.115130729	-0.237289041	0.307488846	0.16297757	-0.243974707	0.744253429	0.419393599
B ₁₉	9.78426144	-3.46057511	-0.398375729	-0.366427903	-0.053037614	0.051133947	-0.178676831	0.108327013	0.54684543	0.611905323
B ₂₀	6.091147634	-1.989766914	-0.114177923	-0.246427413	-0.238473763	-0.108244705	-0.460376882	0.025416797	0.238384618	0.422567922
B ₂₁	8.765005575	-2.336580993	-0.166740637	0.157430751	-0.232196526	0.077582747	-0.412590705	-0.156673135	0.049869787	0.217485402
B ₂₂	5.510798867	-0.925231641	0.014280023	0.326172209	-0.475844345	-0.312349099	-0.466365698	-0.556420332	0.117711386	0.067085792
B ₂₃	7.983490741	-1.732993359	0.12639885	0.126156072	0.051890973	-0.020673858	-0.361544482	-0.335423356	-0.237610254	-0.088434744
B ₂₄	5.315065106	-0.753889706	0.353212822	0.083850915	-0.154405382	-0.320393977	-0.67371314	-0.36948217	-0.241691111	0.137127883
B ₂₅	7.00092659	-1.505576482	0.049001136	0.295210724	-0.008304127	0.162006505	-0.302039819	-0.274728099	-0.299615879	-0.12785958
B ₂₆	4.337287216	-0.640920617	-0.087983806	0.197632901	-0.396472183	-0.223231619	-0.389534494	-0.492680155	0.107931413	0.173395417
B ₂₇	6.021875361	-1.616592913	-0.044343163	-0.02031309	-0.066732108	-0.077430859	-0.2640278	-0.113488806	0.150718266	0.226749405
B ₂₈	3.830471912	-1.199310064	-0.006183266	-0.263864185	-0.270397005	-0.186049823	-0.324834066	0.005138038	0.190529641	0.43253818
B ₂₉	5.515912707	-1.756609799	-0.29390291	-0.159196914	-0.339752078	-0.016477923	-0.291467853	-0.015105957	0.216564241	0.610372678
B ₃₀	3.066981758	-1.128793805	-0.56459523	-0.053295688	-0.477542618	-0.079105303	-0.093339775	-0.003577348	0.445030918	0.435489136
B ₃₁	4.940627816	-1.478439257	-0.381963681	-0.300940833	-0.412976057	-0.242394963	-0.224872562	0.024348262	0.399954495	0.610995921
B ₃₂	2.860017369	-1.316860234	-0.102379321	-0.196963783	-0.037154331	0.07249447	0.003621996	0.045660925	0.135894403	0.299123648
B ₃₃	4.842575783	-1.453113879	0.004248514	-0.102150154	-0.216556707	-0.091610604	-0.197812091	0.015299213	0.187590546	0.402507109
B ₃₄	2.730849946	-0.777636625	-0.155229164	0.328060687	0.064281059	0.205418993	-0.025943113	-0.199395332	-0.016096879	-0.166462029
B ₃₅	4.574329069	-0.841584893	-0.004739683	0.199335904	-0.018710965	-0.028216057	-0.25744717	-0.238255863	-0.110793818	-0.144539331
B ₃₆	2.875670419	-0.733527917	0.199981339	0.235189046	0.201326549	0.168285943	-0.037858951	-0.082762897	-0.232134665	-0.420757692
B ₃₇	4.680472786	-1.189254555	0.207253116	0.197282129	0.180094697	0.164068563	-0.196690813	-0.145480233	-0.285062161	-0.215995932
B ₃₈	2.831847588	-0.600803913	0.051854171	0.259054326	0.014209403	0.124717596	-0.138829383	-0.184812565	-0.188795527	-0.298634849
B ₃₉	3.961250916	-1.076353672	-0.048116066	0.168359103	0.048748327	0.191377279	-0.042110008	-0.073472097	-0.123604249	-0.222121458
B ₄₀	2.560753881	-0.892259289	-0.047968258	-0.021891861	0.008743671	0.132858045	-0.019721183	-0.115674247	-0.020972115	-0.028641461
B ₄₁	3.918339138	-1.714588742	-0.228780903	-0.068689885	0.201847774	0.352111383	0.124138425	0.096324332	0.072291326	0.080047909
B ₄₂	2.219250551	-1.111871573	-0.40705477	-0.079675099	-0.076550111	0.146907613	0.001261274	0.087186749	0.285827573	0.28305503

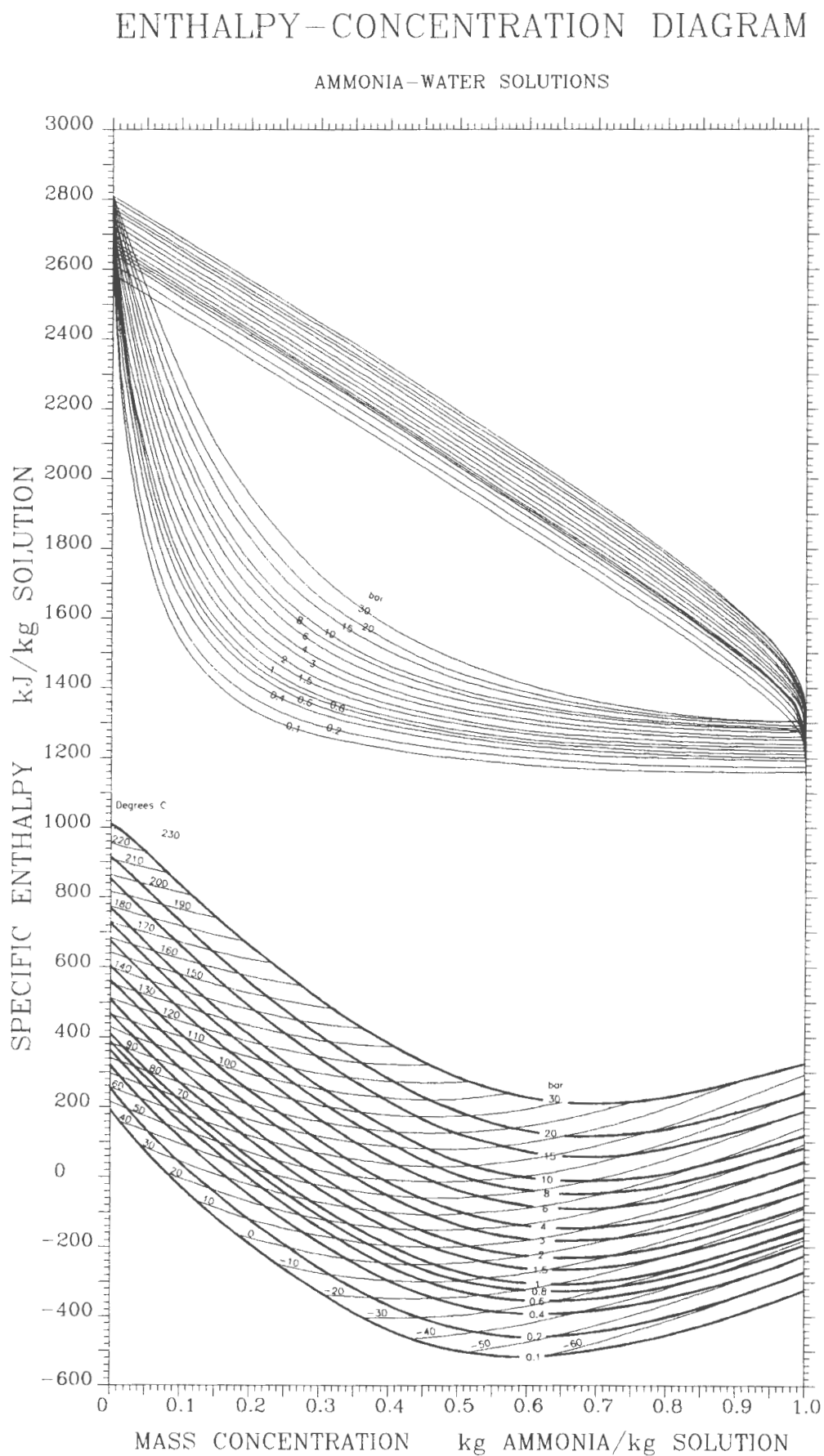


Figure C 1 The reconstructed enthalpy-concentration diagram using the equations in APPENDICES A, B and C

APPENDIX D

D1. PHYSICAL PROPERTIES OF THE WORKING FLUIDS

$$[T=K, t=^{\circ}C]$$

D1.1. Ammonia liquid:

$$C_p = 19.577576 - 0.25297(T) + 1.536215 \times 10^{-3}(T)^2 - 4.0974 \times 10^{-6}(T)^3 + 4.1 \times 10^{-9}(T)^4$$

$$Pr = -3931.39 - 2.37933(T) + 1.256 \times 10^{-3}(T)^2 + 78879.76/(T) + 748.7102 \ln(T)$$

$$\rho = 639.4976 - 1.50925(t) - 3.6 \times 10^{-3}(t)^2 + 5.44 \times 10^{-5}(t)^3$$

$$k = 5.38 \times 10^{-4} - 7.79 \times 10^{-7}(t) - 1.38 \times 10^{-8}(t)^2 + 6.96 \times 10^{-11}(t)^3$$

$$\mu = -0.720765 - 4.789657 \times 10^{-4}(T) + 2.74484 \times 10^{-7}(T)^2 + 13.61845/(T) + 0.13929 \ln(T)$$

$$\sigma = 0.016$$

D1.2. Ammonia vapour:

$$v = 4.29 \times 10^{-8}(t) - 8.7 \times 10^{-11}(t)^2 + 2.8 \times 10^{-13}(t)^3 + 9.138 \times 10^{-6}$$

$$\rho = -3.048 \times 10^{-3}(t) + 6.872 \times 10^{-6}(t)^2 + 0.7823$$

$$\mu = v \cdot \rho$$

D1.3. Water:

$$Pr = 22.14339 + 0.035961(t) - 3.1 \times 10^{-5}(t)^2 - 11.4457/(t) - 5.11174 \ln(t)$$

$$\rho = 999.5444 + 0.018128(t) - 5.5 \times 10^{-3}(t)^2 + 1.15 \times 10^{-5}(t)^3$$

$$k = 5.51 \times 10^{-4} + 2.6 \times 10^{-6}(t) - 1.4 \times 10^{-8}(t)^2 + 1.28 \times 10^{-11}(t)^3$$

$$C_p = 5.5088 - 8.24476 \times 10^{-3}(T) + 1.27576 \times 10^{-5}(T)^2$$

$$\mu = -6.96965 - 3.44829 \times 10^{-3}(T) + 1.5453 \times 10^{-6}(T)^2 + 160.49376/(T) + 1.285278 \ln(T)$$

$$\sigma = -7.46351 - 3.739 \times 10^{-3}(T) + 1.405 \times 10^{-6}(T)^2 + 172.333/(T) + 1.395 \ln(T)$$

D1.4. Solution:

$$(\text{Property})_s = x \cdot (\text{Property})_A + (1-x)(\text{Property})_w$$

D1.5. Air properties:

The air properties from [M6] are converted as:

$$\text{Property} = A_1 + A_2 (T) + A_3 (T)^2 + A_4 / (T) + A_5 \ln(T)$$

	A₁	A₂	A₃	A₄	A₅
k W/mK	0.053835435	7.41947E-05	-7.3169E-09	-1.71770853	-0.00752037
ρ kg/m³	0.087688058	1.76078E-05	-3.3279E-09	350.4610848	-0.01433442
C_p J/kgK	2420.846464	0.790146503	-0.00016515	-21047.351	-274.807111
μ kg/ms	-3.6872E-05	2.12319E-08	-1.1998E-12	0.000353703	8.39096E-06

The Prandtl number is constant for the range of temperatures considered: Pr=0.69

APPENDIX E

E1. CONDENSER CALCULATIONS

The condenser is designed with the following set characteristics:

- cooling water velocity $V=1.2\text{m/s}$
- inner tube diameter $d_i=3\text{mm}$
- outer tube diameter $d_o=6\text{mm}$
- number of tubes $N=19$
- temperature of the condensate $T_{\text{sat}}=25^\circ\text{C}$
- temperature of the inlet water $T_{wi}=18^\circ\text{C}$
- condenser's load $Q_c=1240\text{W}$

The values of the specific heat C_{pw} , the density ρ_w , the conductivity k_w , the Prandtl number Pr_w , and the viscosity μ_w , are evaluated from the polynomials in APPENDIX D, for an average temperature between the inlet and outlet cooling water temperatures:

$$T_w = \frac{T_{wi} + T_{wo}}{2} = 18^\circ\text{C} \quad (\text{first approximation}) \quad \{\mathbf{e1}\}$$

For this average temperature the mass rate of the cooling water is:

$$m_w = N \cdot \rho_w \cdot V \cdot \pi \cdot r_i^2 = 0.161 \text{ kg/s}$$

Then the outlet temperature of the cooling water is:

$$T_{wo} = \frac{Q_c}{m_w C_{pw}} + T_{wi} = 19.83^\circ\text{C}$$

The iterative calculations from equation $\{\mathbf{e1}\}$ converge at:

$$T_{wo}=19.84^\circ\text{C}$$

$$m_w=0.161 \text{ kg/s}$$

$$Pr_w=7.18$$

$$k_w=0.5953 \text{ W/m K}$$

$$\mu_w = 1.0515 \times 10^{-3} \text{ kg/m s}$$

The Reynolds number is $Re = 3416.8$

For $Re > 3000$, the friction factor is given by:

$$\xi = (0.790 \ln(Re) - 1.64)^{-2} = 5.45 \times 10^{-3}$$

The inner heat transfer coefficient is given by Gnielinski's formula [M6], an updated version of [W1]:

$$a_i = \left(\frac{k_w}{d_i} \right) \frac{\left(\frac{\xi}{8} \right) \cdot Pr_w (Re - 1000)}{1 + 12.7 \left[(Pr_w)^{\frac{2}{3}} - 1 \right] \left(\frac{\xi}{8} \right)^{0.5}} = 5284.7 \text{ W/m}^2 \text{ K}$$

For a condensing refrigerant on the outside surfaces of tubes, the heat transfer coefficient is given by:

$$a_o = 0.728 \left[\frac{(\rho_l - \rho_v) \cdot g \cdot h_{fg}^* \cdot k_l^3}{N \cdot v_l \cdot D \cdot (T_{sat} - T_{wall})} \right]^{\frac{1}{4}} \quad \{e2\}$$

where

$N = 19$ is the number of passes of tubes within the shell and

$$h_{fg}^* = h_{fg} + \left(0.683 - \frac{0.228}{Pr_l} \right) \cdot Cp_l \cdot (T_{sat} - T_{wall}) \quad \{e3\}$$

is the latent heat correction, according to Sadasivan and Lienhard [M6]

The properties of the saturated condensate at temperature $T_{sat} = 25^\circ\text{C}$, are given by the polynomials in APPENDIX D:

$$\rho_L^a = 600.37 \text{ kg/m}^3$$

$$\rho_V^a = 0.710 \text{ kg/m}^3$$

$$\mu_L^a = 1.336 \times 10^{-4} \text{ kg/m s}$$

$$v_L^a = 2.23 \times 10^{-7} \text{ m}^2/\text{s}$$

$$Cp_L^a = 4516.6 \text{ J/kg K}$$

$$Pr_L^a = 1.276$$

$$k_L^a = 0.511 \text{ W/m K}$$

The enthalpies of the saturated vapour and liquid at $T_{\text{sat}}=25^\circ\text{C}$ are given by the correlations in APPENDIX B:

$$\begin{aligned} h_v &= 1706.6 \times 10^3 \text{ J/kg} \\ h_L &= 542.02 \times 10^3 \text{ J/kg} \\ \text{and } h_{fg} &= h_v - h_L = 1164.5 \times 10^3 \text{ J/kg} \end{aligned}$$

Assuming a wall temperature $T_{\text{wall}} = \frac{T_{wi} + T_{wo}}{2} + 0.5$ as a first approximation and substituting the above values into {e3}:

$$h_{fg}^* = 1177.25 \times 10^3 \text{ J/kg}$$

and from {e2}:

$$a_o = 7781.6 \text{ W/m}^2 \text{ K}$$

The resistances due to fouling are:

$$\text{by the refrigerant } R_{\text{ref}} = 0.00025$$

$$\text{by the water } R_{\text{FF}} = 0.0001$$

$$\text{the wall conductance } k_w = 40 \text{ W/m}$$

The overall heat transfer coefficient is given by the relation:

$$\frac{1}{U} = \frac{1}{\left(\frac{r_i}{r_o}\right) a_i} + \frac{r_o \ln\left(\frac{r_o}{r_i}\right)}{k_t} + \frac{1}{a_o} + R_{\text{ref}} + R_{\text{FF}} = 1100.18$$

The heat flux is:

$$\frac{Q}{A} = U \cdot (T_{\text{sat}} - T_w) = 6688.34 \text{ W/m}^2$$

The tube length is then:

$$L = \frac{Q_c}{\frac{Q}{A} \cdot (N \cdot \pi \cdot d_o)} = 0.518 \text{ m}$$

The wall temperature T_{wall} is given by:

$$T_{\text{wall}} = T_{\text{sat}} - \frac{Q}{A \cdot a_o} = 24.14^\circ\text{C}$$

This wall temperature is iteratively calculated from equation {e2} until convergence.

The end results are:

Wall temperature	$T_{\text{wall}} = 24.5^\circ\text{C}$
Water-side heat transfer coefficient	$a_i = 5284.7 \text{ W/m}^2\text{K}$
Ammonia-side heat transfer coefficient	$a_o = 14154.6 \text{ W/m}^2\text{K}$
Overall heat transfer coefficient	$U = 1175 \text{ W/m}^2\text{K}$
Tube length	$L = 0.49 \text{ m}$

APPENDIX F

F1. SOLUTION HEAT EXCHANGER

The solution heat exchanger is designed with the following set characteristics:

- heat exchanger's load $Q_{EX} = 1016 \text{ W}$
- shell-tube outside diameter $D_o = 10 \text{ mm}$
- shell tube wall thickness $t_{w2} = 1 \text{ mm}$
- inside-tube outside diameter $d_o = 6 \text{ mm}$
- inside-tube wall thickness $t_{w1} = 1 \text{ mm}$
- tube material steel
- strong solution concentration $x^{ST} = 0.433$
- weak solution concentration $x^{WE} = 0.305$
- inlet temperature of strong solution $T_{13} = 25^\circ\text{C}$
- inlet temperature of weak solution $T_8 = 100^\circ\text{C}$
- outlet temperature of strong solution $T_{10} = 73.9^\circ\text{C}$
- mass rate of weak solution $m_8 = 3.67 \text{ g/s}$
- mass rate of strong solution $m_7 = 4.52 \text{ g/s}$

For an average temperature $T = \frac{T_8 + T_9}{2}$ the specific heat Cp_{WE} is calculated by the equations given in

APPENDIX D.

Thus for $T = \frac{T_8 + T_9}{2} = 100^\circ\text{C}$ (first approximation) {f1}

A heat balance across the heat exchanger gives:

$$Q_{EX} = m_{ST} \cdot Cp_{ST} \cdot (T_{10} - T_{13}) = m_{WE} \cdot Cp_{WE} \cdot (T_8 - T_9)$$

and

$$T_9 = T_8 - \frac{Q_{EX}}{m_8 \cdot Cp_{WE}}$$

Iterative calculations from equation {f1} gives:

$$T_9 = 37.6^\circ\text{C}$$

The Logarithmic Mean Temperature Difference is given by:

$$\Delta T_{Ln} = \frac{(T_8 - T_{10}) - (T_9 - T_{13})}{Ln \left(\frac{(T_8 - T_{10})}{(T_9 - T_{13})} \right)} = 18.53 \text{ } ^\circ\text{C}$$

For the average temperatures $T_{WE} = \frac{T_8 + T_9}{2} = 68.8 \text{ } ^\circ\text{C}$

$$T_{ST} = \frac{T_{13} + T_{10}}{2} = 49.45 \text{ } ^\circ\text{C}$$

and for a hydraulic diameter $D_H = 2 \text{ mm}$

The fluid velocities are calculated by:

$$V_{WE} = \frac{m_{WE}}{\rho_{WE}} \cdot \frac{1}{\pi \cdot r_i^2} = 0.346 \text{ m/s}$$

$$V_{ST} = \frac{m_{ST}}{\rho_{ST}} \cdot \frac{1}{\pi \cdot \left(r_o^2 - \left(\frac{d_i}{2} \right)^2 \right)} = 0.256 \text{ m/s}$$

The Reynolds numbers for the flows are:

$$Re_{ST} = 1146$$

$$Re_{WE} = 3682.6$$

The individual stream heat transfer coefficients are calculated by equations corresponding to inner and annular flows. Thus for turbulent flow, Gnielinski's formula gives:

$$\xi_{WE} = \left(0.790 Ln(Re_{WE}) - 1.64 \right)^{-2} = 0.0426$$

$$a_i = \left(\frac{k_{WE}}{D_i} \right) \frac{\left(\frac{\xi_{WE}}{8} \right) \cdot Pr_{WE} (Re_{WE} - 1000)}{1 + 12.7 \left[\left(Pr_{WE} \right)^{\frac{2}{3}} - 1 \right] \left(\frac{\xi_{WE}}{8} \right)^{0.5}} = 2850.9 \text{ W/m}^2 \text{ K}$$

For a first estimation of the tube length $L=7\text{m}$ the heat transfer coefficient for the strong solution is:

$$a_o = 0.86 \left(\frac{D_i}{D_o} \right)^{0.16} \left(\frac{k_{ST}}{D_H} \right) \left[3.66 + \frac{0.065 \left(\frac{D_H}{L} \right) \cdot \text{Re}_{ST} \cdot \text{Pr}_{ST}}{1 + 0.04 \left[\left(\frac{D_H}{L} \right) \cdot \text{Re}_{ST} \cdot \text{Pr}_{ST} \right]^{\frac{2}{3}}} \right] = 1062 \text{ W/m}^2 \text{ K} \quad \{f2\}$$

The overall heat transfer coefficient is given by the relation:

$$\frac{1}{U} = \frac{1}{\left(\frac{r_i}{r_o} \right) \cdot a_i} + \frac{r_o L \ln \left(\frac{r_o}{r_i} \right)}{k_t} + \frac{1}{a_o} = 1.395 \times 10^{-3}$$

The area for the heat transfer is:

$$A = \frac{Q_{EX}}{\Delta T_{Ln} \cdot U} = 0.0764 \text{ m}^2$$

and the length of the tube is:

$$L = \frac{A}{\pi \cdot D_o} = 4.056 \text{ m}$$

Iterative calculations from {f2} give the following results:

heat transfer coefficient for the strong solution	$a_o = 1072.7$
overall heat transfer coefficient	$U = 720.1 \text{ W/m}^2 \text{ K}$
total area for heat transfer	$A = 0.0761 \text{ m}^2$
length of double tube	$L = 4.037 \text{ m}$

APPENDIX G

G1. ABSORBER

The absorber is designed for the following set characteristics:

- cooling water velocity $V_w = 0.4 \text{ m/s}$
- mass rate of ammonia vapour $m_{\text{ref}} = 0.847 \text{ g/s}$
- concentration of ammonia vapour $y = 0.990$
- concentration of weak solution $x^{\text{WE}} = 0.305$
- concentration of strong solution $x^{\text{ST}} = 0.433$
- mass rate of weak solution $m_s = 3.67 \text{ g/s}$
- saturation temperature of the strong solution $T_s = 25^\circ\text{C}$
- inlet cooling-water temperature $T_{wi} = 18^\circ\text{C}$
- inner diameter of core-tubes $d_i = 17 \text{ mm}$
- outer diameter of core-tubes $d_o = 20 \text{ mm}$
- inner diameter of shell tube $D_i = 69 \text{ mm}$
- number of core-tubes $N = 7$

G1.1 Water-side partial heat transfer coefficient a_c

For an average temperature between inlet and outlet

$$T = \frac{T_{wi} + T_{wo}}{2} = 18^\circ\text{C} \quad (\text{first approximation}) \quad \{\mathbf{g1}\}$$

The water properties are calculated by the equations in APPENDIX D.

The mass flow rate of cooling water is:

$$m_w = \frac{V_w \cdot \rho_w \cdot \pi (D_i^2 - N \cdot d_o^2)}{4} = 0.612 \text{ kg/s}$$

The outlet water temperature is given by:

$$T_{wo} = T_{wi} + \frac{Q_{abs}}{C_{pw} \cdot m_w} = 18.7^\circ\text{C}$$

Iterative calculations from {g2} give the following results:

water density	$\rho_w = 998.1 \text{ kg/m}^3$
dynamic viscosity	$\mu_w = 1.067 \times 10^{-3} \text{ kg/m s}$
mass flow rate of water	$m_w = 0.615 \text{ kg/s}$
specific heat	$C_{pw} = 4189 \text{ J/kg K}$

The hydraulic diameter and Reynolds number are:

$$D_H = \frac{D_i^2 - N \cdot d_o^2}{D_i + N \cdot d_o} = 9.383 \times 10^{-3} \text{ m}$$

$$Re_H = \frac{\rho_w \cdot V_w \cdot D_H}{\mu_w} = 3512.2$$

The Logarithmic Mean Temperature Difference is given by:

$$\Delta T_{Ln} = \frac{(T_{Si} - T_{Ci}) - (T_{So} - T_{Co})}{\ln \left(\frac{(T_{Si} - T_{Ci})}{(T_{So} - T_{Co})} \right)} = 11.74^\circ\text{C}$$

The friction factor and water-side heat transfer coefficient are given by:

$$\xi_w = \left(0.790 \ln(Re_H) - 1.64 \right)^{-2} = 5.404 \times 10^{-3}$$

$$a_c = 0.86 \left(\frac{k_w}{D_H} \right) \left(\frac{D_2}{D_o} \right)^{0.16} \cdot \frac{\left(\frac{\xi_w}{8} \right) \cdot Pr_w (Re_H - 1000)}{1 + 12.7 \left[\left(Pr_w \right)^{\frac{2}{3}} - 1 \right] \left(\frac{\xi_w}{8} \right)^{0.5}} = 1837.36 \text{ W/m}^2 \text{ K}$$

G1.2 Solution-side partial heat transfer coefficient a_i

The partial heat transfer coefficient is calculated according to Horn [H9], for a two-phase flow of the solution in the inner tubes.

$$a_i = \varepsilon \cdot a_{LF} + (1 - \varepsilon) \cdot a_{LP} \quad \{g2\}$$

According to Hubbard [S5], the thickness of the falling film on the inside surface of a tube is approximately given by:

$$\delta = 0.05 d_i = 8.5 \text{ mm} \quad (\text{first approximation})$$

The velocity of the vapour can be found by:

$$u_G = \frac{m_{ref}}{N \frac{\rho \cdot \pi D_i^2}{4}} = 0.676 \text{ m/s}$$

Moissis [M7] has expressed the absolute liquid film velocity of descending liquid of thickness δ as:

$$V_f = \frac{1}{4\delta \cdot (D - \delta)} \left[\left(1.2 \frac{Q_G + Q_L}{A} + C \sqrt{gD} \right) (D - 2\delta)^2 - D^2 \frac{Q_G + Q_L}{A} \right] = 0.5122 \text{ m/s} \quad \{g3\}$$

If the liquid film is assumed to fill the gap between the wall and the rising slug, the characteristic length for the Reynolds number Re_{LF} is 2δ .

Thus the Reynolds number Re_{LF} is:

$$Re_{LF} = \frac{4\delta \cdot \rho_f \cdot V_f \cdot \left(1 - \frac{\delta}{D} \right)}{\mu_f} = 1415.9$$

For $Re_{LF} > 400$ the film thickness is re-calculated by the following equation [B11]:

$$\delta = 0.302 (Re_{LF})^{\frac{8}{15}} \left(\frac{3 \cdot \left(\frac{\mu}{\rho} \right)^2}{g} \right)^{\frac{1}{3}} = 7.052 \times 10^{-4} \text{ m}$$

Iterative calculations form {g3} give the following results

film thickness	$\delta = 7.676 \times 10^{-3} \text{ m}$
liquid film velocity	$V_f = 0.67 \text{ m/s}$
Reynolds number	$Re_{LF} = 1660$

For $Re_{LF} > 800$, the Nusselt number is given by:

$$Nu_{LF} = 0.00621 (Re_{LF})^{\frac{14}{15}} \cdot Pr^{0.344} = 9.231$$

and the heat transfer coefficient for the liquid film is:

$$a_{LF} = Nu_{LF} \frac{k_{sol}}{D_i} = 309.5 \text{ W/m}^2 \text{ K}$$

The velocity of the liquid slug in the tubes is dependant on the mass flow rate of the weak solution. Thus

$$m_{LP} = \frac{m_G}{N} \cdot \frac{y - x^{ST}}{x^{ST} - x^{WE}} = 5.265 \times 10^{-4} \text{ kg/s}$$

$$u_{LP} = \frac{m_{LP}}{\rho_L} \frac{\pi \cdot D_i^2}{4} = 2.65 \times 10^{-3} \text{ m/s}$$

and

$$Re_{LP} = \frac{u_{LP} D_i \rho_L}{\mu_L} = 58.83$$

The void fraction “ ε ” is defined as:

$$\varepsilon = \frac{\text{section area of tube} - \text{area of the film ring}}{\text{section area of tube}} = \frac{(D_i - 2\delta)^2}{D_i^2} = 0.828$$

For a vapour concentration in ammonia $y=0.990$, its saturated solution has concentration:

$$x^* = 0.535$$

With reference to section 2.1, the driving potential for mass transfer is:

$$dx = x^* - x^{WE} = 23.04 \quad dx = \% \text{ concentration difference}$$

Assuming a volumetric mass transfer coefficient $K_L \alpha = 18$ as a first approximation, the height “ z ” of the tubes necessary to absorb all the incoming vapour is given by:

$$z = \frac{m_{ref}}{dx \cdot K_L \alpha \cdot \frac{\pi \cdot d_i^2}{4} \cdot N} = 0.683 \text{ m} \quad \text{Keizer [K5]} \quad \{g4\}$$

For $Re_{LP} < 2300$, according to Edwards, Denny and Mills [M6], the Nusselt number is

$$Nu_{LP} = 3.66 + \frac{0.065 Re_{LP} Pr \left(\frac{D_i}{z(1-\epsilon)} \right)}{1 + 0.04 \left(Re_{LP} Pr \left(\frac{D_i}{z(1-\epsilon)} \right) \right)^{\frac{2}{3}}} = 5.079 \quad \{g5\}$$

and the liquid phase heat transfer coefficient is:

$$a_{LP} = Nu_{LP} \frac{k_{sol}}{D_i} = 170.29 \text{ W/m}^2 \text{ K}$$

Substituting these values into equation {g1}, the partial heat transfer coefficient is:

$$a_i = 285.48 \text{ W/m}^2 \text{ K}$$

The overall heat transfer coefficient is given by:

$$\frac{1}{U} = \frac{1}{a_i} + \frac{D_i}{2k_t} \ln \left(\frac{D_o}{D_i} \right) + \frac{D_i}{D_o \cdot a_c}$$

and

$$U = 250 \text{ W/m}^2 \text{ K}$$

The height of the tubes for heat transfer is given by:

$$z = \frac{Q_{abs}}{U \cdot \Delta T_{Ln} \cdot N \cdot \pi \cdot D_o} = 1.42 \text{ m}$$

Iterative calculations from {g5} give the height of the tubes as $z = 1.43 \text{ m}$

Iterative calculations from {g4} converge in evaluating the volumetric overall mass transfer coefficient $K_L \alpha$ and the height of the core tubes. The results are as follows:

volumetric overall mass transfer coefficient $K_L \alpha = 8.92 \text{ kg/m}^3 \text{ s}$

height of the tubes $z = 1.38 \text{ m}$

solution partial heat transfer coefficient $a_i = 296.04 \text{ W/m}^2 \text{ K}$

overall heat transfer coefficient $U = 258.06 \text{ W/m}^2 \text{ K}$

G1.3 Heat and mass transfer computations

Despite the comments of paragraph 4.2, computations were performed on a seven-tube absorber of 17mm internal tube diameter, Table G1. In these computations, the absorbing height is calculated in terms of a varying volumetric overall mass transfer coefficient $K_L\alpha$ and compared to the height necessary to remove the heat generated by gas absorption and by cooling the weak solution to the absorber's temperature.

Table G1				
7-tube absorber				
$Q_{ABS}=1835W$ $D_i=17mm$, $D_o=20mm$, $T_{Wf}=18^\circ C$, $V_w=0.4m/s$				
Overall heat transfer coefficient	Tube height necessary for heat transfer	Heat transfer coefficient to produce $z_h=z_m$	Tube height necessary for mass transfer	Volumetric overall mass transfer coefficient
U [W/m ² K]	z_h [m]	U [W/m ² K]	z_m [m]	$K_L\alpha$ [kg/m ³ s]
247.059	1.438	57.822	6.146	2.0
258.058	1.377	155.644	3.073	4.0
258.058	1.377	231.289	1.536	8.0
258.058	1.377	462.578	0.768	16.0
258.058	1.377	925.156	0.384	32.0
258.058	1.377	1850.312	0.192	64.0
258.058	1.377	3700.624	0.096	128.0
258.058	1.377	7401.247	0.048	256.0

The first two columns represent the overall heat transfer coefficient U and tube height z_h necessary to remove the absorber's heat and therefore, are independent of mass transfer variations. On the contrary, the fourth column indicates that there is a large variation of the tube height z_m when the mass transfer coefficient changes. The third column gives an indication of what the overall heat transfer coefficient should have been, for the tube height z_m to be sufficient for heat transfer too.

It can be seen that in the region of an overall volumetric mass transfer coefficient $K_L\alpha = 8.0$ kg/m³s the height of the tubes for heat transfer is close to the height of the tubes for mass transfer. Further calculations reveal, as shown in Table G2, that a height of 1.38m would be sufficient for both heat and mass transfer.

Table G2				
7-tube absorber				
$Q_{ABS}=1835W$ $D_i=17mm$, $D_o=20mm$, $T_{Wf}=18^\circ C$, $V_w=0.4m/s$				
Overall heat transfer coefficient	Tube height necessary for heat transfer	Heat transfer coefficient to produce $z_h=z_m$	Tube height necessary for mass transfer	Volumetric overall mass transfer coefficient
U [W/m ² K]	z_h [m]	U [W/m ² K]	z_m [m]	$K_L\alpha$ [kg/m ³ s]
258.06	1.38	258.06	1.38	8.92

APPENDIX H

H1 EVAPORATOR COILS

H1.1 tube definition for annular-flow

The evaporator is designed for the following set characteristics:

- evaporator load $Q_E = 1000 \text{ W}$
- evaporator pressure $P_{\text{low}} = 2.2 \text{ bar}$
- inlet temperature $T_{15} = -15 \text{ }^\circ\text{C}$
- outlet temperature $T_5 = -12 \text{ }^\circ\text{C}$
- mass rate of ammonia $\dot{m}_{\text{ref}} = 0.847 \text{ g/s}$
- inlet water temperature $T_{\text{wi}} = 18 \text{ }^\circ\text{C}$
- length of refrigerant tube $L = 2 \text{ m}$
- quality of the refrigerant vapour in the tube $\phi = 0.6$

Assume an inner tube diameter $d_i = 13 \text{ mm}$ as a first approximation.

The mass velocities are defined as:

$$G = \frac{4\dot{m}_{\text{ref}}}{\pi \cdot d_i^2} = 6.385$$

$$G_g = \phi \cdot G = 3.831$$

$$G_L = (1 - \phi) \cdot G = 2.554$$

The refrigerant properties are calculated from APPENDIX D, for an average temperature:

$$T = \frac{T_{15} + T_5}{2} = -13.5^\circ\text{C} \quad \{h1\}$$

The single-phase Reynolds numbers are:

$$\text{Re}_g = \frac{G_g \cdot d_i}{\mu_g} = 7065.94 \quad \{h2\}$$

$$\text{Re}_L = \frac{G_L \cdot d_i}{\rho_L \cdot \nu_L} = 172.35$$

The friction factors and the single-phase pressure gradients are:

$$\xi_g = \left(0.79 \ln(\text{Re}_g) - 1.64\right)^{-2} = 3.48 \times 10^{-2}$$

$$\xi_L = \left(0.79 \ln(\text{Re}_L) - 1.64\right)^{-2} = 0.1696$$

$$\left(\frac{dP}{dL}\right)_g^{SP} = \xi_g \frac{G_g^2}{2\rho_g d_i} = 23.83 \text{ Pa/m}$$

$$\left(\frac{dP}{dL}\right)_L^{SP} = \xi_L \frac{G_L^2}{2\rho_L d_i} = 0.0646 \text{ Pa/m}$$

The superficial velocities are:

$$u_{GS} = \frac{G_g}{\rho_g} = 4.65 \text{ m/s}$$

$$u_{LS} = \frac{G_L}{\rho_L} = 3.88 \times 10^{-3} \text{ m/s}$$

For an average temperature given by equation {h1}, and pressure P_{Low} , the saturated liquid and vapour enthalpies are:

$$h_g = 1244.47 \text{ kJ/kg} \quad \text{\textbf{{h3}}}$$

$$h_f = -165.47 \text{ kJ/kg}$$

Thus the latent heat is:

$$h_{fg} = h_g - h_f = 1409.94 \text{ kJ/kg}$$

Assume a tube length $L=2\text{m}$

The pressure gradients due to momentum change are:

$$\left(\frac{dP}{dL}\right)_M = \left(\frac{G}{(1-\Phi)\rho_L}\right)^2 \rho_L^2 \left(\frac{Q_{REF}}{L \cdot m_{ref} \cdot h_{fg}}\right) = 0.162 \text{ Pa/m}$$

The Total Pressure Drop is the sum of all the pressure gradients:

$$T.P.D. = \left[\left(\frac{dP}{dL}\right)_g^{SP} + \left(\frac{dP}{dL}\right)_L^{SP} + \left(\frac{dP}{dL}\right)_M \right] \cdot L = 173.67 \text{ Pa/m}$$

The pressure in the evaporator is set as $(P_{\text{Low}} - T.P.D.)$ and the enthalpies of the liquid and vapour mixtures are re-calculated from equation {h3} until convergence occurs in the total pressure drop.

Then the flow map parameters according to Taitel and Dukler [T1] are checked:

$$X = \left[\frac{\left(\frac{dP}{dL} \right)_L^{SP}}{\left(\frac{dP}{dL} \right)_g^{SP}} \right]^{\frac{1}{2}} = 5.206 \times 10^{-2}$$

$$F = u_{GS} \cdot \left[\frac{\rho_g}{(\rho_L - \rho_g) \cdot g \cdot D_i} \right]^{\frac{1}{2}} = 0.4608$$

For annular flow, the flow parameters must be $X < 0.1$ and $F > 0.9$.

The diameter of the refrigerant tube is adjusted and rounded off to the nearest standard size, and the calculations are repeated from equation {h2}.

The final results give the following characteristics:

$$d_i = 9 \text{ mm}$$

$$\text{TPD} = 271.75 \text{ Pa/m}$$

$$X = 0.049$$

$$F = 1.155$$

H1.2 heat transfer calculations

H1.2.1 The refrigerant-side heat transfer coefficient

For a mostly annular flow in the evaporator tube, the heat transfer coefficient is related to the Nusselt number by the following relations as suggested by V. Klimenko [M6] (1988):

For nucleate boiling, the parameter Φ must be smaller than 16000. This parameter is given by:

$$\Phi = \frac{G \cdot h_{fg}}{q} \cdot \left[1 + \phi \left(\frac{\rho_L}{\rho_V} - 1 \right) \right] \cdot \left(\frac{\rho_V}{\rho_L} \right)^{\frac{1}{3}} \quad \{h4\}$$

where q is the wall-heat flux given by:

$$q = \frac{Q}{\pi \cdot d_i \cdot L} \quad \{h5\}$$

The refrigerant properties are given by the equations of APPENDIX D for an average temperature by equation {h1}.

thus:

Prandtl number	$Pr_L = 2.78$
wall conductivity	$k_w = 40 \text{ W/m K}$
conductivity	$k_L = 0.551 \text{ W/m K}$
liquid density	$\rho_L = 711.73 \text{ kg/m}^3$
vapour density	$\rho_V = 0.824 \text{ kg/m}^3$
specific heat	$Cp_L = 4346 \text{ J/kg K}$
surface tension	$\sigma = 0.02573 \text{ N/m}$
latent heat	$h_{fg} = 1410.21 \times 10^3 \text{ J/kg}$
pressure	$P = 2.198 \text{ bar}$

Equation {h5} gives: $q = 17692.85 \text{ W/m}^2$

Equation {h6} gives: $\Phi = 48.98$ (hence nucleate boiling dominates)

The Nusselt number is given by:

$$Nu = 0.0074 (q^*)^{0.6} (P^*)^{0.5} (Pr_L)^{-\frac{1}{3}} \left(\frac{k_t}{k_L} \right)^{0.15} \quad \{h7\}$$

where

thermal diffusivity $\alpha_L = \frac{k_L}{\rho_L \cdot Cp_L} = 1.781 \times 10^{-7} \text{ m}^2/\text{s}$

characteristic length (size of bubble) $L_c = \left[\frac{\sigma}{(\rho_L - \rho_V) \cdot g} \right]^{\frac{1}{2}} = 1.921 \times 10^{-3} \text{ m}$

$$q^* = \frac{q \cdot L_c}{h_{fg} \rho_V \cdot \alpha_L} = 193923$$

$$P^* = \frac{P}{[\sigma \cdot g \cdot (\rho_L - \rho_V)]^{\frac{1}{2}}} = 16412.7$$

Substituting into equation {h7}, the Nusselt number is:

$$Nu = 1908.65$$

The two-phase heat transfer coefficient is given by:

$$\alpha_i = Nu \cdot \frac{k_L}{L_c} = 5.47 \times 10^5 \text{ W/m}^2 \text{ K}$$

The forced-convection heat transfer coefficient is neglected as being small in comparison with the two-phase heat transfer coefficient.

H1.2.2 The water-side heat transfer coefficient

Water flows through the annulus of the shell and tube evaporator. The inner diameter of the shell and the outer diameter of the core tube are:

$$D_i = 18 \text{ mm}$$

$$d_o = 12 \text{ mm}$$

The water temperature output is 10°C as a first approximation. The refrigerant properties are calculated from APPENDIX D, for an average temperature:

$$T = \frac{T_{wi} + T_{wo}}{2} = 14^\circ\text{C} \quad \{h8\}$$

The mass flow rate of water is calculated by:

$$m_w = \frac{Q_{ref}}{Cp_w \cdot \Delta T} = 0.030 \text{ kg/s}$$

In the annular space between the shell tube and the inner tube, the water velocity is

$$V_w = 0.211 \text{ m/s}$$

The Reynolds number is:

$$Re_w = 1060.8$$

For water flow in the annular space the heat transfer coefficient is given by Gnielinski's relations:

$$\alpha_c = 0.86 \cdot \left(\frac{D_i}{d_o}\right)^{0.16} \cdot \left(\frac{k_w}{D_H}\right) \cdot \left[3.66 + \frac{0.065 Re_H Pr_w \cdot \left(\frac{D_H}{L}\right)}{1 + 0.04 \left(Re_H Pr_w \cdot \left(\frac{D_H}{L}\right)\right)^{\frac{2}{3}}} \right] = 512.8 \text{ W/m}^2 \text{ K}$$

and

$$\frac{1}{UP} = \frac{1}{\pi \cdot d_i \cdot a_i} + \frac{1}{2\pi \cdot k_t} \ln\left(\frac{d_o}{d_i}\right) + \frac{1}{\pi \cdot d_o \cdot a_c} = 0.0529$$

The number of transfer units is:

$$N_{TU} = \frac{UP \cdot L}{m_w \cdot Cp_w} = 0.302$$

and the effectiveness is given by:

$$\mathcal{E} = 1 - e^{-N_{TU}} = 0.261$$

For an average temperature of the refrigerant $T_b^{\text{ref}} = -13.5^\circ\text{C}$, the water outlet temperature is calculated by:

$$T_{wo} = T_{wi} - (T_{wi} - T_b^{\text{ref}}) \cdot \mathcal{E} = 9.786$$

Iterative calculations from equation {h8} give the following results:

effectiveness	$\mathcal{E} = 0.275$
mass rate of water	$m_w = 0.028 \text{ kg/s}$
water velocity	$V_w = 0.195 \text{ m/s}$
water temperature drop	$\Delta T = 8.7^\circ\text{C}$

H2 PRE-COOLER TUBES

The design is of a shell-and-tube type heat exchanger, with the vapour flowing through the tubes and the liquid through the shell. Choice is made for nineteen vapour-tubes with diameter $d_i=3.0 \text{ mm}$ and thickness 1.5mm in the core of the pre-cooler.

Assume an average temperature in the liquid stream, between points 3 and 4.

$$T_b = \frac{T_3 + T_4}{2} = 12.5^\circ\text{C} \text{ (first approximation)}$$

The physical properties of each stream are evaluated for an average temperature according to equations given in APPENDIX D.

The heat loss across the hot stream is $Q_{PR}=108.29 \text{ W}$

The exit temperature T_4 is evaluated using the energy balance: $Q_{PR} = m_{ref} \cdot C_p \cdot \Delta T$

The temperature T_4 converges to: $T_4 = 0.36^\circ\text{C}$

The pressure at the inlet of the pre-cooler is that of the outlet of the evaporator:

$$P = P_{LOW} - TPD_{EV}$$

Assume a tube-length:

$$L = 0.5\text{m}$$

{h9}

Due to the pressure drop in the tubes of the pre-cooler the average pressure will be:

$$\bar{P} = \frac{2 \cdot (P_{LOW} - TPD_{EV}) - TPD_{PR}}{2} = 2.198 \text{ bar} \quad (\text{first approximation}) \quad \mathbf{\{h10\}}$$

Equations in APPENDICES A and C establish the vapour concentration “y” and its liquid “x” at the pre-cooler, for the average pressure from eq. {h10} and total concentration $x_{ref}=0.9905$ thus:

$$y = 0.99988 \quad x = 0.7014$$

The masses of the liquid and vapour are:

$$m_L = m_{ref} \frac{y - x_{ref}}{y - x} = 2.67 \times 10^{-5} \text{ kg/s}$$

$$m_V = m_{ref} - m_L = 8.2 \times 10^{-4} \text{ kg/s}$$

The vapour quality is calculated as the ratio of the mass of the vapour to the total mass of the refrigerant

$$\phi = \frac{m_V}{m_{ref}} = 0.968$$

The mass velocities are:

$$G = \frac{4m_{ref}}{N \cdot \pi d_i^2} = 6.3098$$

$$G_g = \phi \cdot G = 6.1109$$

The single-phase Reynolds number is:

$$Re_g = \frac{G_g \cdot D_i}{\mu_g} = 2554.9$$

The friction factor and the single-phase pressure gradient are:

$$\xi_g = \left(0.79 \ln(Re_g) - 1.64\right)^{-2} = 4.81 \times 10^{-2}$$

$$\left(\frac{dP}{dL}\right)_g^{SP} = \xi_g \frac{G_g^2}{2\rho_g d_i} = 390.97 \text{ Pa}$$

The total pressure drop gives:

$$TPD_{PR} = \xi_g \frac{G_g^2}{2\rho_g d_i} \cdot L = 781.93 \text{ Pa}$$

Iterative calculations from eq. {h10} yield the following results:

mass of liquid	$m_L = 8.2 \times 10^{-4} \text{ kg/s}$
mass of vapour	$m_V = 2.66 \times 10^{-5} \text{ kg/s}$
average pressure	$P = 2.19 \text{ bar}$
total pressure drop in the pre-cooler	$TPD_{PR} = 782.17 \text{ Pa}$
concentration of vapour	$y = 0.99988$
concentration of liquid	$x = 0.6997$
vapour quality	$\phi = 0.968$

H2.1 Vapour-side heat transfer coefficient

The friction factor converges to:

$$\xi_g = \left(0.790 \ln(Re_g) - 1.64\right)^{-2} = 4.81 \times 10^{-2}$$

The exit temperature T_6 from the pre-cooler is calculated in section 3.4.2: $T_6 = 23^\circ\text{C}$

For an average temperature $T_g = \frac{T_5 + T_6}{2} = 5.35^\circ\text{C}$, the vapour conductance and the Prandtl number

are calculated by the equations in APPENDIX D.

For $Re_H > 2300$ the vapour-side heat transfer coefficient is:

$$a_g = \left(\frac{k_g}{d_i}\right) \frac{\left(\frac{\xi_g}{8}\right) \cdot Pr_g (Re_g - 1000)}{1 + 12.7 \left[\left(Pr_g\right)^{\frac{2}{3}} - 1\right] \cdot \left(\frac{\xi_g}{8}\right)^{0.5}} = 60.39$$

H2.2 Liquid-side (annular flow) heat transfer coefficient

The hydraulic diameter D_H is calculated according to the geometry of the core tubes. With reference to Figure 4.5 in Chapter 4, the internal diameter of the shell tube it is given by:

$$D_i = 5.(d_o + 2t) = 0.050 \text{ m}$$

where $t=2.0$ mm, is the radial increase around each tube to allow for welding.

and

$$D_H = \frac{D_i^2 - N.d_o^2}{D_i - N.d_o} = 0.011$$

For Reynolds number $Re_H = 43.7$, the heat transfer coefficient for the liquid is given by Gnielinski's relations:

$$a_c = 0.86 \left(\frac{D_i}{d_o} \right)^{0.16} \left(\frac{k_w}{D_H} \right) \left[3.66 + \frac{0.065 Re_H Pr_w \left(\frac{D_H}{L} \right)}{1 + 0.04 \left(Re_H Pr_w \left(\frac{D_H}{L} \right) \right)^{\frac{2}{3}}} \right] = 201.8 \text{ W/m}^2 \text{ K}$$

For an effectiveness $\mathcal{E} = 0.7$, the number of transfer units NTU is:

$$N_{TU} = -\ln(1 - \mathcal{E}) = 1.204$$

The UP product is calculated from the relation:

$$\frac{1}{UP} = \frac{1}{\pi \cdot D_i \cdot a_g} + \frac{1}{2\pi \cdot k_t} \ln \left(\frac{D_o}{D_i} \right) + \frac{1}{\pi \cdot D_o \cdot a_s} = 0.1091$$

and the tube length is calculated by:

$$L = \frac{m_g C_{p_g} \cdot N_{TU}}{UP} = 0.5106 \text{ m}$$

Iterative calculations from eq. {h9} convergence to a tube-length of:

$$L = 0.51 \text{ m}$$

APPENDIX I

I1 GENERATOR

The generator is of a shell and tube design capable for 3000 W heat supply. Hot air passes through the core-tubes and the solution is within the shell.

I1.1 The solution-side heat transfer coefficient

Assume a value for the boiler's effectiveness $\mathcal{E} = 0.95$ and an inlet temperature of the hot gases $T_{in} = 950$ K. For a temperature $T_b = 359$ K (the average temperature of the solution in the generator), the outlet and average temperatures of the gases T_{out} and T_m respectively are given by:

$$T_{out} = T_{in} - \mathcal{E}(T_{in} - T_b) = 308.56 \text{ K (first approximation)} \quad \{1\}$$

$$T_m = \frac{T_{in} + T_{out}}{2} = 669.35 \text{ K}$$

Also assume as a design condition, that the wall temperature T_{wall} is well above the bulk temperature of the mixture in the generator and is given by:

$$T_{wall} = \frac{T_b + T_m}{2} = 514.18 \text{ K}$$

According to Unal [U1], the wall temperature can be evaluated from the correlations of the wall superheat during nucleate boiling of binary mixtures.

The ratio of the wall superheat for a real binary mixture to the wall superheat for an ideal binary mixture is given the following relations :

$$b_1 = (1-x) \cdot \ln\left(\frac{1.01-x}{1.01-y}\right) + x \cdot \ln\left(\frac{x}{y}\right) + |y-x|^{1.5} = 1.715$$

$$b_2 = 0 \quad \text{if } x > 0.01$$

for the ammonia-water binary mixture the ratio $P/P_c = 0.07$

$$b_3 = 152 \left(\frac{P}{P_c} \right)^{3.9} = 4.76 \times 10^{-3}$$

$$b_4 = 0.92|y - x|^{0.001} \left(\frac{P}{P_c} \right)^{0.66} = 0.159$$

and

$$\frac{\Delta T_m}{\Delta T_i} = [1 + (b_1 + b_2) \cdot (1 + b_3)] \cdot (1 + b_4) = 3.156 \quad \{i2\}$$

The concentrations of the bulk solution “x” and its vapour “y” are the concentrations x_b and y_{11} respectively, calculated within the simulation model. Thus:

$$x = 0.369$$

$$y = 0.960$$

The saturation temperatures of pure substances at the generator pressure are:

$$T_A = 297.2 \text{ K and } T_W = 452.4 \text{ K}$$

The ideal solution superheat is calculated from:

$$\Delta T_1 = T_{wall} - T_A = 216.98$$

$$\Delta T_2 = T_{wall} - T_W = 61.81$$

$$\Delta T_i = \Delta T_1(x) + \Delta T_2(1 - x) = 119.06 \quad \{i3\}$$

From eq. {i2} and eq. {i3} the actual mixture wall superheat ΔT_m is given by:

$$\Delta T_m = \left(\frac{\Delta T_m}{\Delta T_i} \right) \cdot \Delta T_i = 375.82$$

A choice is made for the following design parameters:

number of tubes	$N=7$
outside diameter	$d_o = 0.015 \text{ m}$
tube-thickness	$t = 0.0015 \text{ m}$
tube length	$L = 1.2 \text{ m}$

The heat transfer rate and the boiling heat transfer coefficient are:

$$\dot{Q} = \frac{Q_{in}}{N \cdot L \cdot \pi \cdot d_o} = 7582.65 \text{ W/m}^2$$

$$\alpha_s = \frac{\dot{Q}}{\Delta T_m} = 20.18 \text{ W/m}^2 \text{ K}$$

11.2 The gas-side heat transfer coefficient

The physical properties for the hot-air are calculated from APPENDIX D [M6] for the average temperature T_m . Thus:

conductivity	$k_{\text{air}} = 4.872 \times 10^{-2}$
density	$\rho_{\text{air}} = 0.528$
specific heat	$C_{p_{\text{air}}} = 1056.3$
viscosity	$\mu_{\text{air}} = 3.19 \times 10^{-5}$
Prandtl number	$Pr_{\text{air}} = 0.69$

The velocity of the hot-air is given by:

$$V_g = \frac{4\dot{Q}_{gen}}{N \cdot (\pi \cdot d_i^2) \cdot \rho_g \cdot C_{p_g} \cdot (T_{in} - T_{out})} = 12.1 \text{ m/s}$$

which yields a Reynolds number $Re = 2402.3$

The friction factor and the vapour-side heat transfer coefficient can be evaluated.

$$\xi_g = (0.790 \ln(Re) - 1.64)^{-2} = 6.15 \times 10^{-3}$$

$$\alpha_g = \left(\frac{k_g}{D_i} \right) \frac{\left(\frac{\xi_g}{8} \right) \cdot Pr_g (Re - 1000)}{1 + 12.7 \left[\left(Pr_g \right)^{\frac{2}{3}} - 1 \right] \left(\frac{\xi_g}{8} \right)^{0.5}} = 30.89$$

and

$$\frac{1}{U\mathcal{P}} = \frac{1}{\pi \cdot D_i \cdot \alpha_g} + \frac{1}{2\pi \cdot k_t} \ln \left(\frac{D_o}{D_i} \right) + \frac{1}{\pi \cdot D_o \cdot \alpha_s} \quad \{i4\}$$

The number of transfer units N_{TU} is calculated from the effectiveness value as:

$$N_{TU} = -\ln(1 - \mathcal{E}) = 2.997$$

The UP product is also calculated from the relation:

$$\frac{1}{UP} = \frac{N_{TU} \cdot m_g C_{p_g}}{N \cdot L} = 1.905 \quad \{i5\}$$

The UP product calculated by eq. {i5} is dependent to the inlet temperature T_{in} of eq. {i1} while the UP product calculated by eq. {i4} is directly dependent on the boiler heat transfer characteristics and fluid properties.

The inlet temperature of the gases T_{in} is adjusted in eq. {i1} until the two UP products coincide in value.

The iteration yields the following:

velocity of gases	$V_{air} = 12.1 \text{ m/s}$
mass rate of gases	$m_{air} = 5.06 \text{ kg/s}$
generator's effectiveness	$\mathcal{E} = 0.950$
inlet temperature of gases	$T_{in} = 676^\circ\text{C}$
outlet temperature of gases	$T_{out} = 115^\circ\text{C}$

APPENDIX J

J1 DISTILLATION COLUMN The M^cCabe-Thiele Method [F1]

With reference to the enthalpy-concentration diagram of Figure 3.15, the reflux ratio is:

$$R = \frac{h_{\pi} - h_1}{h_1 - h_3} = 0.122$$

The top and boiler plates' temperatures are $T_{\text{top-plate}} = 63.4^{\circ}\text{C}$ and $T_{\text{boiler}} = 86.6^{\circ}\text{C}$ respectively.

For a sieve column, the molal vapour density is given by:

$$\rho_{mv} = \frac{\left(\frac{T_{\text{top-plate}} + T_{\text{boiler}}}{2} \right)}{273.15} \cdot \frac{1}{22.4} = 5.69 \times 10^{-2}$$

The indices 1 and 2 refer to the top plate and the bottom plate (generator) respectively.

Thus for the concentrations $x_1 = 0.990$ (distillate) and $x_2 = 0.433$ (strong solution), the ammonia and water molal liquid densities are:

$$\rho_{A1} = 572 \text{ kg/m}^3$$

$$\rho_{A2} = 550 \text{ kg/m}^3$$

$$\rho_{W1} = 980 \text{ kg/m}^3$$

$$\rho_{W2} = 973 \text{ kg/m}^3$$

The molal density of the solution is given by:

$$\rho_{ml} = \frac{1}{2} \cdot \left\{ \left[x_1 \frac{\rho_{A1}}{17} + (1 - x_1) \frac{\rho_{W1}}{18} \right] + \left[x_2 \frac{\rho_{A2}}{17} + (1 - x_2) \frac{\rho_{W2}}{18} \right] \right\} = 39.26$$

The Liquid-Vapour ratio L/V is given by:

$$\frac{L}{V} = \frac{R}{1 + R} = 0.109$$

The intercept Φ for the rectifying column is:

$$\Phi = \frac{L}{V} \sqrt{\frac{\rho_{mv}}{\rho_{ml}}} = 4.15 \times 10^{-3}$$

From the flooding curves for sieve-trays (Figure 8.10 of [F1]), and for a sieve-tray spacing of 100mm, the flooding parameter is estimated as:

$$\lambda_F = 0.02$$

The flooding velocity is given by:

$$u_F = \frac{\lambda_F}{\sqrt{\frac{\rho_{mv}}{\rho_{ml} - \rho_{mv}}}} = 0.525 \text{ m/s}$$

The vapour velocity up the column is given by:

$$V = (R + 1) \cdot m_{ref}^{molar} = 5.6 \times 10^{-5} \text{ m/s}$$

The diameter of the sieve-plate is given by:

$$D_{sieve} = \sqrt{\frac{4V}{0.72u_F \cdot \pi \cdot \rho_{mv}}} = 0.057 \text{ m}$$

APPENDIX K

K1. CORRELATIONS FOR THE INTERFACIAL AREA IN CO-CURRENT GAS-LIQUID FLOW

Chemical absorption of carbon dioxide into sodium hydroxide solutions has been used for the measurement of the interfacial area of two-phase flows. The measured values were correlated against different parameters.

Banerjee [B3] correlated the values of the interfacial area against the frictional energy dissipation parameter E_B , [108].

$$\alpha = n(E_B)^m$$

Where E_B , is the dissipation parameter which according to [B3] is based solely on the liquid phase and is given by:

$$(k1) \quad E_B = \left(\frac{\Delta P}{\Delta L} \right)_{TP} Q_L$$

The dimensions of α , E_B and Q_L are cm^{-1} , Nm/s and cm^3/s respectively.

The values of n and m differ with various coils that have been used in their experiment:

For 18in coil	$n=6.18$	$m=0.4018$
For 12in coil	$n=7.9$	$m=0.642$
For 6in coil	$n=4.43$	$m=0.557$

A similar work by Kulic and Rhodes [K13] as that of Banerjee's [B3], correlates the interfacial area with Jepsen's energy dissipation parameter E_J :

$$\alpha = 4.54(E_J)^{0.262}$$

and

$$(k2) \quad E_J = \left(\frac{\Delta P}{\Delta L} \right)_{TP} (u_{LS} + u_{GS})$$

where u_{LS} and u_{GS} are the liquid and gas superficial velocities.

Gregory and Scott [G3] discuss in their paper the correlations that exist, between pressure drop, overall mass transfer coefficient $k_L^o a$, and individual mass transfer coefficient k_L^o . Their model analyses both chemical and physical gas absorption for the determination of the interfacial area between gas and liquid.

The investigation on mass transfer with chemical reaction is based on the Higbie penetration theory. The interfacial area was expressed in terms of the liquid-phase physical mass transfer coefficient k_L^o . The expressions used were based on Danckwerts solutions for the two types of chemical reaction, that is, a pseudo first order reaction and an infinitely rapid second order reaction.

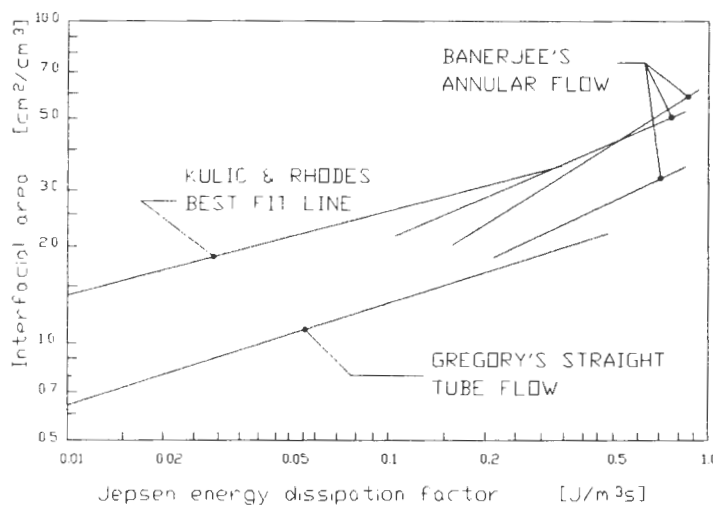


Figure K 1 Relationship between the interfacial area and Jepsen's energy dissipation factor

Kulic and Rhodes used the correlation of eq. (k2) and obtained a best fit line between data points, relating the interfacial area to Jepsen's energy dissipation factor E_J . They also compared graphically their results to those by Banerjee and by Gregory and Scott, Figure K1. The three plots compare favourably with one another, if apparatus geometry and flow patterns are taken into account.

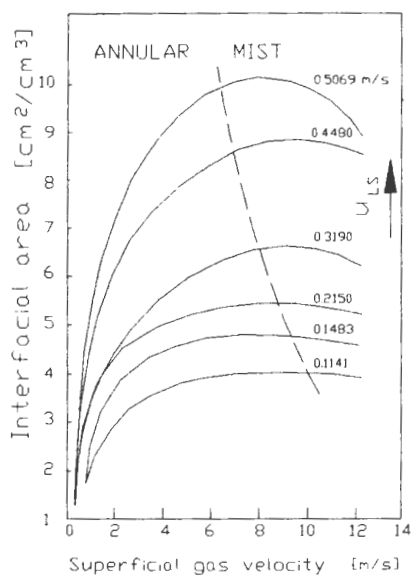


Figure K 2

Kasturi and Stepanek [K1] obtained a correlation of the interfacial area for a 6mm vertical tubes in terms of the frictional pressure drop. They experimented on vertical 1524mm high tube, obtaining data for annular, slug and annular-mist flows. The dependence of the interfacial area on the gas and liquid velocities per unit tube volume is shown in Figure K2. The lines on the plot were determined by visual observation of the flow and their accuracy is not high.

Figure K2 shows that the interfacial area increases with the increase in the liquid and gas velocities up to a certain value and then it decreases. The maxima observed are due to a combined effect of liquid dispersion and holdup.

Their work has also proved, that the co-current flow contactors, support a much larger interfacial area than the equivalent conventional packed and bubble columns. They report values of interfacial areas in the range of 120-1000 m^2/m^3 and claim that if high interfacial areas are required then the flow must be in the region of annular to annular-mist.

Sharma and Danckwerts [S6] report that the interfacial area in a packed column can range between 20 m^2/m^3 to 350 m^2/m^3 .

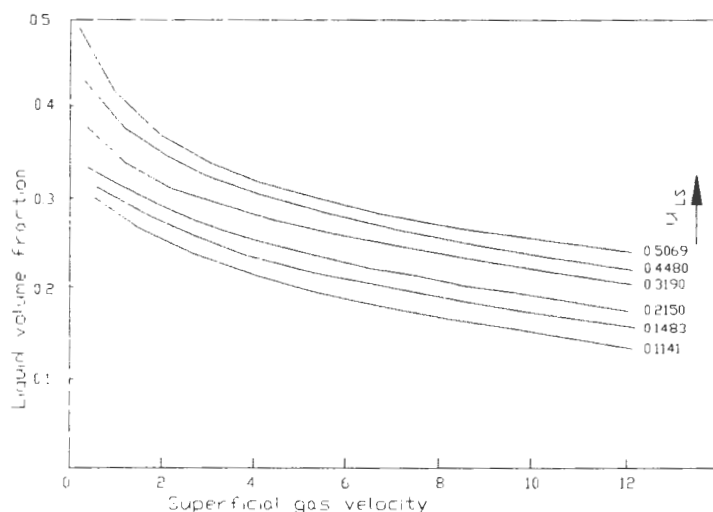


Figure K 3 Relationship between the liquid void fraction ($1-\epsilon$) and the superficial gas and liquid velocities

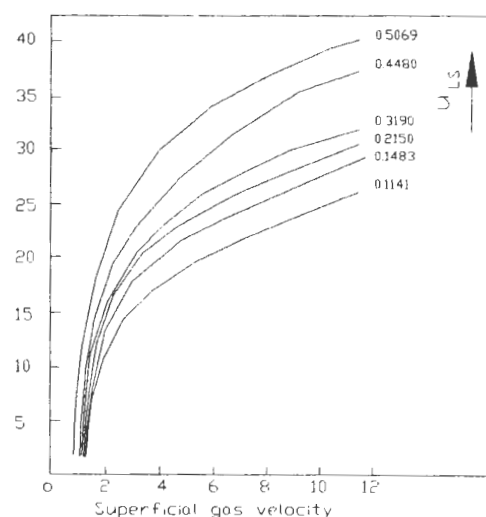
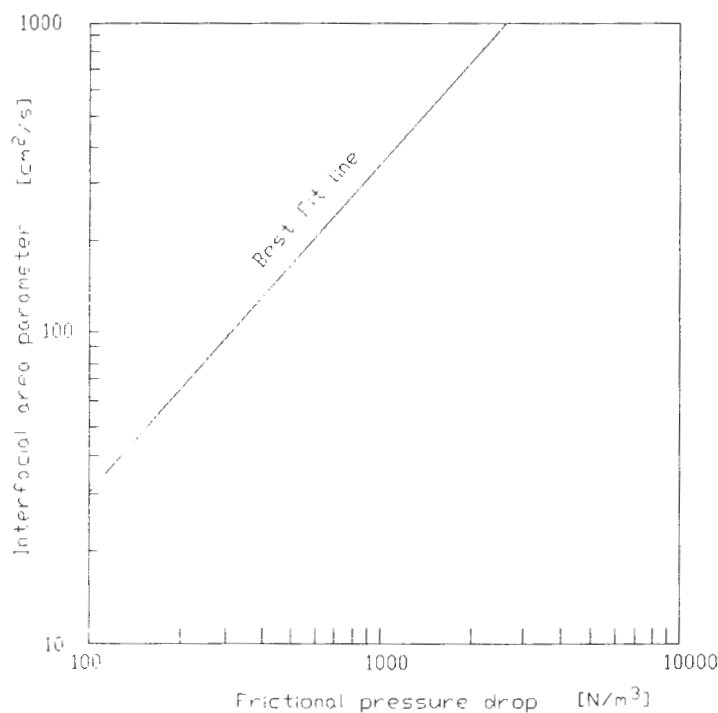


Figure K 4 Relationship between the liquid void fraction related interfacial area $\alpha/(1-\epsilon)$ and the superficial gas and liquid velocities

Figure K3 relates the gas velocity with the liquid void fraction ($1-\epsilon$). The values in Figure K2 are divided by those in Figure K3 for the same hydrodynamic conditions, to produce Figure K4 which shows that there is a continual growth of interfacial area with increasing gas velocity. The curves show a leveling off due to the degree of dispersion becoming so high that the amount of energy required to overcome surface tension in the mist flow is enormous.

The experimental interfacial areas were compared with the correlations proposed by Jepsen and Banerjee, eq. (k1) and (k2) respectively, but it was found on both occasions to have considerable scatter. Very little scatter was indicated when a new parameter $Q_L \alpha / (1-\epsilon)$, defined as "the rate at which the interfacial area is created", was plotted against the frictional pressure drop. It was thought that mechanical energy is required to create the interfacial area, which is then lost irreversibly at the outlet of the tube. This mechanical energy can be related to the frictional pressure drop [K1].

The correlation proposed by Kasturi and Stepanek is almost a linear expression which represents the best fit line in Figure K5.



$$Q_L \frac{\alpha}{1-\epsilon} = 0.226 \left(\frac{\Delta P}{\Delta L} \right)_L^{1.07}$$

Shilimkan and Stepanek [S7] continued the work by Kasturi and Stepanek by proposing correlations for tubes of 10mm, 15mm and 20mm ID.

It was observed that the dependence of the interfacial area on the gas velocity deteriorates as the diameter of the tube increases. A similar attempt to correlate their data by Banerjee's and Jepsen's parameters also gave excessive scatter.

Figure K 5 The correlation of the interfacial area as proposed by [K1]

A very successful correlation was achieved when Kasturi and Stepanek's method was applied to all tubes:

$$Q_L \frac{\alpha}{1-\varepsilon} = n \left(\frac{\Delta P}{\Delta L} \right)_L^m$$

the dimensions of the interfacial parameter is in cm^2/s and the frictional pressure drop in N/m^3

The values of “n” and “m” are defined according to the tube diameter as shown in Table K1

Table K1				
	6mm [K1]	10mm	15mm	20mm
n	226	361	2190	3280
m	1.07	1.31	2.18	1.92

APPENDIX L

L1. MASS TRANSFER IN TWO-PHASE GAS-LIQUID FLOW

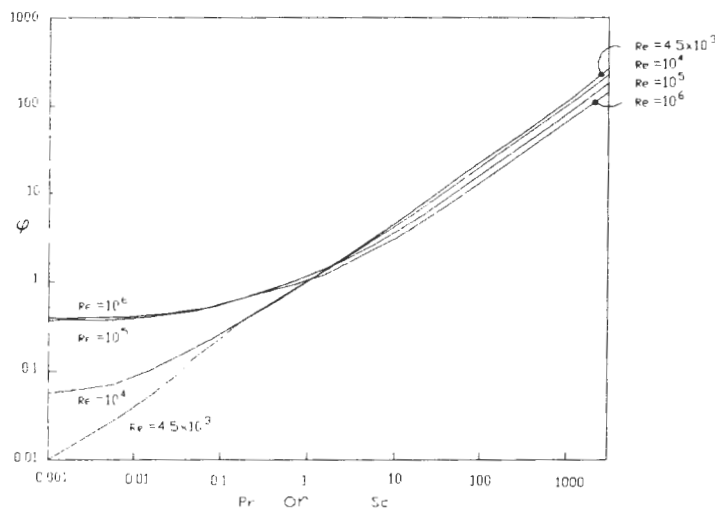
The relations proposed by Kropholler and Carr [K11] are based on a modified form of the Reynolds analogy¹. They were derived in terms of the Stanton, Nusselt, Sherwood, Reynolds, Prandtl and Schmidt numbers, [I1, K11] as follows:

$$Nu = \frac{f}{\phi_H} Re.Pr \quad \text{For heat transfer} \quad Sh_m = \frac{f}{\phi_D} Re.Sh \quad \text{For mass transfer} \quad (I1)$$

Sh_m = modified Sherwood number

ϕ_H = function of the Prandtl number Pr

ϕ_D = function of the Schmidt number Sc



The functions ϕ_H and ϕ_D can be evaluated by the equations derived in [K11] or from the plot of Figure L1, for smooth pipes.

When the fluid was in turbulent motion, eq. (I1) was modified to account for the two-phase flow as:

$$Sh = \frac{f}{\phi} \frac{D_{hG} \cdot u_G}{D_G}$$

Figure L 1 For heat transfer $\phi = \phi_H$ correlates with Pr . For mass transfer $\phi = \phi_D$ correlates with Sc

¹ Chilton and Colburn [C3] in 1934 experimentally proved the Reynolds analogy between mass and momentum transfer:

$$Sh = 0.04 Re^{0.75} Sc^{\frac{1}{3}}$$

This relation applies to gas controlled mass transfer in wetted-wall columns, but not when shear layers exist at the gas-liquid interface.

Ciborowski and Rychlicki [C4] investigated mass transfer in horizontal two-phase gas-liquid wavy flow in round pipes. They developed successful correlations for both gas and liquid mass transfer coefficients for the fluid under turbulent motion. Their theory was based on the mass-momentum analogy of Kropholler and Carr approach [K11].

For the gas-film resistance the correlations are:

$$Sh_h = 0.289 \times 10^{-4} \cdot Re_G^{1.615} \cdot Sc^{0.5} \quad \text{The slip correlation}$$

$$Sh_h = 0.3 \times 10^{-4} \cdot Re_G^{1.46} \cdot Sc^{0.5} \quad \text{The slip-wave correlation}$$

$$Sh_h = 0.00413 Re_G^{0.916} \cdot Sc^{0.5} \quad \text{The liquid film resistance correlation}$$

The model used by Banerjee et al. [B2, B3] proposed a relation which predicts mass transfer rates from a knowledge of some of the fluid properties. The individual mass transfer coefficient k_L^o was shown to be a function of the root mean square vorticity near the interface of a gas-liquid annular or slug-annular flow in coiled tubes. The expression relating the mass transfer coefficient to the root mean square vorticity is:

$$(12) \quad k_L = \sqrt{D_A \cdot \sqrt{\left[\left(\frac{\Delta P}{\Delta L} \right)_f \cdot \frac{R}{2\rho_L} \right]} \cdot \frac{2.25\pi \cdot R \cdot V_L}{Q_L}}$$

where

R = tube radius

V_L = average liquid velocity

This relation could also be applied to thin turbulent films. In cases other than thin turbulent liquid films, such as in thick films that occur in slug flows, the expression deviates by far from the experimental data. Gregory and Scott [G3] report on a best-fit line to the experimental data points, given by the expression:

$$k_L^o = 3.127 \sqrt{D_A \cdot \sqrt{\left[\left(\frac{\Delta P}{\Delta L} \right)_f \cdot \frac{R}{2\rho_L} \right]} \cdot \frac{2.25\pi \cdot R \cdot V_L}{Q_L}} - 0.05$$

Gregory and Scott [G3] modified Banerjee's expression which corresponds to thicker films. They proposed a model of the dissipation of vortex energy which represents in a better way the physical model of a horizontal slug flow. Their expression corresponding to that of Banerjee's is:

$$k_L^o = \sqrt{D_A \cdot \left[\left(\frac{\Delta P}{\Delta L} \right)_f \cdot \frac{R \cdot \sqrt{\varepsilon}}{2\mu \cdot n} \right]}$$

where

$$n = \text{averaging factor dependant on the Reynolds number: } n = 10^9 \cdot (\text{Re})_{TP}^{-1.7}$$

The main difference between Gregory's model and that of Banerjee's, is that the mass transfer

coefficient is proportional to the $\left(\frac{\Delta P}{\Delta L} \right)_f^{\frac{1}{2}}$ instead of $\left(\frac{\Delta P}{\Delta L} \right)_f^{\frac{1}{4}}$

Jepsen [J3] experimented with oxygen absorption and desorption on co-current air-water flow in different diameter horizontal tubes and coils. The objective was to determine the effect of gas-liquid flow rates, tube diameter and tube orientation on the rate of liquid-phase mass transfer in co-current horizontal two-phase flow. A correlation was used for $k_L \alpha$ as a function of the two-phase frictional energy dissipation, liquid-phase physical properties and tube diameter. The liquid-phase mass transfer correlations are:

$$(13) \quad k_L \alpha \dots D^{-0.5} \cdot \sigma^{-0.5} \cdot \mu_L^{-0.05} \cdot d^{0.68} = 1.33 (E_J^{0.4}) \quad \text{for } E_J < 5000 \text{ N/m}^2 \text{s}$$

$$(14) \quad k_L \alpha \dots D^{-0.5} \cdot \sigma^{-0.5} \cdot \mu_L^{-0.05} = 0.59 (E_J^{0.79}) \quad \text{for } E_J > 5000 \text{ N/m}^2 \text{s}$$

Kulic and Rhodes [K13] used Jepsen's work to correlate the mass transfer coefficient $k_L \alpha$ with the frictional pressure drop $\left(\frac{\Delta P}{\Delta L} \right)$ and the energy dissipation E_J .

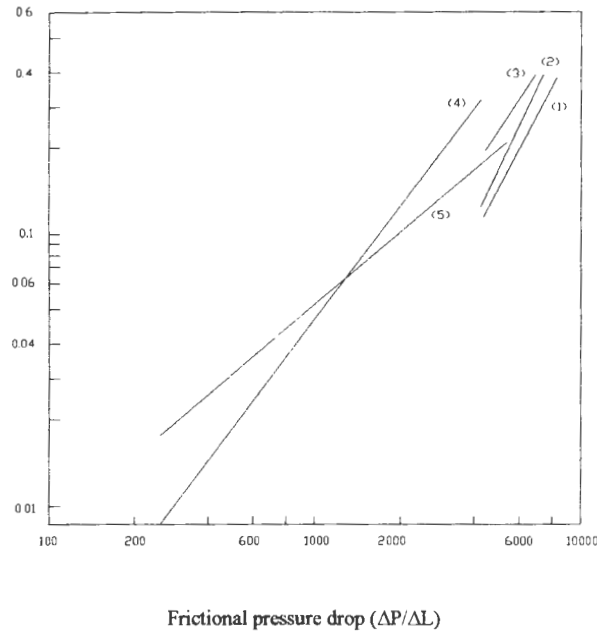


Figure L 2 (1), (2), (3) Banerjee: annular flow in coils

(4) Gregory: horizontal slug flow

(5) Kulic: slug flow in coils

Figure L2 shows comparatively Banerjee and Kulic's relationship between the frictional pressure drop and the volumetric mass transfer coefficients.

Figure L3 shows comparatively Gregory and Kulic's relationship between the volumetric physical mass transfer and Jepsen's energy dissipation factor.

Kulic's proposed straight line correlations for the relationships of Figures L2 and L3 are:

$$k_L \alpha = 34.6 \left(\frac{\Delta P}{\Delta L} \right)^{0.84}$$

$$k_L \alpha = 0.244 (E_J^{0.546})$$

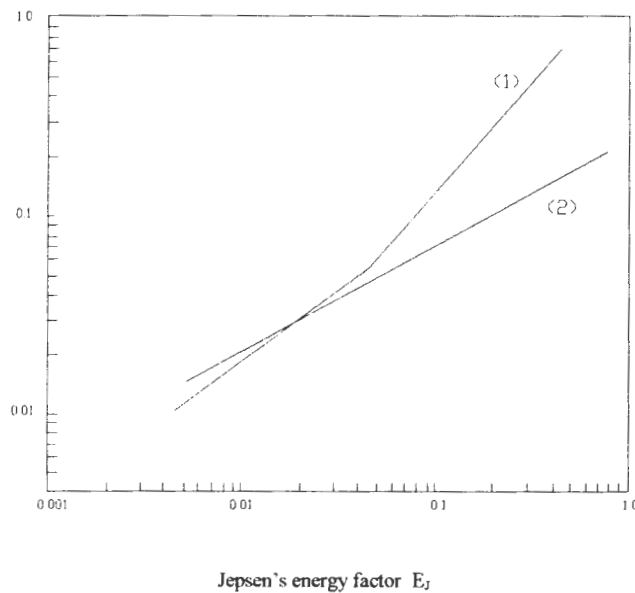


Figure L 3 (1) Kulic: slug flow in coils

(2) Gregory: horizontal slug flow

Jagota et al. [J1] experimented with the absorption of carbon dioxide in water, in vertical gas-liquid annular flow. The volumetric mass transfer coefficient was calculated using Banerjee's theory by an equation similar to eq. (I2):

$$k_L \alpha = \alpha \cdot \sqrt{D_A} \cdot \sqrt{\left[\left(\frac{\Delta P}{\Delta L} \right)_f \cdot \frac{R}{2\rho_L} \right] \cdot \frac{2.25\pi \cdot R \cdot V_L}{Q_L}}$$

The values obtained were correlated with Jepsen's energy factor but it failed to correlate all the data on a single curve.

Since the resistance to mass transfer was in the liquid-phase, Jagota et al. proposed a correlation which resulted in a single straight line through the data points.

$$k_L \alpha = 5.2 \times 10^6 \left[\left(\frac{\Delta P}{\Delta L} \right)_f \cdot u_{SL} \right]^{0.48}$$

Mass transfer coefficients in gas-liquid co-current vertical flow, have also been obtained by Kasturi and Stepanek [K2], Shilimkan and Stepanek [S8, S9] and by Tomida et al. [T3, T4]. The investigations by Stepanek et al. were on flows in tubes of ID 6mm, 10mm, 15mm and 20mm. The flows were mostly for annular flow and some for slug and annular-mist regimes.

The experiments by Tomida were on flows in tubes of ID 10mm, 18mm and 25mm mainly in the froth and wavy-annular regimes.

In all studies, correlations for the volumetric mass transfer coefficient were developed. Only in the literature [K2, S8, S9] correlations for the gas mass transfer coefficient k_G and k_L were obtained. The true mass transfer coefficients could be obtained from the k_L and k_G values and the interfacial area values. The interfacial areas were calculated by the chemical absorption methods.

L2 CORRELATIONS BY KASTURI AND STEPANEK [K1, K2], SHILIMKAN AND STEPANEK [S8, S9]

Kasturi and Stepanek [K2] calculated the volumetric mass transfer coefficient k_L by a diffusion controlled relation suggested by Sharma and Danckwerts [S6]:

$$(15) \quad R \cdot \alpha = k_L \alpha \cdot C^*$$

where

C^* = the concentration of the gas at the interface, the values of which were obtained by [K2] using the experimental method of [S6].

R = the volumetric rate of absorption also discussed in [S6].

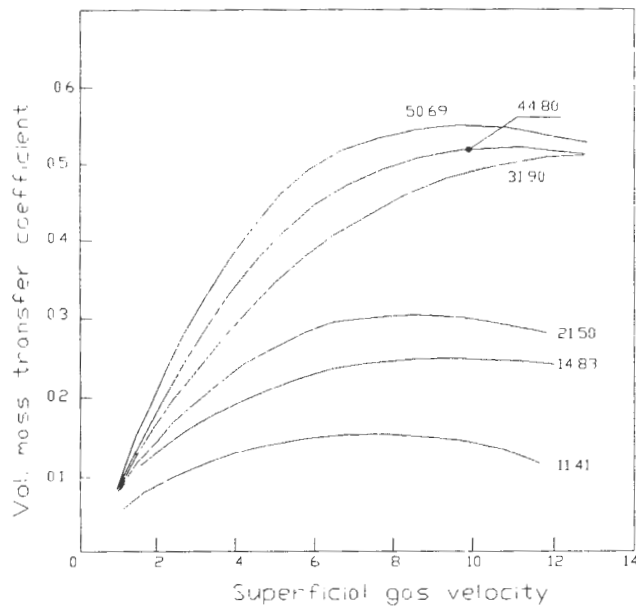


Figure L 4 Relationship between $k_L \alpha$ and the superficial velocities for 6mm ID tube

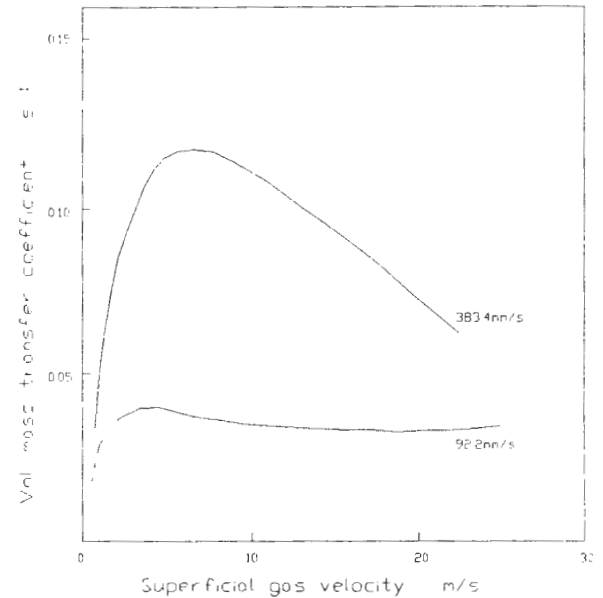


Figure L 5 Relationship between $k_L \alpha$ and the superficial velocities for 10mm ID tube

The values of the volumetric mass transfer coefficient $k_L \alpha$ were plotted against the superficial gas velocity u_{GS} for various superficial liquid velocities u_{LS} Figures L4 to L7. In the small tubes, there is a marked dependence of the volumetric mass transfer coefficient and the gas velocity. This dependence is very much reduced in the 20mm ID tube. In the plots for 6mm, 10mm and 15mm tubes, a maximum on every curve defines a region of decreasing values of $k_L \alpha$ when u_{GS} increases. This phenomenon is more pronounced in the 10mm ID tube results. A similar observation was made on the interfacial area curves in Figure K2 where the maxima occur for almost the same u_{GS} values. The decrease is due to the formation of droplets entrained in the core of the gas flow which due to their rigidity have a low mass transfer coefficient. This theory is well supported by the fact that the maxima on the curves are in close proximity to the critical gas velocities for liquid entrainment. Zhivaikin [S9] produced a relation which is shown as the flow separating line in Figure K2.

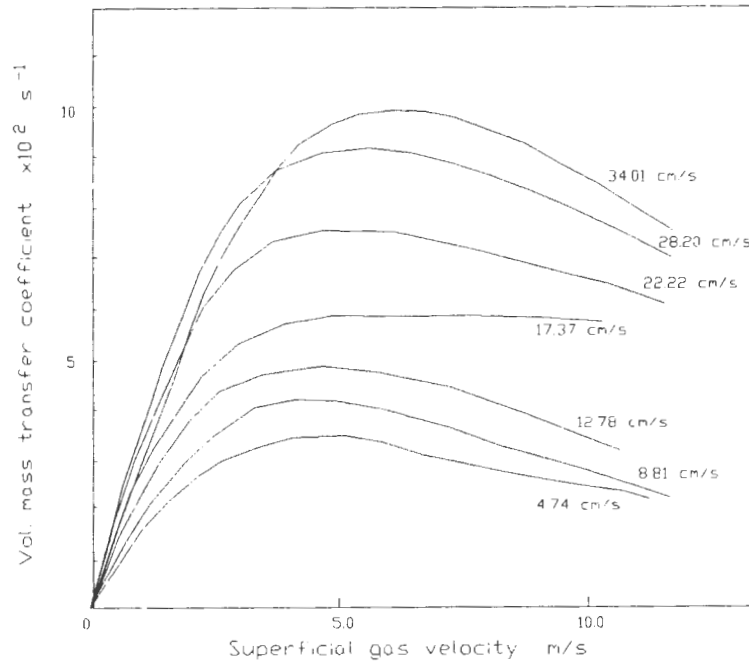


Figure L 6 Relationship between $k_L\alpha$ and the superficial velocities for 15mm ID tube

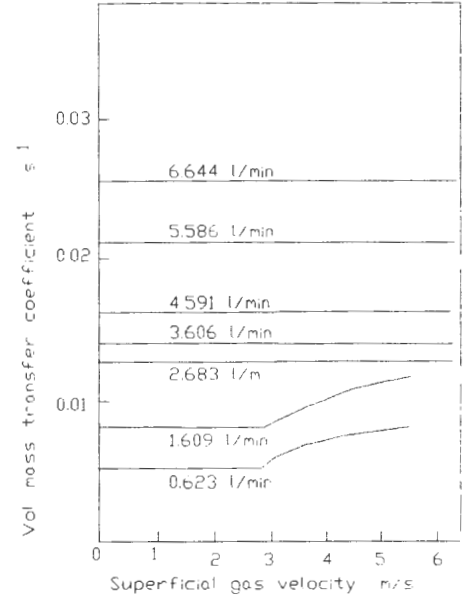


Figure L7 Relationship between $k_L\alpha$ and the superficial velocities for 20mm ID tube

Shilimkan and Stepanek tested Banerjee's suggestion to correlate the volumetric mass transfer coefficient with the frictional pressure drop. Thus for the annular flow (ascending part of Figure L6) :

$$k_L\alpha = 5.9 \times 10^5 \left(\frac{\Delta P}{\Delta L} \right)_f^{1.57} \quad \text{for 15mm ID tube}$$

$$k_L\alpha = 9.3 \times 10^6 \left(\frac{\Delta P}{\Delta L} \right)_f^{1.82} \quad \text{for 20mm ID tube}$$

For the annular-mist flow (descending part of Figure L6):

$$\frac{k_L\alpha}{Q_L} = 10^{-2} \left(\frac{\Delta P}{\Delta L} \right)_f^{-0.36} \quad \text{for 10mm ID tube}$$

The correlations for the 15mm and 20mm tubes gave a standard deviation of 15% while the correlation for the 10mm tube gave a standard deviation of 12%.

When the volumetric mass transfer coefficient was correlated with Jepsen's energy dissipation parameter the correlation gave a standard deviation of 25% for the ascending part of the curves. For the descending part, the correlation failed completely.

The values of the true mass transfer coefficient k_L were found by dividing the values of the volumetric mass transfer coefficient by the interfacial area obtained for the same hydrodynamic conditions.

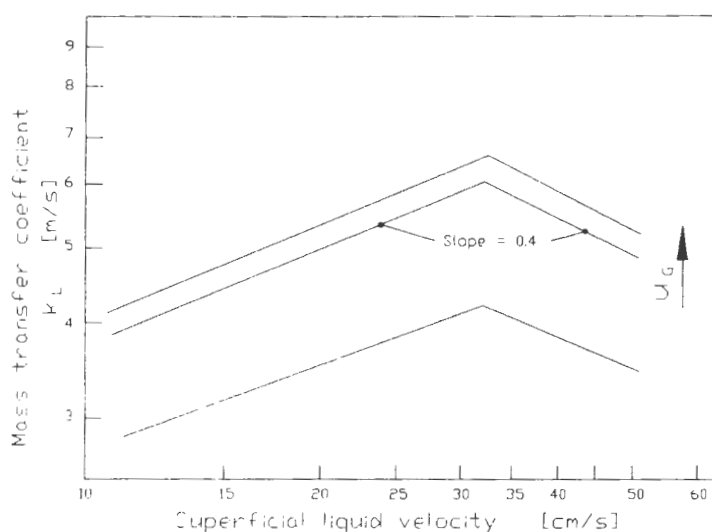
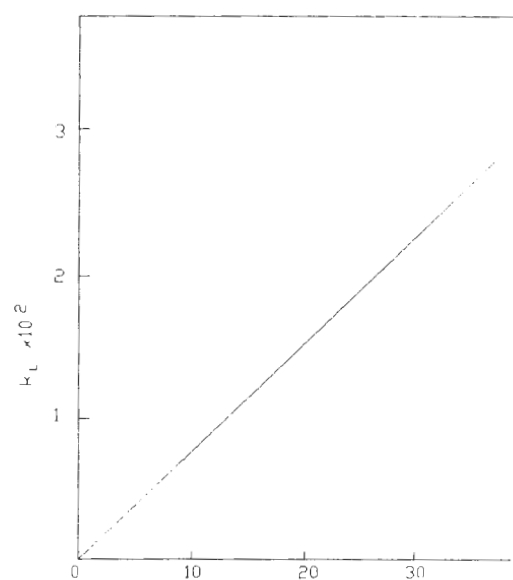


Figure L 8 Relationship between k_L and superficial velocities



$$\left(u^*\right)^{1.5} D^{0.5} \rho_L^{0.5} \sigma^{-0.5}$$

Figure L 9 Correlation of k_L according to the Levich theory of turbulence, for 10mm, 15mm and 20mm, ID tubes

Figure L8 shows such a relationship for the flow in the 6mm ID tube. The increasing and decreasing trends were also observed by Lamont and Scott [S9] who report a slope of 0.45 for bubble flow. The peak was reported to occur due to the change of flow pattern from annular to annular mist, brought by increased entrainment.

Kasturi and Stepanek used an equation, derived by Davies from the Levich model of damped turbulence [S8] to calculate the true mass transfer coefficient k_L , shown in Figure L9:

$$k_L = 0.32 \sqrt{\frac{D_L \cdot (u^*)^3 \cdot \rho_L}{\sigma}}$$

where

u^* is the friction velocity related to the interfacial shear stress by the expression: $(u^*)^2 = \frac{\tau_i}{\rho_L}$

However the predicted values were much lower than the experimental by a factor ranging from 1.5 to 10 times. Even with a correction factor proposed by Davies, there were still values differing by a factor up to 7 times.

A better agreement is achieved for the tube with 6mm ID with Banerjee's formula, eq. (I2). The difference between the predicted and the experimental results ranges from 1.5 to 2 times.

The same magnitude of agreement was encountered when the mass transfer coefficient was correlated with Jepsen's energy dissipation factor.

Kasturi and Stepanek [K2] suggested a model based on the analogy between momentum and mass transfer and on the penetration theory. This gave a correlation for the true mass transfer coefficient as:

$$(I6) \quad k_L = \frac{\tau_i}{u_L} \sqrt{\frac{D_L}{\mu_L \cdot \rho_L}} \approx \frac{\tau_w}{u_L} \sqrt{\frac{D_L}{\mu_L \cdot \rho_L}}$$

The interfacial shear stress was approximated by the wall shear stress which in turn was expressed in terms of the frictional pressure loss. The frictional pressure drop was calculated according to the correlation in eq. (I2). This correlation gave 20% deviation between the predicted and the experimental results. According to Kasturi and Stepanek, this could be due to the approximation of the interfacial shear stress to the wall shear stress. In vertical upward flow the interfacial shear is higher than the wall shear.

On the gas-side mass transfer correlations, plots for the 10mm and 20mm ID tubes are shown in Figures L10 and L11 respectively. The plot for the 15mm tube showed an identical trend to that for the 10mm tube but the spreading for different liquid velocities started at a lower gas velocity. As it is shown in Figure L11 there is no spreading, probably because all the data was collected for low gas

velocities [S9]. It is also observed that at the investigated region of gas velocities, the variation of liquid velocities had an adverse effect on the experimental data, but no reason was given for such an effect.

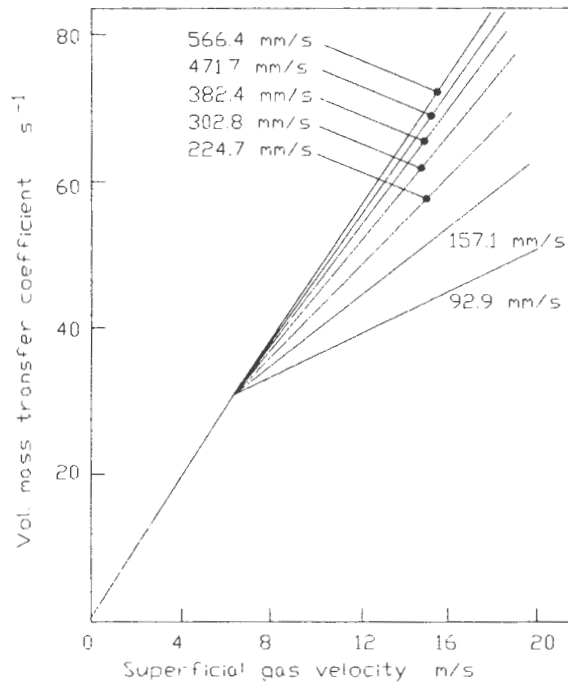


Figure L10 Relation of the gas-side mass transfer coefficient $k_G\alpha$ with the superficial velocities for 10mm ID tubes

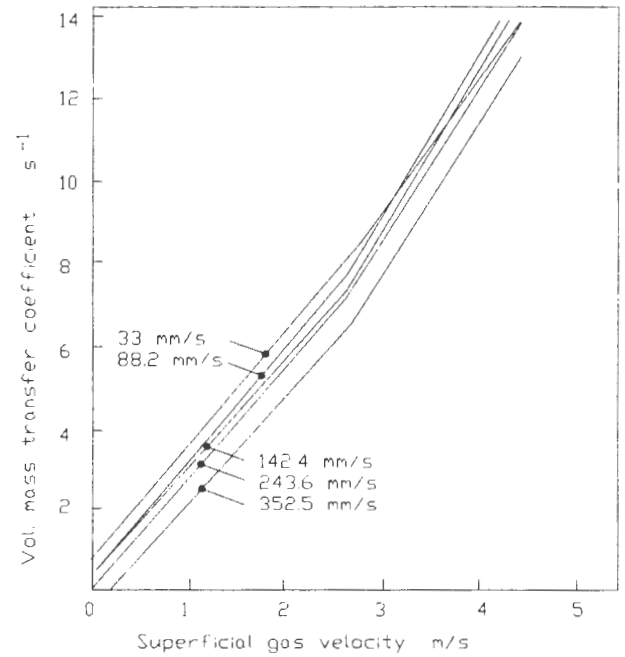


Figure L11 Relation of the gas-side mass transfer coefficient $k_G\alpha$ with the superficial velocities for 20mm ID tubes

The correlations for the volumetric mass transfer coefficient were in terms of a parameter related to the frictional pressure drop:

$$k_G\alpha \cdot D \cdot u_{LS}^{\frac{1}{3}} = 3.933 \times 10^{-7} \left[\left(\frac{\Delta P}{\Delta L} \right)_f \cdot u_G \right]^{0.923} \quad \text{for } \left(\frac{\Delta P}{\Delta L} \right)_f \cdot u_G < 104 \text{ N/m}^2\text{s}$$

$$k_G\alpha \cdot D \cdot u_{LS}^{\frac{1}{3}} = 4.353 \times 10^{-6} \left[\left(\frac{\Delta P}{\Delta L} \right)_f \cdot u_G \right]^{0.642} \quad \text{for } \left(\frac{\Delta P}{\Delta L} \right)_f \cdot u_G > 104 \text{ N/m}^2\text{s}$$

The deviations observed by using the above correlations are 10%.

The Levich model of damped turbulence, which gave a good match between the predicted and the experimental data, for the liquid-side correlations, failed in the gas-side correlations. This is due to the differences in the mechanisms governing the mass transfer in the liquid annulus and in the gas core.

Shilimkan and Stepanek [S9] correlated their data in terms of dimensionless numbers:

$$Sh_G = 3.29 \times 10^{-4} (Re_G)^{0.756}$$

With this correlation the standard deviation is 23%.

It was also noticed that the exponent of the Reynolds number is close to 0.8, which is frequently used in heat and mass transfer correlations in turbulent flows.

Kasturi and Stepanek [K2] correlated their data for the 6mm ID tube by using an equation similar to eq. (75) for the liquid-side:

$$R.\alpha = k_G \alpha . p$$

where

p = the partial pressure of the solute in the gas.

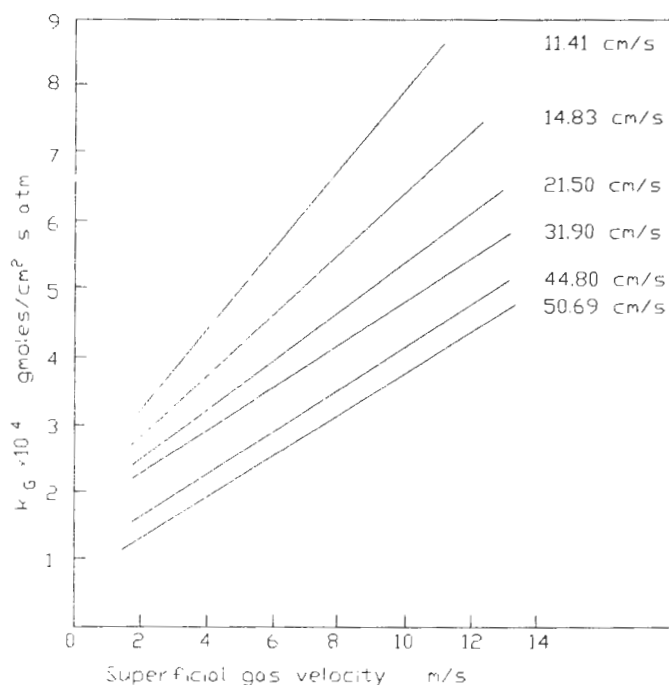


Figure L 12

The true gas-side mass transfer coefficient was calculated by dividing the volumetric mass transfer coefficient by the interfacial area under the same hydrodynamic conditions. The plot in Figure L12 shows the relationship between the k_G and the u_{Gs} for the 6mm tube. The trend of the lines shows that the mass transfer coefficient decreases as the superficial liquid velocity increases. The literature by Kropholler and Carr [K11] indicates the opposite effect. Difficulties were also encountered to correlate the data in a single line in terms of the parameters ΔP , u_{Gs} , and $(1-\epsilon)$. The correlation which was proposed uses dimensionless groups

in the form:

$$Sh_G = K \cdot Re_G^n \cdot Eu_G^m$$

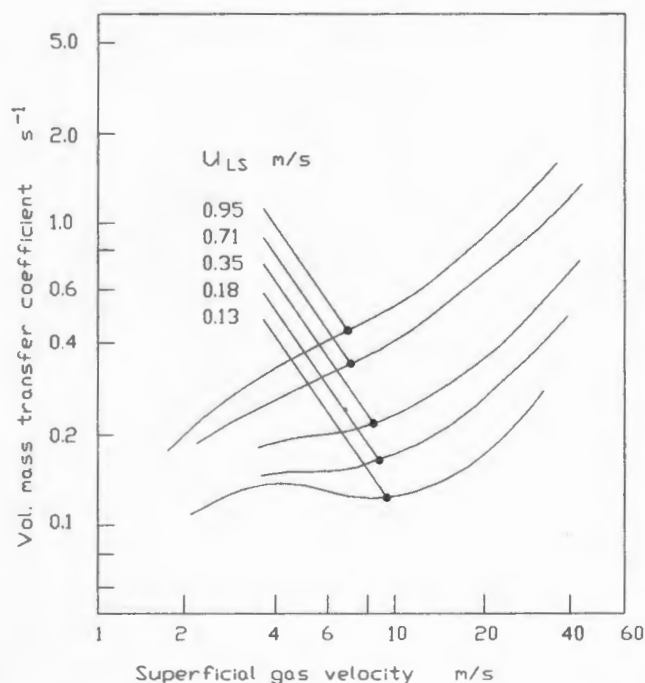
Since the Schmidt number was not varied during their experiment, its influence was included in the constant K . It was also suggested that since the exponent of the Reynolds number was 4.2×10^{-4} , its influence on the Sherwood number was insignificant. Therefore the final correlation was only between the Sherwood and Euler numbers:

$$Sh_G = 22.1(Eu_G)^{-0.39}$$

L3. CORRELATIONS BY TOMIDA ET AL. [T3, T4]

Tomida's work supplements the investigations by Stepanek et al. in a way that the developed correlations for k_L include the effects of the gas-liquid flow rate, tube diameter and liquid viscosity.

The values of k_L were determined from a diffusion model as the values giving the least deviation between the calculated and experimental concentration distribution.



These values were plotted as a function of the superficial gas and liquid velocities u_{GS} and u_{LS} respectively, Figure L13.

This plot reveals that the volumetric mass transfer coefficient increases with the increase of both the superficial velocities.

This tendency is different to that reported by Kasturi and Stepanek [K2] and by Shilimkan and Stepanek [S8, S9] Figures L4 to L7 where the values of $k_L \alpha$ level off or show a maximum when the gas velocity increases. This is apparent in all tube diameters investigated by Stepanek et al.

Figure L 13 Relationship between $k_L \alpha$ and the superficial velocities

regardless of the fact that Tomida's tubes were close in diameters.

As reported in [T3] the main reasons could have been:

- the difference in the flow regimes where the investigations were carried out. Stepanek et al. experimented in the annular to annular-mist region, while Tomida experimented in the region of wavy-annular flow.

The correlations for $k_L\alpha$ in Stepanek's et al. investigations were based on the frictional pressure drop, while Tomida correlated $k_L\alpha$ with Jepsen's energy dissipation factor E_J and with Jagota's frictional energy dissipation parameter, but the later correlation did not fit the data.

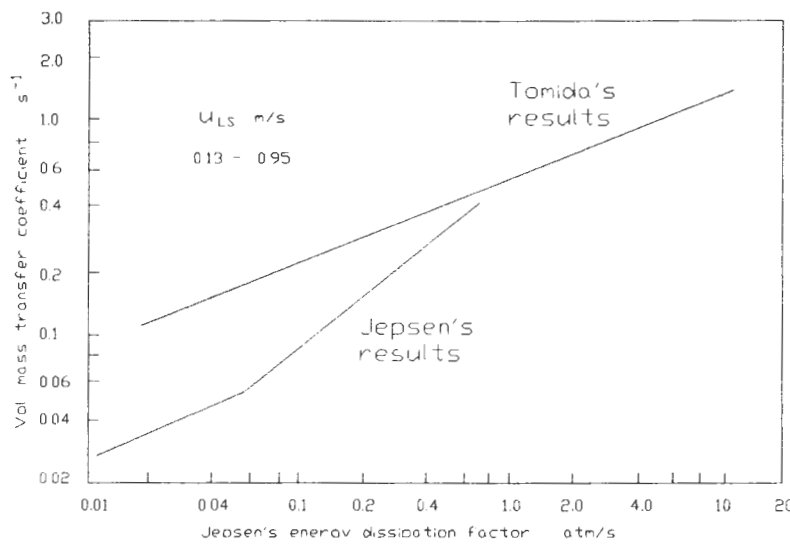


Figure L 14

Figure L14 shows comparatively the correlations with Jepsen's factor as used by Jepsen and by Tomida.

The equation of the straight line is:

$$(17) \quad k_L\alpha = 0.54(E_J^{0.4})$$

It is interesting that the 0.4 power dependence on E_J in vertical flow was equal to that observed by Jepsen [J3] for horizontal and coiled tubes in the range of $E_J < 5000 \text{ N/m}^2\text{s}$.

For values of $E_J > 5000 \text{ N/m}^2\text{s}$, the dependence according to Jepsen is to the 0.79 power, eq. (13) and (14) respectively.

Considering the penetration theory, Tomida et al. developed a correlation similar to eq. (16) proposed by Kasturi and Stepanek [K2]:

$$(18) \quad k_L = \frac{\tau_i}{u_{Li} - u_{Lb}} \sqrt{\frac{D_L}{\mu_L \cdot \rho_L}}$$

where

u_{Li} = liquid linear velocity at the interface

u_{Lb} = liquid linear velocity at the bulk

The interchange velocity of momentum ($u_{Li}-u_{Lb}$) could be obtained from experimental data.

When the momentum interchange Reynolds number $\frac{D(u_{Li}-u_{Lb})\rho_L}{\mu_L}$ is plotted against the liquid viscosity, a straight line correlation was obtained:

$$(19) \quad \frac{D(u_{Li}-u_{Lb})\rho_L}{\mu_L} = 10^4 \cdot \left(\frac{\mu_L}{\mu_w} \right)^{-1.7}$$

where

$$\mu_w = \text{reference viscosity} = 10^{-3} \text{ Ns/m}^2$$

This implies that ($u_{Li}-u_{Lb}$) is independent of the gas and liquid velocities.

With the combination of eq. (17) and (18), together with the correlation of the interfacial shear stress τ_i with the total pressure drop, a relation for $k_L\alpha$ is obtained as:

$$(110) \quad \frac{k_L\alpha}{\sqrt{D_L}} = 10^{-4} \frac{\left(\frac{\Delta P}{\Delta L} \right)_T \cdot \left(\frac{\mu_L}{\mu_w} \right) D_L \cdot \varepsilon \cdot \sqrt{\rho_L}}{\mu_L^{1.5}}$$

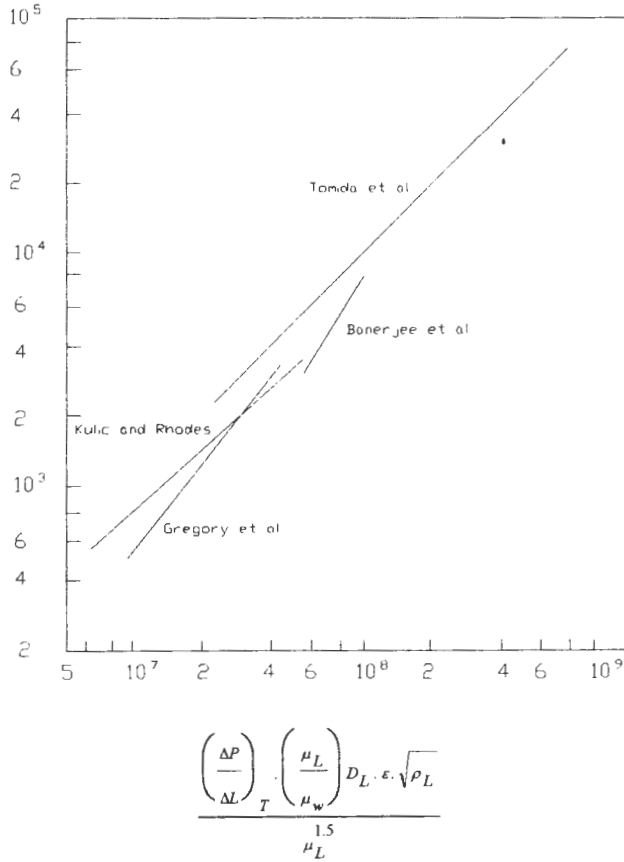


Figure L 15 Correlation of $k_L \alpha$ in terms of the proposed parameters

In Figure L15, the values of the left hand side are plotted against the parameter on the right hand side of eq. (I10). The dependence of $k_L \alpha$ on the pressure drop is slightly different to the other studies. Gregory and Scott [G3] report a power of 1.2 for slug flow in horizontal tubes; Banerjee [B2, B3] reports a power of 1.5 to 2 for annular flow in coiled tubes; Kulic and Rhodes [K13] report a power of 0.84 for slug flow in coiled tubes.

In upward two-phase flow, there are two flow mechanisms, that of the pseudo-laminar flow and that of turbulent. The former appears in the region of $NRe < 1000$ and the latter in the region of $NRe > 1000$ and are both independent of the flow pattern.

Considering eq. (I9), Tomida et al. developed an expression for the turbulent and the pseudo-laminar region in dimensionless form as:

$$\frac{Sh_L \cdot \alpha \cdot D_L}{(Eu_L)(Sc_L)^{0.5}(Re_{LS})\left(\frac{\mu_L}{\mu_w}\right)^{1.7} \epsilon} = 10^{-4} Re_{LS} \quad \text{for the turbulent region}$$

$$\frac{Sh_L \cdot \alpha \cdot D_L}{(Eu_L)(Sc_L)^{0.5}(Re_{LS})\left(\frac{\mu_L}{\mu_w}\right)^{1.7} \epsilon} = 7 \times 10^{-8} Re_{LS}^2 + 0.0035 \quad \text{for the laminar region}$$

These correlations appear not to be affected by changes in the flow patterns. This may be due to the influence on the pressure drop and void fraction neutralising the influence on $k_L \alpha$.